Aspects of the Requirements of ISO 15263.4 and the Design and Development of Bicycle Racks

by

Martin Robson

A thesis submitted in partial fulfilment for the requirements for the degree of MSc by Research at the University of Central Lancashire in collaboration with Pendle Engineering Ltd.

March 2008

University of Central Lancashire

Student Declaration

Concurrent registration for two or more academic awards

Either *I declare that while registered as a candidate for the research degree, I have not been a registered candidate or enrolled student for another award of the University or other academic or professional institution

or ~~*I declare that while registered for the research degree, I was with the University's specific permission, a *registered candidate/*enrolled student for the following award~~:

____

Material submitted for another award

Either *I declare that no material contained in the thesis has been used in any other submission for an academic award and is solely my own work.

or ~~*I declare that the following material contained in the thesis formed part of a submission for the award of~~

____

(state award and awarding body and list the material below):

Collaboration

Where a candidate's research programme is part of a collaborative project, the thesis must indicate in addition clearly the candidate's individual contribution and the extent of the collaboration. Please state below

Signature of Candidate [signature]

Type of Award MSc (by Research)

Department Technology.

Abstract

There is currently interest in developing safety standards for bicycle racks for cars. A draft international standard (ISO 15263.4) has been developed. However, there have been many objections to the proposals and a redraft has been made. This study aims to investigate a part of the standard that involves performing deflection tests on bicycle racks.

The research evaluates the deflection tests, considering how representative and repeatable they are. The results are compared to predictions made using FEA software and calculations of stress and deflection. The required deflection tests were performed on a Pendle Bike Rack.

The results show that the Pendle TBM 3 rack would pass the deflection test part of the draft ISO 15263.4 standard. It is demonstrated that by measuring strain with strain gauges and calculating the stress, the rack operates within 35% of the ultimate tensile strength of the material. The effectiveness of FEA software is demonstrated by achieving similar results from FEA simulations to the experimental data. The application of FEA is explored, using it to simulate changes to the design of the rack to make it stronger and lighter.

The discussion shows that the draft ISO 15263.4 standard needs further development. The draft ISO 15263.4 standard doesn't address issues of fatigue failure, could focus more on ensuring safe use of the product and

is more suited to certain types of bicycle rack. In this thesis it is suggested that further drafts of the ISO 16263.4 standard should incorporate different testing schedules for different types of bicycle rack. Tests for racks that are bolted onto a tow ball should differ from tests for racks that strap onto the tailgate.

Contents

Page

List of Figures 3

List of Graphs 4

List of Tables 5

Acknowledgements 6

1. Introduction 7

   1.1 Background 8
   1.2 Early Safety Concerns 8
   1.3 Aims of the Research 9
   1.4 Methods of Research 10

2. Literature Review 11

   2.1 Types of Bicycle Rack 12
      2.1.1 Car Mountings 12
      2.1.2 Bicycle Mountings 12

   2.2 Historical Development of Bicycle Racks 14
   2.3 Early Safety Concerns 14
   2.4 Draft International Standard 16
   2.5 Development of Standards 18
   2.6 Review of Measurement Technology 21
   2.7 Measurement Terminology 21
   2.8 Measurement Devices 23
   2.9 Finite Element Analysis 23
   2.10 Previous Research in FEA 24
   2.11 Calculating Deflections 26
   2.12 Component Life 27
   2.13 Predicting Fatigue Life 30
   2.14 Strain Gauges 31

3. Methods 33

   3.1 Tests Performed 34
   3.2 Deflection Tests 34
      3.2.1 The Test Bench 34
      3.2.2 Applied Loads 36
      3.2.3 Measurements 39
   3.3 Strain Gauge Measurement 40
   3.4 Finite Element Analysis 41
   3.5 Further Testing with FEA 42

4. Results 44

4.1 Deflection Measurements 45
4.2 Strain Gauge and FEA Measurements 51

5. Discussions 64

5.1 Interpreting Deflection Tests 65
5.2 Strain Gauge Results 69
5.3 Finite Element Analysis 70
5.4 Predicting Failure 75
5.5 Changing the Design 80
5.6 Interpretations of the Standard 82
5.7 Developing the Draft International Standard 84

6. Conclusions 89

6.1 Deflection Tests 90
6.2 Strain Gauge Measurements and FEA 91
6.3 Predicting Failure 92
6.4 Developing the Draft ISO 15263.4 Standard 92

7. Further Work 94

7.1 Improving Fatigue Estimates 95
7.2 Other Types of Rack 96
7.3 Developing Standards 96

References 97

List of Figures

1.1 Early Research Flow Chart 8

2.1 Example of Pendle Bike Racks 13
2.2 Order of ISO 15263.4 Tests 17
2.3 Measurement Systems Block Diagram 21
2.4 Differences Between Accuracy and Repeatability 23
2.5 Bending Moments Diagram 26
2.6 Endurance Limits for Steel 29
2.7 The Gerber, Goodman and Soderberg Relationships 30

3.1 Illustration of test rack. 34
3.2 Photo of the Rack Mount on the Test Bench 35
3.3 Drawing of the Test Bench and Bicycle Rack 36
3.4 Side View of the Vertical Load Test 37
3.5 Drawing of the Longitudinal Test 38
3.6 Drawing of the Lateral Test 39
3.7 Photograph of the Lateral Test 39
3.8 Wheatstone Bridger Circuit Diagram 41
3.9 Sample FEA Output 46

4.1 Vertical Displacement Test 45
4.2 Longitudinal Displacement Test 46
4.3 Lateral Displacement Test 47

5.1 Drawing of the Block Assembly 66
5.2 Sample FEA Output 72
5.3 Gerber Goodman and Soderberg Relationships 77
5.4 Soderberg Analysis of the TBM 3 Rack 78
5.5 Soderberg Analysis of the TBM 3 Rack with Lower Yield Strength 79
5.6 Diagram Showing Tighter Bend Radius Effects 82
5.7 Proposed Structure of an ISO Standard 86

List of Graphs

4.1 Vertical Downward Deflections 48
4.2 Vertical Upward Deflections 48
4.3 Longitudinal Forward Deflections 49
4.4 Longitudinal Backward Deflections 49
4.5 Lateral Left Deflections 50
4.6 Lateral Right Deflections 50
4.7 Strain v Load Downward 54
4.8 Strain v Load Upward 54
4.9 Strain v Load Forward 55
4.10 FEA v Measured Strain Downward 56
4.11 FEA v Measured Strain Upward 56
4.12 FEA v Measured Strain Longitudinal 57
4.13 Comparison between Calculated Deflections and FEA 61
4.14 Comparison of FEA Results for Different Mesh Densities 61
4.15 Comparison of FEA Results for Reduced Wall Thickness 62
4.16 Comparison of FEA Results for Changes in Load Position 63

List of Tables

2.1 Advantages and Disadvantages of Types of Car Mounting 12
2.2 Advantages and Disadvantages of Types of Bicycle Mounting 13
2.3 Significant Bicycle Rack Patents 14
2.4 Survey of Measuring Equipment 23
2.5 Modes of Failure 27
2.6 Characterisation of Failure 27
2.7 Conditions of Fatigue 28
2.8 Advantages and Disadvantages of Using Strain Gauges 32

3.1 Loads Applied During Deflection Tests 36

4.1 Vertical Downward Deflections 45
4.2 Vertical Upward Deflections 45
4.3 Forward Longitudinal Deflections 46
4.4 Backward Longitudinal Deflections 46
4.5 Lateral Left Deflections 47
4.6 Lateral Right Deflections 47
4.7 Strain Gauge Measurements Downward 51
4.8 Strain Gauge Measurements Upward 52
4.9 Strain Gauge Measurements Forward 53
4.10 FEA Results for Downward Vertical Loads 55
4.11 FEA Results for Upward Vertical Loads 55
4.12 FEA Results for Forward Longitudinal Loads 55
4.13 Vernier Calibration Results 57
4.14 Average Vernier Errors 57
4.15 FEA Results for Varying Mesh Density 58
4.15a Standard Mesh 58
4.15b Fine Mesh 58
4.15c Coarse Mesh 58
4.16 FEA Results for Reduced Wall Thickness 58
4.17 FEA Results for 1.5mm Wall Thickness 58
4.18 FEA Results for 1mm Wall Thickness 59
4.19 FEA Results for 3.5mm Wall Thickness 59
4.20 FEA Results for 40mm x 2.7mm Tube 59
4.21 FEA Results for 40mm x 2mm Tube 59
4.22 FEA Results for Increased Bending Moment 59
4.23 FEA Results for Decreased Bending Moment 60
4.24 FEA Results for Decreased Bend Radius 60
4.25 Comparison of Deflection Calculations and FEA 60
4.26 Measurements of the Test Racks 63

Acknowledgements

I would like to thank Ray Smith and all the staff of Pendle Engineering Ltd for the opportunity to do this research and their support throughout the project. I would also like to thank my supervisors, Professor Ian Sherrington, Dr. John Calderbank and Doug Robinson for the help, support and advice during the project. Finally I would like to thank my Mum and Dad, friends and family for their support in getting this far.

Martin Robson

Chapter 1

Introduction

1 Introduction

1.1 Background

Bicycle racks are a popular method of transporting bicycles by car. They are commonly fitted to the roof or to the rear of a car. This research is concerned with the design of rear mounted bicycle racks. There are many safety issues concerned with the transportation of a bicycle on a car. The rack may obstruct the lights, the bicycles may fall off into traffic or the rack may fall off causing damage to the car or a hazard to other motorists. The consequences of an accident where the bicycle or rack falls off the car could be very serious for the bicycle, car and other road users.

1.2 Early Safety Concerns

Clinton (1995) first questioned the safety of racks writing about the risks of incorrectly fitting a rack. In 1996 the Department of Trade and Industry (DTI) started research into the bicycle rack market. They considered the incidence of accidents, the types of rack on the market and typical consumers experiences. They produced an advisory leaflet in the year 2000 to warn about the risks of improperly fitting bicycle carriers.

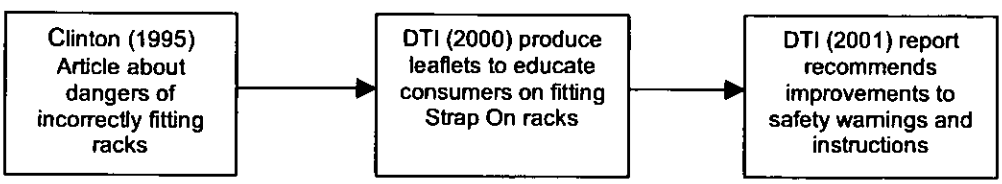


Figure. 1.1. The early pieces of research in bicycle rack safety that led to the creation of the draft ISO 15263.4 Standard.

In 2001 the ISO began to develop a standard for bicycle racks in response to DTI (2001) research, DTI (2001) showed that, as their popularity grew, bicycle racks were becoming more important to road safety considerations. The ISO draft standard ISO 15263.4 published in 2003 aimed to provide a minimum level of safety from racks that meet its requirements.

The standard consists of a set of displacement tests, dynamic tests and a requirement for a range of warnings to be marked on the rack and included in the instruction documentation. The deflection and dynamic tests allow the rack and bicycles to be displaced to specified limits; displacements in excess of these distances cause the rack to not meet the requirements of draft ISO 15263.4.

Previous work to evaluate a different standard was performed by Bevan and De Souza (1990). They evaluated a draft ISO standard for computer menu interface design. They designed several new interfaces that met the standard, then tested them against previously defined criteria on interface quality to see if the standard achieved its aims. They conclude that the standard was hard to interpret and design to but the resulting interfaces did meet most of the evaluation criteria. They also thought that it would be difficult for experienced designers to incorporate the standard in its present form into their established methods of design.

1.3 Aims of the Research

This research project examines whether the draft ISO 15263.4 standard met its aims, considers if it may be improved and how the requirements may be integrated into the design process. If the standard can successfully achieve a minimum level of safety in bicycle racks, it should form the basis of all design and evaluation work involving bicycle racks.

The objectives of the research are:

- decompose the draft ISO15263.4 standard,
- identify the significant portions of the standard,
- devise a testing program,
- design and build the test equipment,
- perform tests described by the draft ISO 15263.4 standard on the Pendle TBM 3 Rack,
- develop, model and perform finite element analysis,
- analyse and compare the results,
- evaluate the work and draw conclusions,
- identify areas for further work.

1.4 Methods of Research

The displacement tests as specified in ISO 15263.4 were performed on the Pendle TBM 3 rack, a 3 bicycle capacity tow ball mounted rack. These tests were repeated using strain gauges to measure strain and calculate stresses during the tests. Finite Element Analysis (FEA) was also performed to model the displacement tests to investigate what role this type of software may play in the design process. FEA could be used to predict performance in testing and reduce the number of tests required during product design and development.

Chapter 2

Literature Review

2. Literature Survey

2.1 Types of Bicycle Rack

A survey of the bicycle rack market using brochures, websites and magazine reviews found that although there are many types of bicycle rack they can be categorised by the method used to attach the rack to the car and the method used to secure the bicycles.

2.1.1 Car Mountings

The two types of car mounting are tow bar mount and strap on. Tow bar mounting racks are clamped around the tow ball. Strap On racks are attached to the tailgate using hooks that fit into the panel gaps around the edge of the tailgate. A summary of the advantages and disadvantages of these types of mountings from the reviews published in MBUK (1999), Singletrack (2005), What Car (2003), Practical Caravan (2003) and Auto Express (1998) are listed in Table 2.1.

Table 2.1. Advantages and Disadvantages of Tow Bar and Strap On mountings.

Tow Bar Mounting	Strap On Mounting
+ Strong and solid mounting on car	+ Quick and easy to fit
+ All tow balls identical	+ Usually cheaper
+ Large capacity tow balls can carry several heavy bicycles	
- Car must be fitted with a tow ball	- Won't fit all models of car
- May be hard to fit	- Care must be taken to get a secure fitting
- Usually more expensive	- Straps must be checked and retightened at regular intervals

2.1.2 Bicycle Mountings

There are two types of bicycle mounting. Hang on mounting comprise two horizontal bars that the frame of the bicycle may be hung on. Wheel support racks comprise a support for the tyres of the bicycle and a support for the frame. A summary of the advantages and disadvantages of these types of mountings again collected from the reviews published in MBUK (1999), Singletrack (2005), What Car (2003), Practical Caravan (2003) and Auto Express (1998) are listed in Table 2.2.

Table 2.2 The Advantages and Disadvantages of hang On and Wheel Support mountings.

Hang On Mounting	Wheel Support
+ Easy to use	+ Fits a wider range of frame shapes
+ Racks are usually lighter and cheaper	+ Kinder to the frames by keeping the bicycles separate
- Bicycles rely on straps for securing	- Usually more expensive or heavy
- May not fit non-traditional frame shapes	
- Bicycles may get scratched due to not being separated	

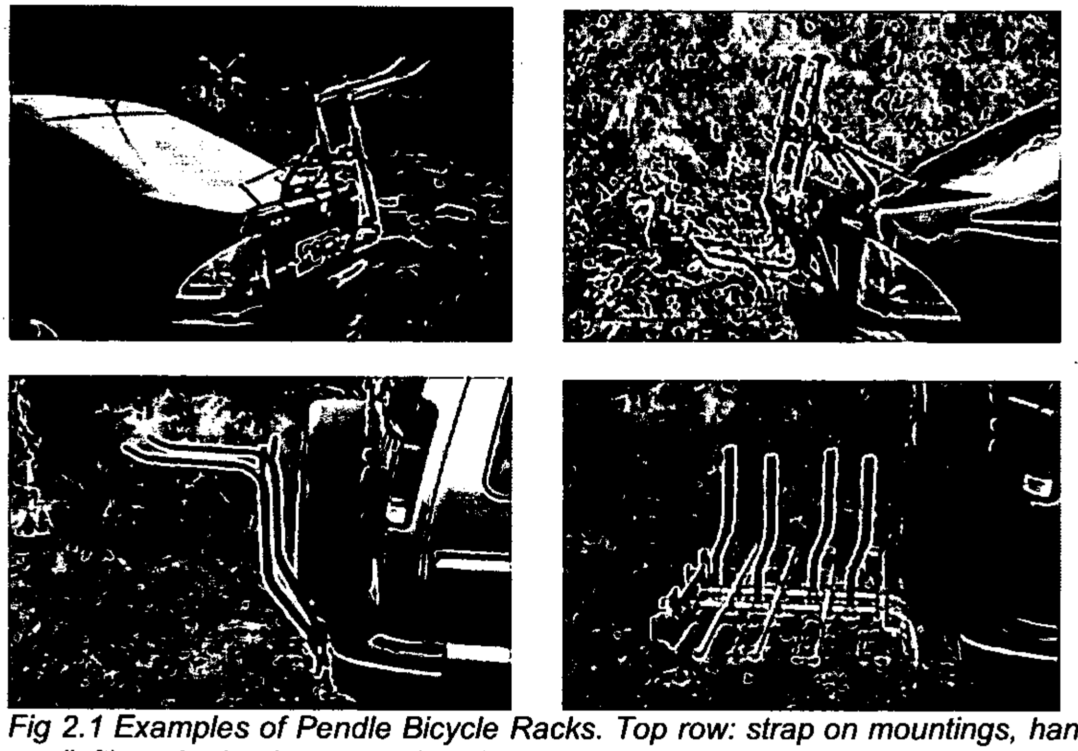


Fig 2.1 Examples of Pendle Bicycle Racks. Top row: strap on mountings, hang on (left) and wheel support (right). Bottom row: tow bar mountings, hang on (left) and wheel support (right).

2.2 Historical Development of Bicycle Racks

A search of patent records on the esp@cenet site, (gb.espacenet.com) reveals the date of many of the important innovations in bicycle rack design. The following Table 2.3 summarises the major developments in chronological order.

Table 2.3 Chronological List of Significant Patents Related to Bicycle Rack Design, results from esp@cenet (www.gb.espacenet.com)

Year	Patent No.	Description
1946	GB581582	Awarded to Frank Schwinn for a bracket welded to a bicycle frame which could be bolted to another bracket welded to the bumper of a car.
1972	US3670935	Awarded to Paul Hinkston for a rack that resembles the modern wheel support rack.
1975	US3923221	Awarded to James Ballinger for a rack which mounts on the bumper, but also uses straps to secure it to the car. This is the first use of straps for securing the rack.
1975	US3927810	Awarded to a company called Leisure Moments for a rack that is strapped to the car's tailgate. This is the first example of a strap on rack.
1980	US4182467	Awarded to Graber for a strap on rack with collapsible arms. This is very similar to modern strap on racks.
1985	NL8304175	Awarded to Cornellis Roordink for a mounting that clamps onto the towball.

2.3 Early Safety Concerns

During the 1990's the first research into the safety of cycle racks appears to have taken place. Clinton (1995) wrote about the importance of fitting strap on racks properly. Clinton (1995) also offers a checklist of precautions to be taken when fitting strap on rack. The checklist covers lighting regulations, maximum capacity, secure fitting and road safety.

The Department of Trade and Industry (DTI) commissioned a report on the safety of bicycle racks. According to DTI (2001) the report was commissioned after a fatal accident in Lincolnshire in 1996 when bicycles fell off a car into the path of a motorcyclist. DTI (2001) indicate that research following this accident found that incidents of detachment were rare, but there was room for improvement in the instruction manuals/leaflets, fitting and checking.

The DTI produced a leaflet titled "Leave Danger Behind" in 2000. This was distributed to consumers at the point of sale of bicycle racks (DTI (2000)). The leaflet presents many of the same suggestions and warnings as Clinton (1995). The emphasis is on the practice of fitting and using the rack rather than the safety of the rack itself. DTI (2000) and Clinton (1995) assume that the design and manufacture of the rack is safe.

The research by the DTI (2001) went further by addressing the safety of the design, the ergonomic problems associated with the design, the information at the point of sale and the instructions supplied. In compiling this information the DTI held group discussions with several groups of potential and current users. They found that there was an element of "out of sight, out of mind" and that people avoided thinking about the consequences of an accident. Their report concluded that most consumers assume that the product has been designed to be safe, but neglected their obligation to use the product safely, although there was much support for improving warnings given at the point of sale and on-product warnings.

DTI (2001) recommended that on product warnings were improved, information relating to safety was made freely available at the point of sale and that the rack manufacturers should raise awareness of safety issues. This research only considered strap on carriers, despite one of the aims being to assess the range of carriers on sale. The report was made available for the development of the draft International standard.

2.4 Draft International Standard

The draft ISO standard for Rear Mounted Cycle Carriers (ISO 15263-4) was published for review and voting in 2003. It stated that the draft standard has been written to

> *"Establish technical specifications and test methods, which offer users of cycle carriers and road users, a minimum level of safety when used in accordance with the manufacturers instructions."*

The draft standard was produced with input from DTI (2001) and national standardisation bodies from 18 countries took part in its development.

The draft standard comprises tests to ascertain whether the rack is safe or not and a requirement for warnings to be printed on the rack covering safe use. The warnings were created directly from the DTI (2001) report and appear unchanged from that report. The ISO committee (Committee ISO TC 22 SC14) responsible for the draft standard designed the practical tests.

The definitions at the start of the draft standard clearly describe different types of rear-mounted rack that are included in its scope. The racks included are "strap on" and "tow bar" mounting in both wheel support and hang on types. The draft standard (ISO 15263.4) is the first significant published work on bicycle rack safety that has included the tow bar mounting type.

The testing part of the standard describes static bench tests and dynamic tests. The dynamic tests involve driving at 25 km/h for 1000m on a Belgian Block Road and passing over a speed bump at a speed of 30 km/h. The dynamic tests involve applying loads of between 2.7 and 3.75 times the maximum load in the x, y and z axes.

In the static tests the residual deflection after loading must be less than 50mm (later revised to 20mm in a 2006 re-draft). The aim of this test is to establish the stability of the rack and the strength of the mounting. The order of testing, stated in the draft standard (ISO 15263.4) is detailed in Figure 2.2.

Lifting Force Test – Vertical loads 3.75 x Maximum capacity.

Residual Deflection < 20mm

↓

Longitudinal Force Test – Longitudinal loads 3.75 x Maximum capacity.

Residual Deflection < 20mm

↓

Lateral Force Test – Lateral loads 2.7 x Maximum capacity.

Residual Deflection < 20mm

↓

Dynamic Tests – Speed Bump at 30 km/h
Belgian Block Road at 25 km/h
Bicycles remain fixed, Residual Deflection < 20mm

Figure 2.2 – Order of testing to draft standard ISO 15263.4.

If any test is failed then it may be repeated up to 3 times before the rack is failed in it's current design. After passing these tests the rack is inspected for the appropriate safety warnings and instruction content.

A vote was taken in 2005 and it failed to gain the required 66% approval score. A re-draft was published in 2006. The main change was a reduction in the allowable residual deflections from 50mm to 20mm. The tests were not changed, although many diagrams were redrawn to illustrate the definitions more clearly. This draft also failed to secure approval at the voting stage in late 2006 achieving a score of 53% approval against the 66% requirement. The British Standardisation Institute (BSI) voted against the standard because they felt that the deflection tests might be superfluous if the dynamic tests had been successful. BSI also felt that if the deflection tests were included, then the loads should be decreased or higher deflections allowed.

2.5 The Development of Standards

Woodward (1972) documented the historical development of standardisation. Standardisation began in 1840 when Whitworth standardised nuts and bolts because threads used by different manufacturers were not interchangeable with each other. Woodward (1972) stated that the benefits of standardisation include: reductions in design time, costs and improvements in quality. After Whitworth developed a standard for nuts and bolts in 1840 and American organisation called "American Sellers" produced their own standard for nuts and bolts, which differed completely from Whitworth's. Woodward (1972) goes on to explain that eventually many items were standardised during the late 19th century, but these standards only had a national scope at best.

In 1901 the British Standardisation Institute (BSI) was founded. According to Woodward (1972) due to Britain's powerful position in the world of politics and industry many other countries adopted BSI standards for their own national standards. He suggests that value of this system was seen during the First World War when Belgian guns could use shell casings manufactured in England.

After the Second World War, in 1946, the International Organisation for Standardisation (ISO) was founded in Switzerland to develop internationally recognised standards on behalf of the various national standardisation bodies.

Woodward (1972) categorized standards into 4 types:

> *"Glossary of Terms – A standard that lists terminology to be used in a particular industry or for describing certain products.*
>
> *Dimensional – A standard that specifies the dimensions for particular components or products so that they fit together with other components or products.*
>
> *Performance – A standard that specifies performance parameters for a product.*

Testing Method – A standard that describes a testing process that a product must pass."

A standard may fit into one or more of these categories.

The ISO website (www.iso.org) lists the stages of the process of developing a standard.

1. *"Initially an industry or regulatory body must suggest the idea of a standard to their national standards body.*
2. *The national standards body will put the standard forward to ISO who will assign a technical committee to the project.*
3. *The technical committee will produce a draft international standard (DIS) and circulate this to national bodies, members of the industry and the public.*
4. *Anyone who reads the DIS may submit comments to their national standards body.*
5. *The national standards body will vote for or against the standard and submit a summary of comments.*
6. *If the standard is voted for, changes may be made based on the comments submitted before a final vote and publication.*
7. *If the standard is voted against then the committee may re-draft and re-publish up to 3 times."*

2.6 Review of Measurement Terminology

In this work the load applied to the bicycle rack produces the input displacement. The output value is the measurement of the displacement. The measurement system is the method chosen to measure the displacement.

Bentley (1996) describes a measuring system as one that takes an input, processes it and outputs a value.

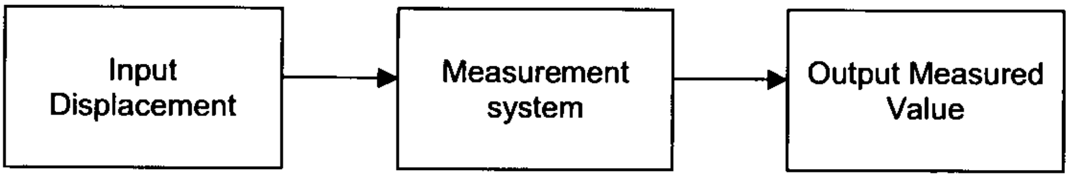


Figure 2.3 A measurement system as described by Bentley (1996) applied to the measurement of displacement.

The measurement system must be chosen to provide a suitable measurement with minimum error. Kirkup (1994) and Ray (1988) state experiments should be designed so that error is minimized. Before specifying a measuring instrument, some terminology related to measurement will be defined.

2.7 Measurement Terminology

The following paragraphs define various terms associated with measurement that are relevant to the design of the test apparatus..

Error – Hayward (1977) states that error only applies to a measurement and not an instrument. An instrument cannot have an error. The error is the difference between the measured value and the true value. Hayward (1977) and Bolton (1991) state that error may be systematic or random.

A random error is defined by Bolton (1991) as "the errors that may occur due to environmental changes such as temperature or a change in the operator and their methods". Systematic error is defined by Bolton (1991) as being "within the instrument". For example, "the needle on a dial may not return to zero when the input is at zero."

Bolton (1991) offers solutions to both classes of error. Repeating the measurement and taking mean averages will reduce random errors. Hayward (1977) says as a rough guide the uncertainty of the mean may be taken to be $1/\sqrt{n}$ of the uncertainty of the single values, where 'n' is the number of results. Bolton (1991) states that calibrating the instrument by comparing like for like readings with other instruments minimizes systematic error.

Repeatability and Accuracy – Hayward (1977) defines repeatability as the ability of an instrument to give identical responses for repeated application of the same value. Repeatability may also be known as precision. The accuracy is the instrument's ability to measure to the true value. Hayward (1977) compares the two properties as "Accuracy is the instrument's ability to tell the truth, repeatability is its ability to stick to it."

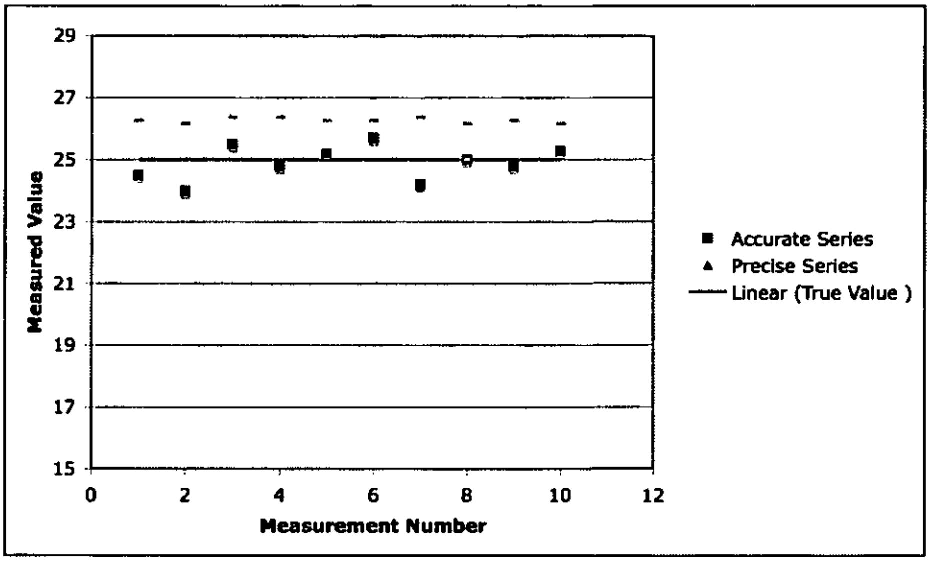


Figure 2.4 The differences between Accuracy and Repeatability. The accurate data series points are closer to a true value, the precise data series points are closer to each other but not necessarily to a true value.

2.8 Measurement Devices

A survey of measuring devices was conducted, Table 2.4 summarises this survey.

Table 2.4 Summary of a Survey of Measuring Equipment.

	Cost	Range	Accuracy	Resolution
Ruler	£10	0-150mm	0.5mm	0.5mm
Vernier Calliper	£20-250	0-150mm	0.02mm	0.1mm
Digital Vernier Calliper	£30-250	0-150mm	0.02mm	0.01mm
Dial Test Indicator (DTI)	£20-200	0-25mm	16 µm	0.1mm
Digital DTI	£100-500	0-25mm	0.003mm	0.001

2.9 Finite Element Analysis (FEA)

FEA is often used in modern design processes. As manual drafting has been replaced by CAD packages, FEA software is becoming available as a supplement to calculations and testing. The Pendle TBM 3 rack was designed using CAD software, the next step is to test the design using FEA software.

An FEA of a bicycle rack design starts with an accurate CAD model of the rack. The Autodesk Inventor software used by Pendle uses a built in analysis tool. The restraints and loads must be defined, the mesh is created automatically although the user may control the size of the mesh to allow for faster solutions or a potentially more accurate result.

McMahon and Browne (1998) explain that the more elements there are in the mesh the more accurate the solution is likely to be. More elements will take longer to process or require a more powerful computer. Rooney and Steadman (1993) state that as computer power increases then more complex solutions will become possible or simpler solutions may be split into smaller elements increasing accuracy. This prediction should be accurate because the rate of improvement in microprocessor design is governed by Moore's Law, according to Intel (2007). Gordon Moore was the founder of Intel who suggested in 1965 that the number of transistors in a processor would double every 24 months while still remaining economically viable. This law has remained viable up to now and Intel (2007) predict that it will remain viable for several years.

2.10 Previous Research in FEA

Many previous researchers have looked at the application of FEA, often comparing it to experimental data. The following describes a selection of this work. Ray (1988) considered applications in roadside crash barrier design and found that the solutions could be quite accurate, though it was important to make sure the material properties used accurately reflected those of the real crash barrier and not the claimed properties of the material.

Cristofolini and Viceconti (2000) investigated the possibilities of using FEA to model the stresses in artificial hip joints. They conducted experiments on the joints taking measurements using strain gauges. Then they replicated the experiments in FEA and compared the results. They found that there were small errors in the FEA but calculated correction factors that could be incorporated into the software to improve the results.

Yahiaoui et al (2001) modelled the effect of cracks in bent pipes carrying hot water. They utilise a technique called node separation that allows cracks to be modelled as breaks in the mesh. There is no physical gap between the elements of the mesh but the software considers them to be unattached.

Berghini and Betini (2001) compared strain gauge measurements with FEA for drilled holes under stress. They used ANSYS software with a variable geometry mesh, this creates smaller elements near the hole with larger ones further away to achieve a compromise between high accuracy and fast processing time. They achieved an accuracy of 0.1%.

Kaye and Heller (2001) documented the development of a new bulkhead in the F/A 18 fighter jet. The bulkhead from a production F/A 18 was analysed, then redesigned to reduce weight in low stress areas and increase strength in high stress areas. The results was a bulkhead that was lighter but also had a 27% reduction in hoop stress.

2.11 Calculating Deflection

Hartog (1962) describes the Myosotis formulae for the angular and linear deflection of a cantilever. Applying these to the simplified bike rack of Figure 2.5 gives a vertical deflection:

$$\text{Deflection} = \frac{PDL_1}{EI} * L_2 + \frac{PD^3}{3EI} + \frac{PD^2}{2EI} * (L_2 - D)$$

Where E is Young's modulus and I is the 2nd moment of area. P, D and L are defined in figure 2.5 on page 28.

The results of these deflection calculations may be compared with the measurements made during deflection tests and FEA predictions.

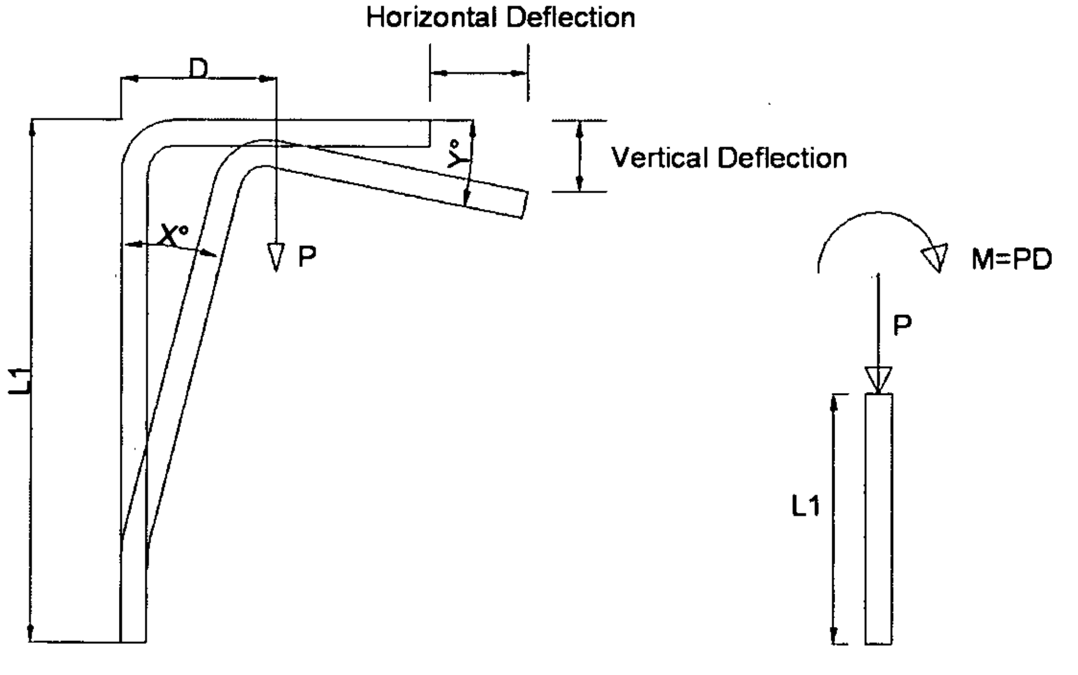


Figure 2.5 Diagram of Bending moments and deflections.

In this case there is a deflection due to the moment M=PD and the load P.

4 of the 6 Myosotis Formulae will be used, they are, Hartog (1962)

Loading	Angle at the free end, $\frac{dy}{dx}$	Deflection at free end, y.
End moment, M.	$\frac{ML}{EI}$	$\frac{ML^2}{2EI}$
End load, W	$\frac{WL^2}{2EI}$	$\frac{WL^3}{3EI}$

2.12 Component Life

Lewis and Samuel (1989) define product lifetime as "the period of time something performs successfully." It will depend on the stresses and strains it is subjected to. They state that the probability of failure must be reduced to an acceptable value. Lewis and Samuel (1989) go on to classify modes of failure. They list 3 modes of failure as summarised in Table 2.5.

Table 2.5 Modes of Failure classified by Lewis and Samuel (1989).

Mode of Failure	Causes
Fracture	Excessive static loading, dynamic loads (fatigue), impact, corrosion.
Excessive Deflection	Elastic or Plastic deflections in excess of intended performance
Wear or Damage	Wear through use or damage from misuse renders the item unable to perform as intended

Dym and Little (2004) present a characterization of failures, these are listed in Table 2.6.

Table 2.6 Characterization of Failure from Dym and Little (2004)

Failure Characteristic	Description
In Service	Breaks while in use
Incidental	Wears out during use
Catastrophic	Complete System Breakdown

When considering the life of a bicycle rack, any of the listed modes of failure could occur. Apart from static loading, cyclic loading or fatigue is another high risk of failure for a bicycle rack. The nature of carrying bicycles on a car means that the rack is subject to dynamic loading due to road surface and the movement of the car. In the case of a bicycle rack, an incidental failure may be acceptable or indeed unavoidable, but an in service or catastrophic failure must be minimized.

Fatigue failure would be catastrophic for a bicycle rack. This draft ISO standard (ISO15263.4) is designed to prevent failures but does not include fatigue. Lewis and Samuel (1989) list the conditions of stress that may be present under dynamic loading. Table 2.7 lists these conditions of stress and gives examples of whether they would be present in a bicycle rack.

Table 2.7 Conditions of Dynamic Loading that would lead to Fatigue Failure presented by Lewis and Samuel (1989).

Condition	Example of Occurrence on a Bicycle Rack
Sinusoidal	Car travels at constant speed over equally spaced expansion joints.
Fluctuating Stress	Car travelling on a road that dips and rises causing a fluctuation in the vertical load.
Repeated Stress	Cornering loads applied as car travels along a winding road.
Reversed Stress	Lateral loads left and right on a twisty road.
Varying Sinusoidal	Car changes speed whilst travelling over expansion joints or the severity of the joints changes.
Non Sinusoidal (Random)	Any changes in load due to movement of the car or changes in road surface.

Table 2.7 suggests that all conditions of stress may be found in a bicycle rack, suggesting that a bicycle rack would see a high probability of fatigue failure. It is well known see for instance, Shigley (2004), Lewis and Samuel (1989) or Suresh (1991) that steel exhibits a fatigue life and has an endurance limit. This is a stress below which the material will statistically never fail in fatigue and

exhibit infinite fatigue life. Different steels exhibit different endurance limits, Shigley (2004) states that they are usually between 0.35 and 0.6 times the UTS.

However, corrosion will result from exposure to moisture and road salt and failure could follow. Shigley (2004) discusses the reduction in fatigue life due to corrosion. As the material corrodes the component becomes thinner, increasing the stress. As the stress increases it will get closer to the endurance limit. Eventually the stress may exceed the endurance limit causing a fatigue related failure at the corresponding number of cycles. A component that was designed to avoid fatigue failure may also fail if corrosion is not considered.

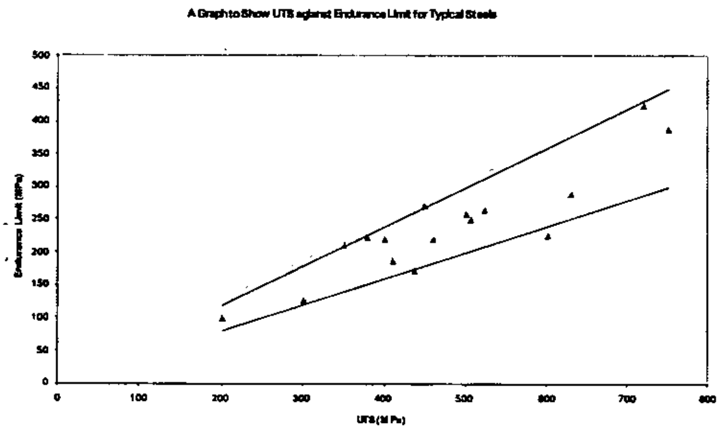


Figure 2.5 Shigley (2004) shows that Endurance limit is typically between 0.35 and 0.6 times the UTS the area in this region is enclosed by the black lines. The points show experimental results collected by Shigley (2004) for various steels.

Lewis and Samuel (1989) discuss designing for a safe life. The product must be designed to have a predicted life in which the designer must be confident. This may require a level of conservatism about predictions of fatigue life. Shigley (2004) and Lewis and Samuel (1989) discuss the merits of experimental fatigue life predictions. They conclude that whilst calculations may be made based on estimates of the endurance limit of the material, many factors may influence the fatigue life so samples of material of the component are tested at various frequencies.

2.13 Predicting Statistical Fatigue Life

Osgood (1970) and Suresh (1991) document the development of fatigue life predictions. According to Suresh (1991), Wohler first observed fatigue in 1806 when designing railway axles. Osgood (1970) states that Gerber suggested a parabolic relationship between the endurance limit and the UTS of the material. If a plot of the mean stress against the peak stress falls under the curve then the component will have an infinite fatigue life. Suresh (1991) considers the Gerber relationship accurately reflects experimental data for brittle steels, but it is less accurate for ductile materials. See Figure 2.6.

According to Osgood (1970) Goodman proposed that a straight Line be drawn between endurance limit and UTS. Suresh (1991) says that this relationship proves conservative for brittle materials, but closely matches the behaviour of more ductile materials. See Figure 2.6.

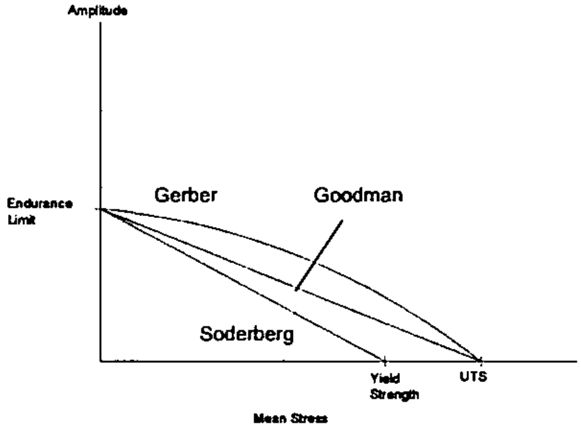


Figure 2.7. Illustrates the Gerber, Goodman and Soderberg Relationships.

According to Suresh (1991) Soderberg proposed a relationship in 1939 which modified Goodman's straight line by plotting the endurance limit against the yield strength. See Figure2.4. Both Suresh (1991) and Osgood (1970) consider this to be a more conservative approach, but it will work with almost any alloy.

2.14 Strain Gauges

The techniques associated with using strain gauges are well documented, see for instance Bentley (1995) or Measurements Group (1992) for descriptions of the techniques of bonding gauges. In summary, the gauge must be bonded fully to the surface of the metal with the correct alignment and in the correct position.

According to Dally and Riley (1991), Lord Kelvin was the first person to notice that the resistance of a wire changes with strain. They also discuss the effects of temperature on a gauge. As the material heats up it will expand and subject the gauge to a strain. This temperature effect has to be compensated for. In this case the bridge circuit, a half bridge is self-compensating because two gauges in tension and compression on either side of the tube create the voltage difference. Both gauges are bonded to the same component so they will be subject to the same expansion. Therefore the bridge output will not be affected by temperature. For the case where the fixed resistors form the other half of the bridge, strain gauges will be bonded to an unstrained sample of material so they will be compensated too. Dally and Riley (1991) also recommends that the gauges used are of the type that are compensated for the material being used. This means that the gauge has the same co-efficient of expansion as the material onto which it is being bonded.

Table 2.8 summarises the advantages and disadvantages of using strain gauges.

Advantages	Disadvantages
+ Produce accurate measurements of the strain at a point on a component.	- Must be bonded correctly and in the correct position
+ Stress may be calculated from the strain so long as Young's modulus for the material is known	- Techniques need to be learnt and mastered for accurate repeatable results
+ Many types of gauge, rosette and bridge circuit are available for different situations	- Temperature compensation must be included
+ Different sizes and material of gauge allow the correct one for the application to be chosen	- The gauge can only measure at the point it is bonded to. Point must be chosen carefully
+ Can be used in service for constant monitoring of critical components during their lifetime.	

Chapter 3

Methods

3 Methods

3.1 Tests Performed

The deflection tests described in the draft ISO 15263.4 standard were performed. Deflection was measured under loading as well as the residual deflection measured as required by the draft standard (ISO 15263.4). The tests were repeated whilst the strain was measured with strain gauges so that the stresses may be calculated. Finally the deflection tests were modelled using FEA software.

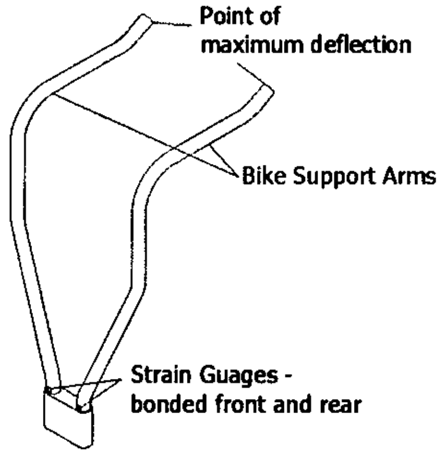


Figure 3.1 The test racks were Pendle TBM 3 racks, a tow bar mounted 3-bicycle capacity hang on rack. Maximum load capacity is 45 kg or 15 kg per bicycle. The rack is manufactured from high tensile 31.75mm tube with 2.7mm wall thickness.

3.2 Deflection Tests

The following section describes the methods used to perform the deflection tests described by the ISO 15263.4 draft standard.

3.2.1 The Test Bench

The bicycle rack was mounted on a large steel bench measuring 4m x 2m, the bench was bolted to a concrete floor. The bench is used for mounting welding fixtures and other jigs. A 90 degree cast steel webbed block was bolted to the bench using three M16 bolts, which were tightened to a torque setting of 249 Nm. A standard 50mm two bolt tow ball was bolted to the vertical face of the steel block. This used two M16 bolts tightened to a torque setting of 249Nm.

The rack was fitted to the tow ball following the instructions supplied with it. The ball-clamp mounting block was bolted onto the tow bar using four M10 bolts, with nuts and spring washers, tightened to a torque of 59 Nm. The arms of the bicycle rack were inserted into the sockets of the mounting block and secured in place using an "R" clip. The assembly is shown in Figure 3.2 and 3.3.

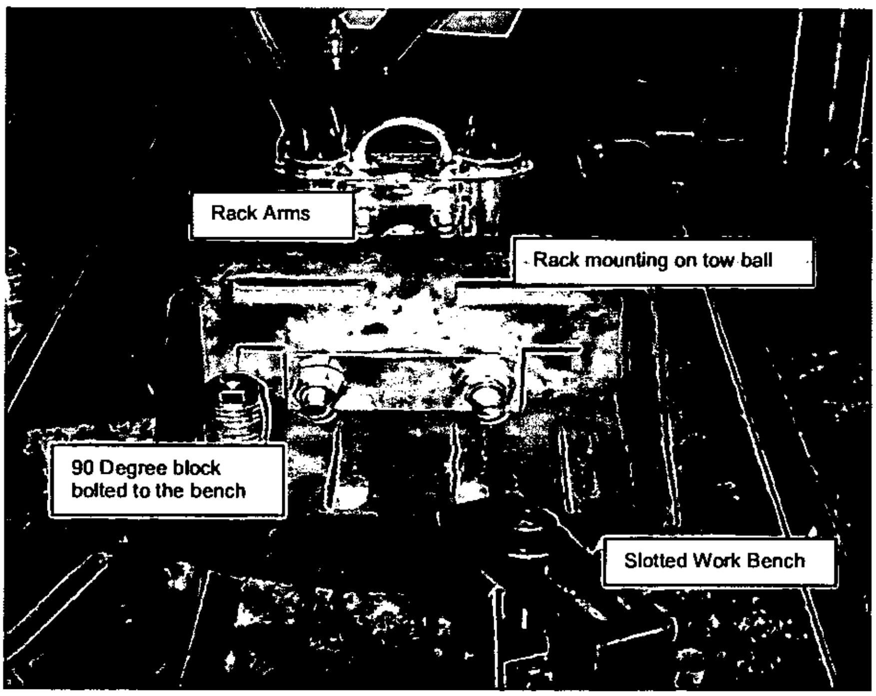


Figure 3.2. A photograph of the rack mounted on the workbench seen from behind.

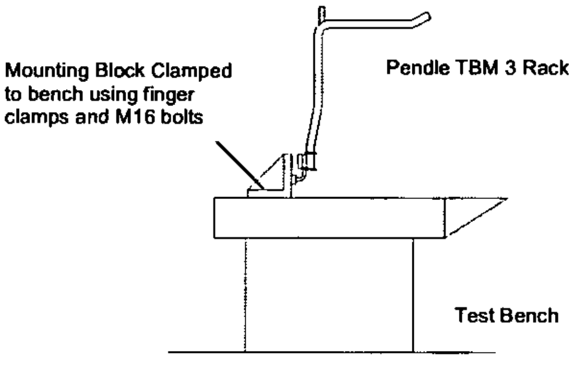


Figure 3.3 Drawing of the side view of the rack, tow ball and bench.

3.2.2 Applied Loads

The loads that were applied to the rack are listed in Table 3.1 below.

Test Type	Load Calculation	Load (kg)
+Z Vertical Downward	3.75 x 45kg	169
-Z Vertical Upward	3.75 x 45kg	169
-X Longitudinal Forward	3.75 x 45kg	169
+X Longitudinal Backward	3.75 x 45kg	169
+Y Lateral Right	2.7 x 45kg	121.5
-Y Lateral Left	2.7 x 45kg	121.5

Table 3.1. Loads applied in the 6 deflection tests.

The test bench arrangement was changed to allow the application of the loads in different directions. The loads were applied from a collection of weights mounted on a pallet. The pallet was lifted to release the tension before being connected to the appropriate part of the rack. The pallet was released using a forklift tuck to apply the load to the rack. The load was allowed to stabilise for 30 seconds then the measurements were taken. After measurement the pallet was lifted to release the load from the rack.

Vertical Deflection +Z,-Z

The rack was fixed near to the edge of the bench so that the bicycle support arms hung over the edge of the bench. The load was applied to the rack using a 600mm x 50mm x 50mm block that was strapped to the load by a 50mm wide webbed strap. A displacement-measuring plane was situated underneath the end of the horizontal part of the bicycle support arms. Fig 3.4 illustrates the arrangement. The load applied was 3.75 x the 45kg maximum load of the rack.

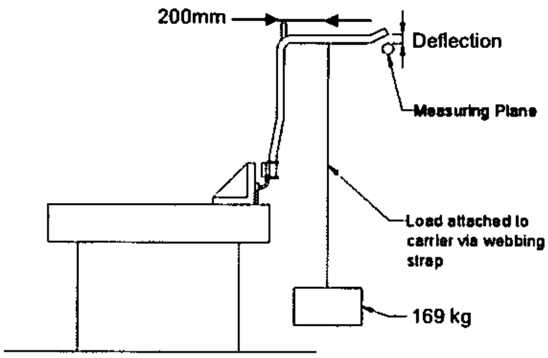


Figure 3.4. Side view drawing of the vertical displacement test.

Longitudinal Deflection +X, -X

An adjustable H shaped frame was used to transfer the vertical load into a longitudinal one. The frame was made from two vertical sections of 50mm Unistrut with 3mm walls, welded to a 5mm thick base plate. A 50mm diameter roller was bolted between the two sections of the 50mm Unistrut at a height of 400mm from the towball. This was applied to the load at a height equivalent to the centre of gravity of the bicycles when the rack was loaded. The measuring plane was situated behind the rack and consisted of a 50 x 50 x 5mm thick box section welded to a 5mm plate, which was bolted to the bench using M16 bolts. A 45 degree brace was used to reduce deflection in the 'H' shaped frame. This brace was made from a 1m length of 50 x 50 x 5mm box section bolted to the

'H' frame at the top and bench at the bottom via 5mm brackets that were welded to it. Figure 3.5 Illustrates the test bench arrangement.

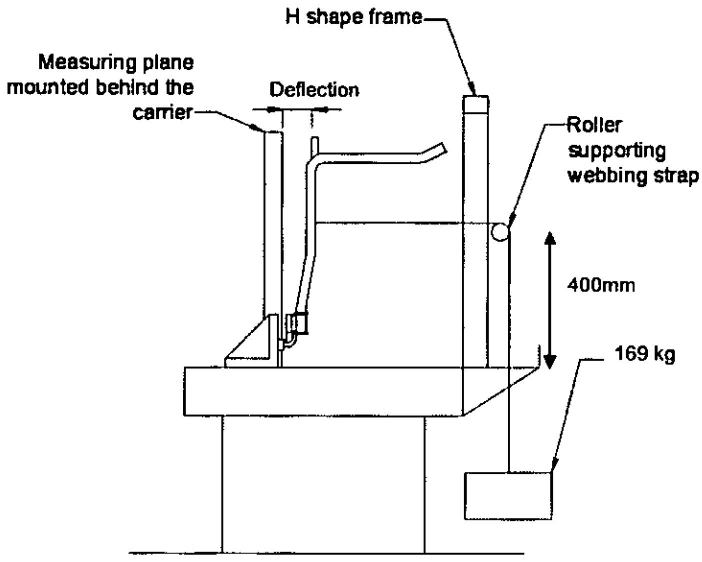


Figure 3.5. Side view drawing of the longitudinal test.

Lateral Deflection +Y, -Y

The lateral deflection tests were performed using a similar method to the longitudinal deflections. The H shaped frame was mounted perpendicular to the rack. A mock up bicycle frame made from 50mm diameter 1.5mm wall tube was strapped to the rack 250mm from the start of the horizontal supports and 400 mm above the towball to replicate the position of the centre of gravity of a fully loaded rack. The measuring plane was the same one used for vertical deflection but mounted perpendicular to the end of the horizontal support arms. Figs. 3.56and 3.7 Illustrate the arrangement of the lateral deflection tests.

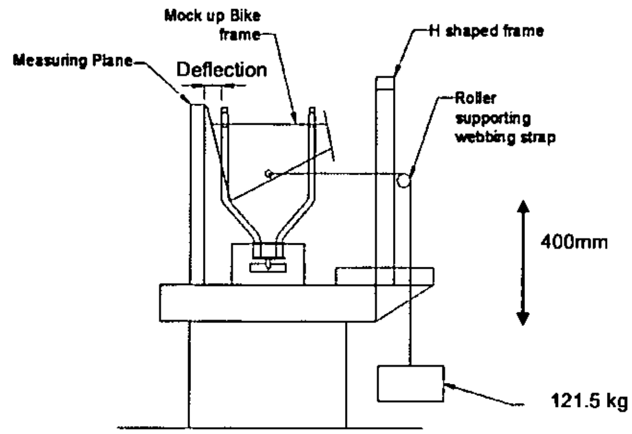


Figure 3.6 Side view drawing of the lateral test.

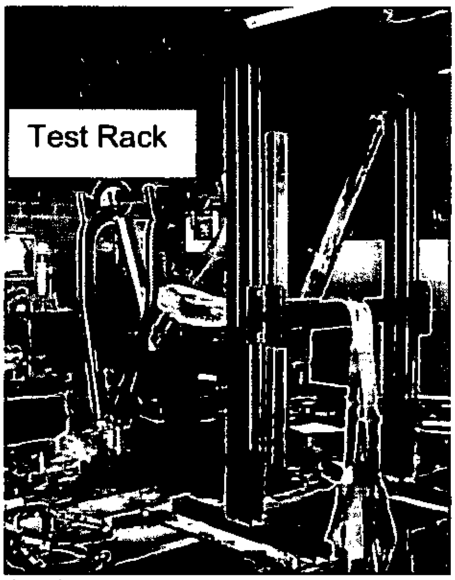


Figure 3.7. A photograph of the lateral test. H shaped frame is at the front of the sitting perpendicular to the rack.

3.2.3 Measurements

All tests were performed 3 times. Deflection was measured during loading as well as the residual deflection required by the draft standard (ISO 15263.4). Deflection was measured using a digital Vernier calliper. In trial runs this proved the most effective method of measuring displacement. Measurements were taken between the point of maximum displacement on the rack and a fixed measuring plane. Each measurement was performed and recorded 3 times to minimize random errors. The digital Vernier calliper was calibrated by measuring an 8mm slip gauge three times using the digital Vernier. This measure was compared to values recorded by another digital Vernier and a DTI.

3.3 Strain Gauge Measurement

The vertical and longitudinal deflection tests were repeated. This time strain was measured using strain gauges. Bending moment calculations predicted that the strain would be highest under the longitudinal loads, see results table 4.9. Vertical loads were tested because the point of maximum stress would be in the same position for both tests so it could be measured with the same strain gauges.

Strain gauges were bonded to the front and back of the rack arms above the point where they fit into the socket. Normal procedures for strain gauge bonding were followed (see RS (1997) or Dally and Riley (1991) for instance). The gauges were connected to a Wheatstone bridge circuit as shown in Figure 3.8. The bridge output was amplified and displayed using an Amplicon MAG-35 strain gauge amplifier with built in 3 ½ digit digital display. The amplifier has 10V excitation voltage and 4 inputs to read from one bridge circuit. The loads were applied in increments of 10kg from 10kg to 200kg.

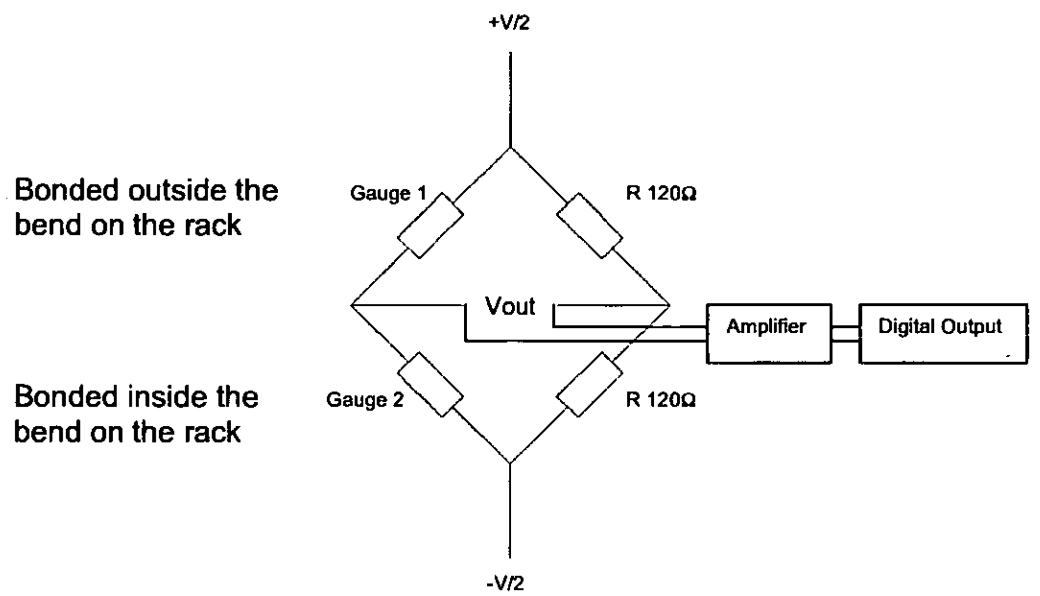


Figure 3.8. Wheatstone bridge circuit output to amplifier and digital display. The circuit is a half bridge circuit, using 2 gauges in compression and tension.

The gauges used were RS 5mm strain gauges temperature compensated for steel with 30mm leads to help prevent heat damage during soldering. The gauges were bonded according to the instructions given by RS (1997). The surface was cleaned using a fine emery cloth to remove the powder coat, then wiped using a lint free cloth with RS Solvent Cleaner as recommended. The gauges were bonded to the metal using the Cyanoacrylate adhesive recommended by RS (1997).

3.4 Finite Element Analysis (FEA)

An FEA was performed using AutoDesk Inventor software. The FEA replicated the conditions of the deflection tests. Results were recorded for deflection and stress for comparison with the physical test results.

A solid model of the rack was created with Autodesk Inventor. The arms were modelled by creating a sweep of the tube profile along a 3d sketch of the tube centreline. The arms of the rack were modelled in a solid block to represent the mounting block. A beam was modelled at the position in which the load would be applied. The beam was moved depending on the direction of the load.

Standard procedures for performing the analysis were followed (See AutoDesk (2007)). The rack was restrained at the mounting block for each test. Figure 3.9 illustrates the FEA output. The mesh was varied by density, the software automatically set the size of the elements which were triangular with 3-5mm sides. The solution times were typically around 1 minute with little difference in speed when using different mesh densities.

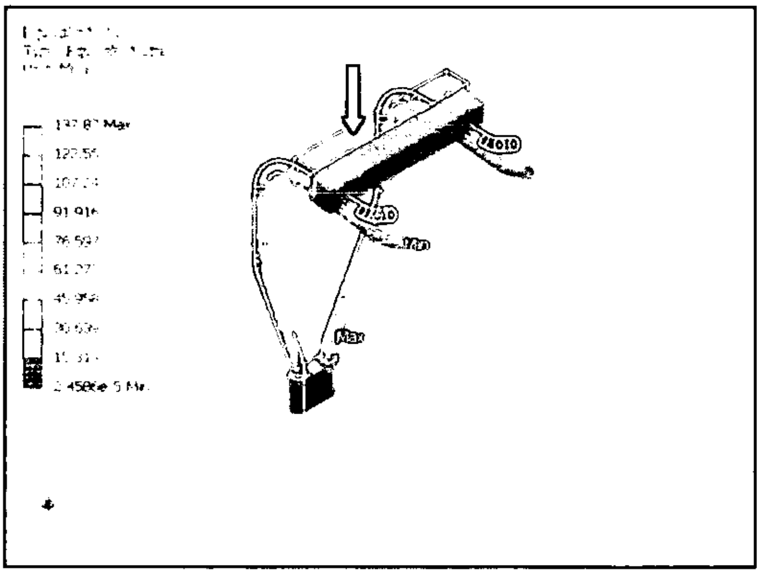


Figure 3.9. FEA output for stress produced for a vertical deflection test. The force was applied to the beam on top of the arms, shown here by the arrow. Red shading represents high stress, blue shading represents low stress. The white outline is the un-deformed model outline. Deformation is exaggerated for visibility.

3.5 Further Testing Using FEA

The FEA was used for further tests, to explore changes to the interpretation of the testing procedures and basic changes to the design of the rack. The interpretation of the tests was based on the position of the load applied to the rack. This may change for different designs of rack or be changed to influence the results of the test. Tests were performed to simulate a reduction and increase in the bending moment. The vertical load test was used, the load applied at +/- 75mm from the centre of the support arm. All boundary conditions were left unchanged.

Other analyses were performed to consider the effect of changing the material's wall thickness. The material was changed from 31.8mm x 2.7mm tube to 31.8mm x 2mm tube. This results in a 35% cost saving on material with no changes to the manufacturing processes involved. The vertical test was performed using the standard model and boundary conditions.

Other analyses focused on finding an ideal tube size to achieve similar or reduced stresses and deflections whilst saving weight. Variations of wall thickness and overall tube diameter were tried.

Larger diameter tubes were tested to determine whether it is possible to reduce weight whilst maintaining the current stresses and deflections. Another analysis used a tighter bending radius of 50mm. This should move the load closer to the car because the bicycle support will be nearer to the towball, reducing the bending moment on the rack.

Chapter 4

Results

4. Results

4.1 Deflection Measurement Results

Vertical Loads

Fig 4.1 Illustrates the vertical displacement test. Refer to method section 3.2.2 for full description.

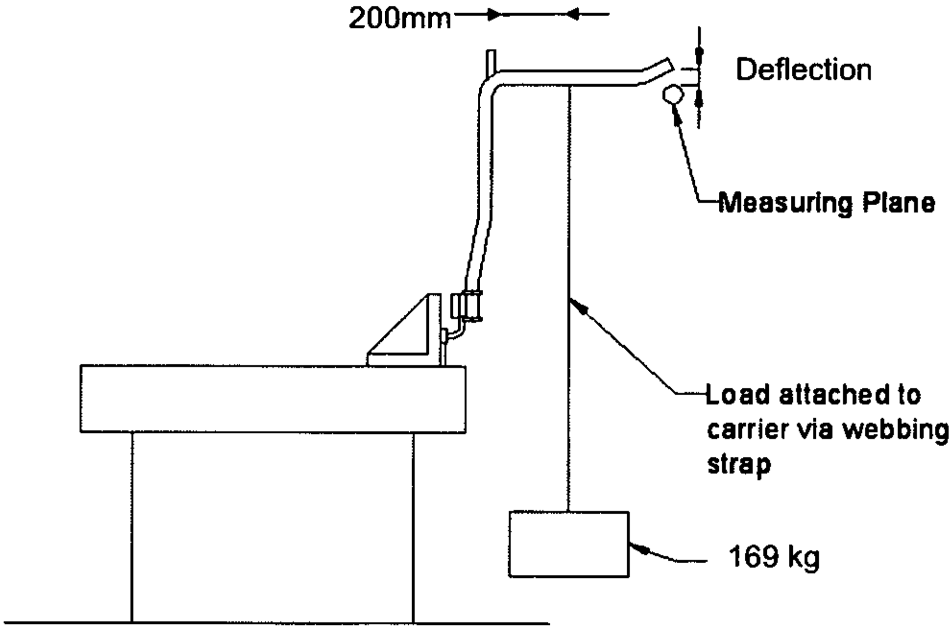


Table 4.1. Vertical Downward Deflection. 3 Racks used to allow for manufacturing differences. 3 separate racks were tested to show any manufacturing differences and repeatability.

Test	Loaded Deflection (mm)	Residual Deflection (mm)
Rack 1	13.51	-0.01
Rack 2	13.47	0.82
Rack 3	13.67	0.00
Average	13.55	0.27

Table 4.2. Vertical Upward Deflection

Test	Loaded Deflection (mm)	Residual Deflection (mm)
Rack 1	-19.32	-1.02
Rack 2	-22.19	-0.69
Rack 3	-18.71	0.20
Average	-20.07	-0.50

Longitudinal Loads

Fig 4.2 Illustrates the longitudinal displacement test. Refer to method section 3.2.2 for full description.

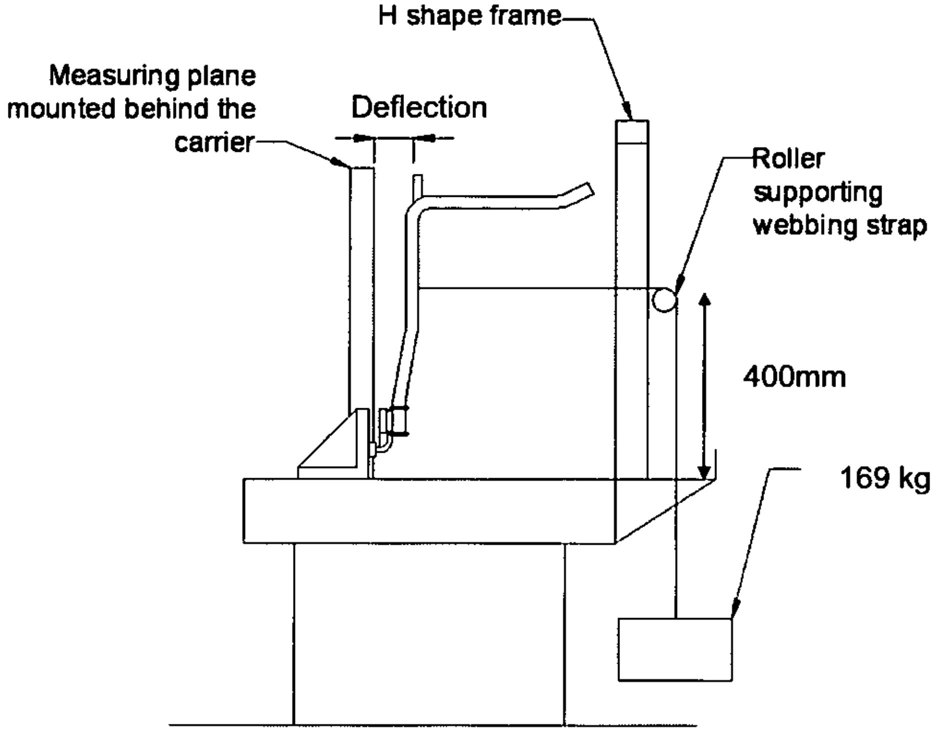


Table 4.3. Longitudinal Forward Deflection

Test	Loaded Deflection (mm)	Residual Deflection (mm)
Rack 1	-27.01	-1.80
Rack 2	-30.06	-0.65
Rack 3	-26.34	0.65
Average	-27.80	-0.60

Table 4.4. Longitudinal Backward Deflection

Test	Loaded Deflection (mm)	Residual Deflection (mm)
Rack 1	15.89	0.58
Rack 2	14.63	1.07
Rack 3	14.18	0.58
Average	14.90	0.74

Lateral Loads

Fig 4.3 Illustrates the lateral displacement test. Refer to method section 3.2.2 for full description.

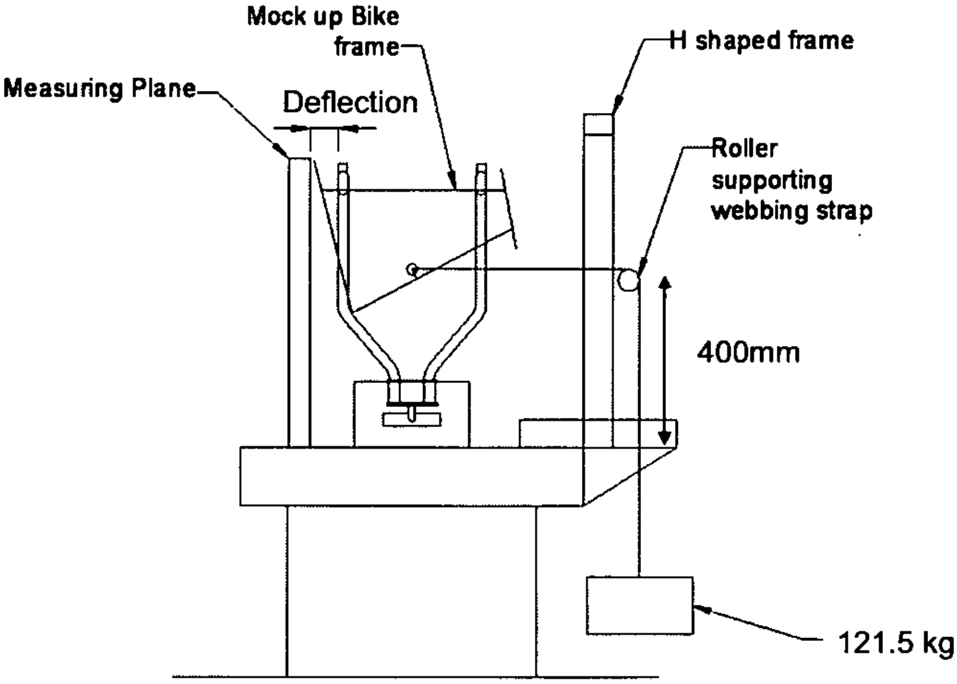


Table 4.5. Lateral Left Deflection

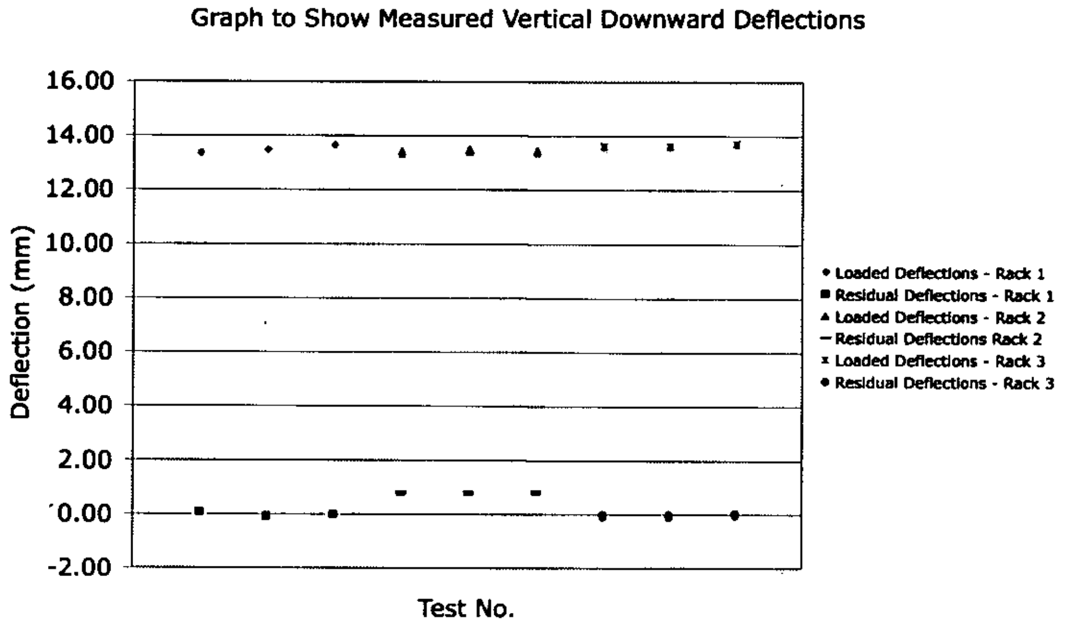
Test	Loaded Deflection (mm)	Residual Deflection (mm)
Rack 1	44.96	0.00
Rack 2	40.84	-0.85
Rack 3	47.60	0.13
Average	44.47	-0.24

Table 4.6. Lateral Right Deflection

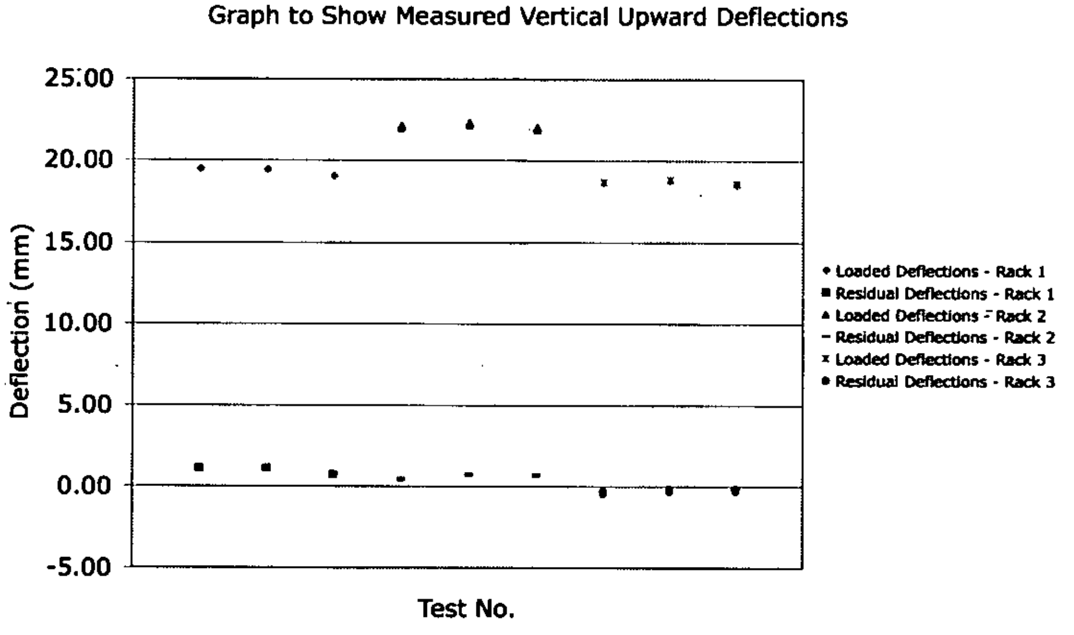
Test	Loaded Deflection (mm)	Residual Deflection (mm)
Rack 1	44.05	0.45
Rack 2	45.63	-0.23
Rack 3	37.75	0.22
Average	42.48	0.15

The following graphs show the results of the deflection tests, they illustrate the loaded and residual deflections for 3 repeats with each of 3 racks.

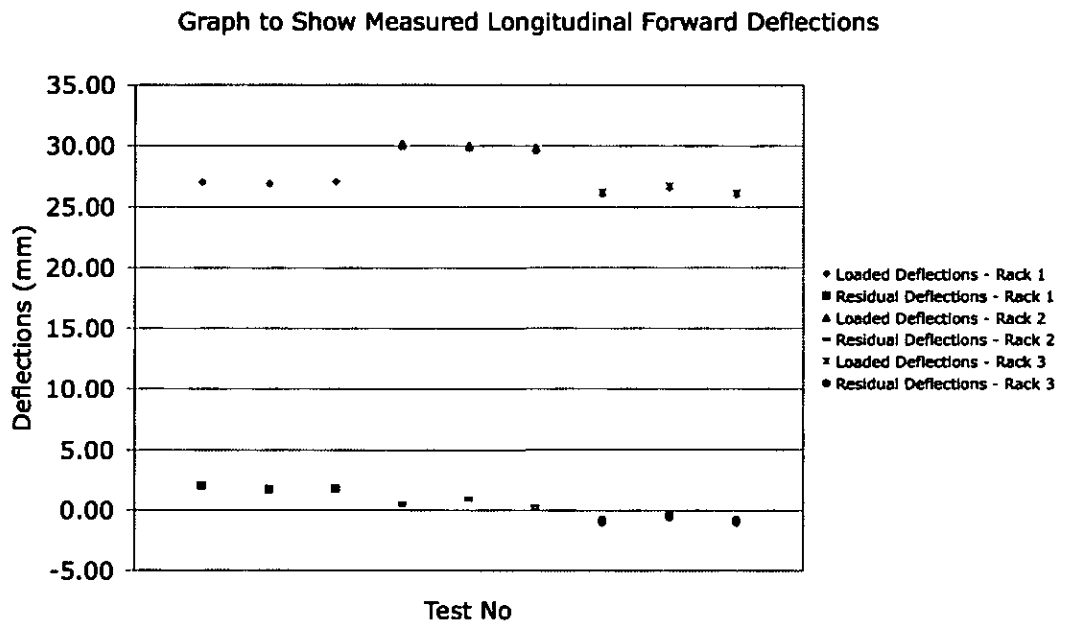
Graph 4.1.



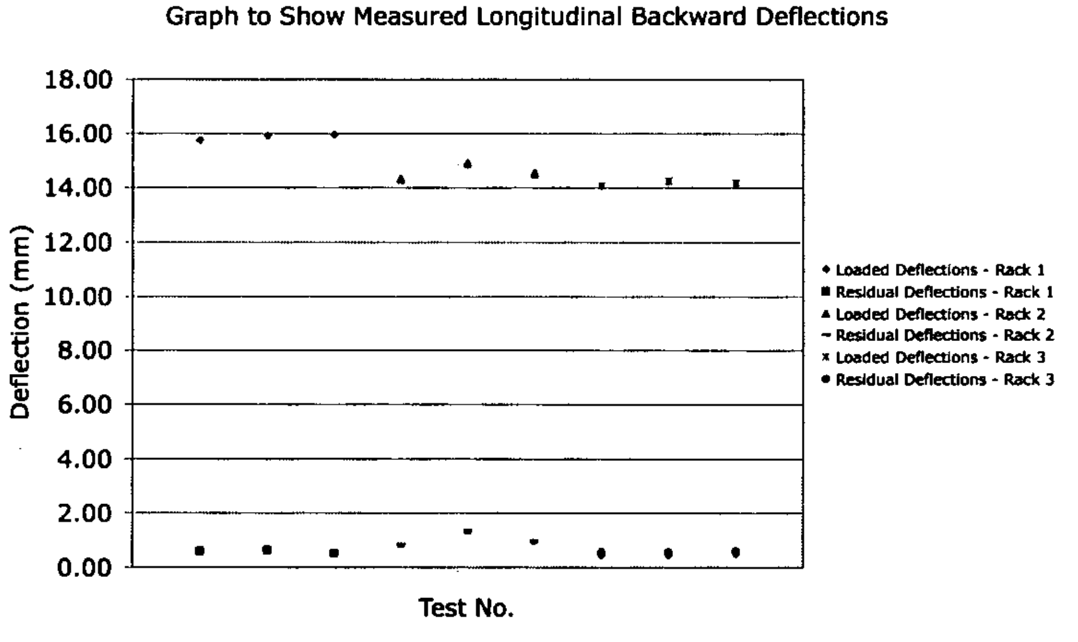
Graph 4.2.



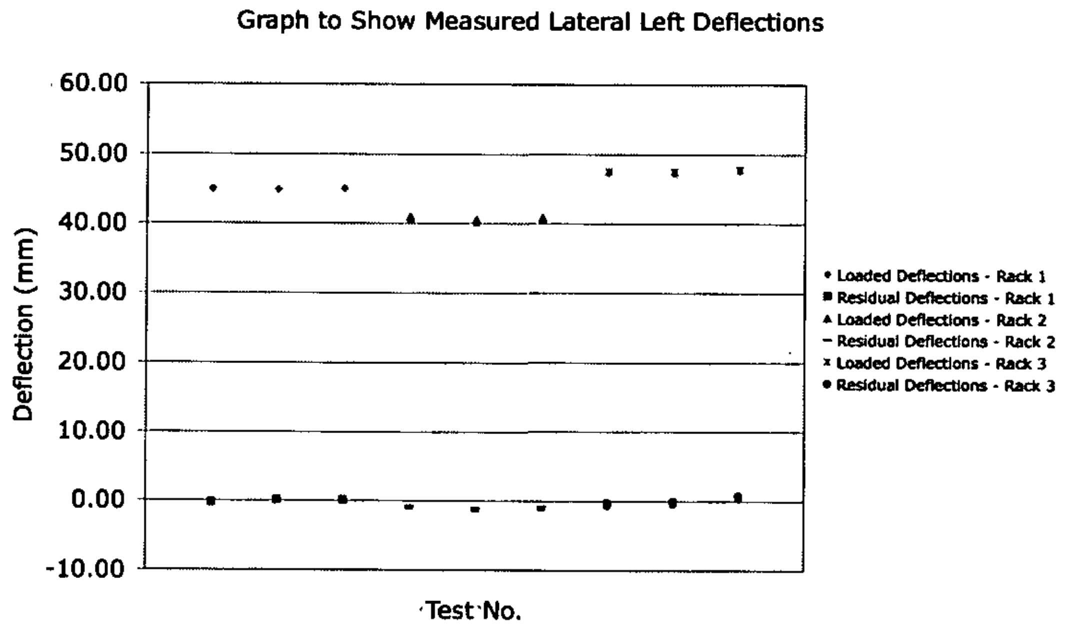
Graph 4.3.



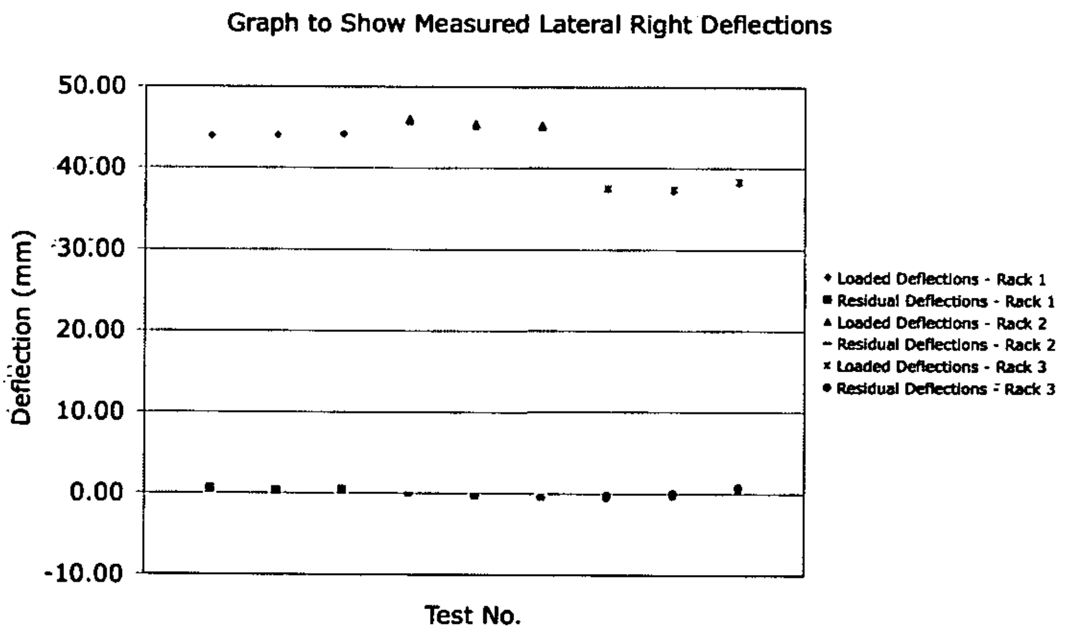
Graph 4.4



Graph 4.5



Graph 4.6



4.2 Strain Gauge Measurement Results

Table 4.7. Strain Gauge Test Results for Vertical Downward Load

Mass (kg)	Average Amplifier Reading (Volts)	Strain $x10^{-6}$	Stress (MPa) $\sigma=E\varepsilon$	Estimated Stress MPa (Bending Moments) $\sigma=My/I$
0	0.000	0.0	0.00	0.0
10	0.012	32.7	6.53	7.2
20	0.025	67.1	13.42	14.4
30	0.037	101.6	20.32	21.6
40	0.050	137.0	27.39	28.8
50	0.063	172.3	34.47	36.0
60	0.076	205.9	41.18	43.2
70	0.089	241.3	48.25	50.4
80	0.101	273.9	54.78	57.7
90	0.113	307.5	61.50	64.9
100	0.126	342.0	68.39	72.1
110	0.138	375.5	75.10	79.3
120	0.151	411.8	82.36	86.5
130	0.164	447.2	89.43	93.7
140	0.177	481.6	96.33	100.9
150	0.190	516.1	103.22	108.1
160	0.203	551.5	110.29	115.3
170	0.216	587.8	117.55	122.5
180	0.229	623.1	124.63	129.7
190	0.242	659.4	131.88	136.9
200	0.256	695.7	139.14	144.1

Table 4.8. Strain Gauge Measurement Test Results for Vertical Upward Load

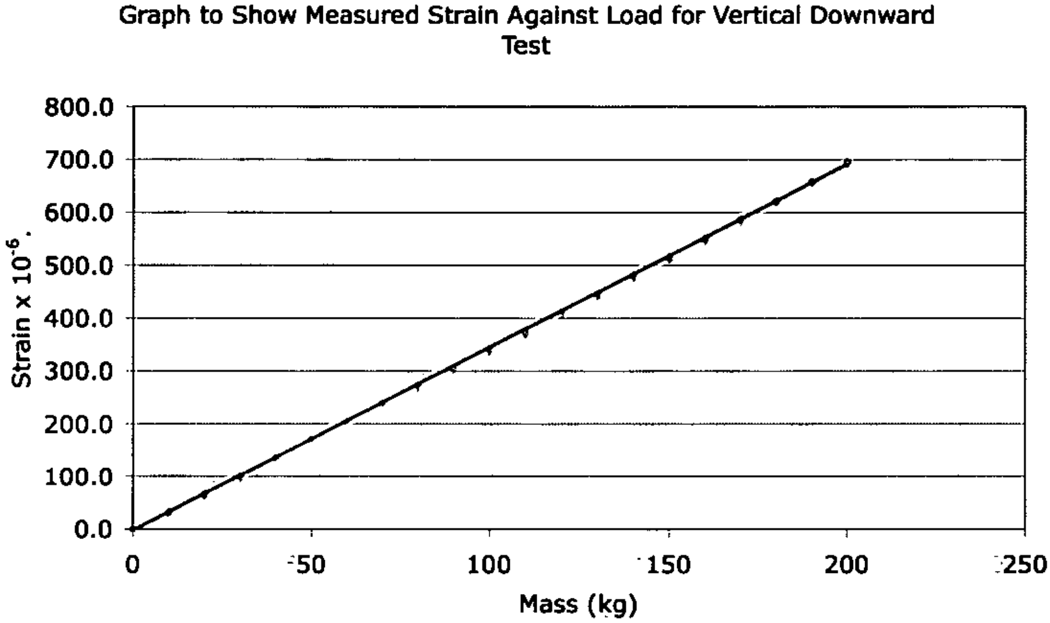
Mass (kg)	Average Amplifier Reading (Volts)	Strain $x10^{-6}$	Stress (MPa) $\sigma=E\varepsilon$	Estimated Stress MPa (Bending Moments) $\sigma=My/I$
0	0.000	0.0	0.000	0.00
10	0.017	36.7	9.070	7.35
20	0.031	76.2	16.871	14.71
30	0.046	115.6	25.034	22.06
40	0.059	149.7	31.927	29.41
50	0.071	182.3	38.639	36.77
60	0.086	221.8	46.621	44.12
70	0.097	249.0	52.789	51.47
80	0.108	276.2	58.776	58.83
90	0.119	308.8	64.943	66.18
100	0.131	341.5	71.293	73.54
110	0.144	374.1	78.186	80.89
120	0.156	406.8	84.717	88.24
130	0.166	435.4	90.522	95.60
140	0.178	470.7	96.871	102.95
150	0.188	503.4	102.132	110.30
160	0.199	527.9	108.118	117.66
170	0.207	553.7	112.653	125.01
180	0.216	579.6	117.551	132.36
190	0.225	604.1	122.630	139.72
200	0.234	627.2	127.166	147.07

Table 4.9. Strain Gauge Measurement Results for Longitudinal Loads

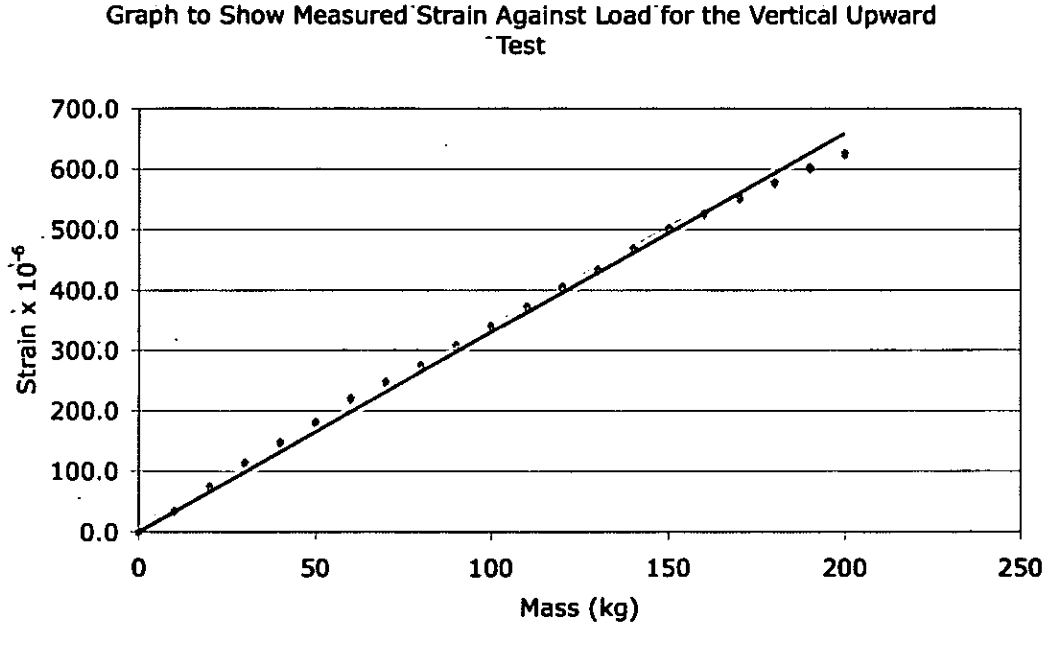
Mass (kg)	Average Amplifier Reading (Volts)	Strain $x10^{-6}$	Stress (MPa) σ=Eε	Estimated Stress MPa (Bending Moments) σ=My/I
0	0.000	0.0	0.00	0.000
10	0.011	49.0	9.80	12.869
20	0.026	98.0	19.59	25.737
30	0.026	144.2	28.84	38.606
40	0.053	193.2	38.64	51.475
50	0.078	247.6	49.52	64.344
60	0.112	304.8	60.95	77.212
70	0.124	367.3	73.47	90.081
80	0.152	429.9	85.99	102.950
90	0.153	495.2	99.05	115.819
100	0.189	571.4	114.29	128.687
110	0.210	620.4	124.08	141.556
120	0.228	666.7	133.33	154.425
130	0.245	745.6	149.12	167.293
140	0.274	813.6	162.72	180.162
150	0.298	859.9	171.97	193.031
160	0.330	919.7	183.95	205.900
170	0.358	985.0	197.01	218.768
180	0.362	1077.6	215.51	231.637
190	0.402	1134.7	226.94	244.506
200	0.402	1194.6	238.91	257.375

In the following graphs 4.7-4.9 of strain against load a linear relationship shows that the material is behaving elastically, therefore the stress is below the yield point.

Graph 4.7.



Graph 4.8.



Graph 4.9

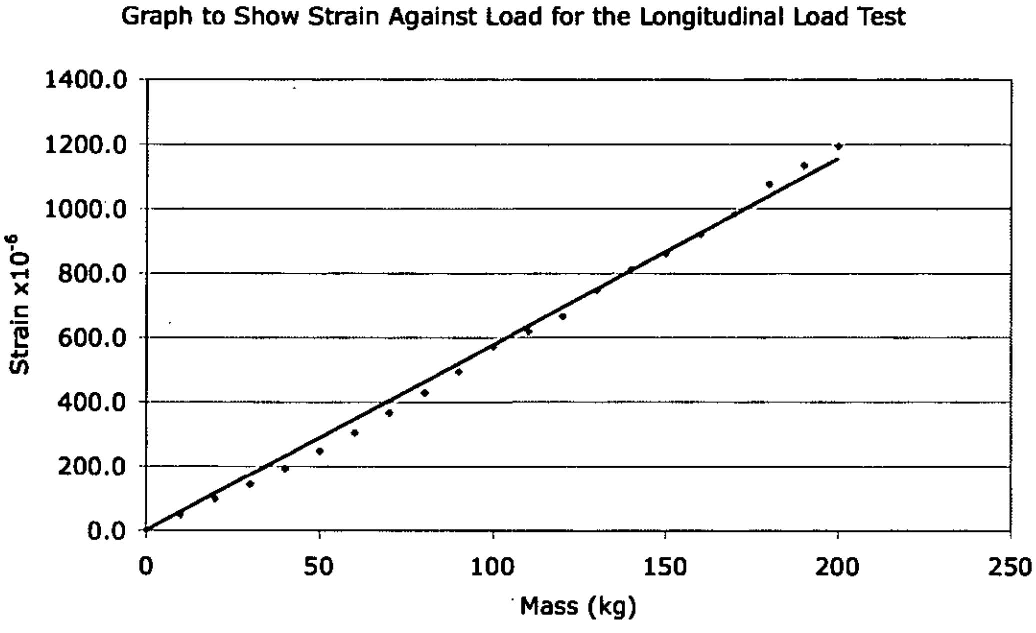


Table 4.10. FEA Results for Vertical Downward Test.

Mass (kg)	Maximum Stress (MPa)	Loaded Deflection (mm)
50	37.667	4.564
100	75.334	9.127
150	113.000	13.691
200	150.670	18.255

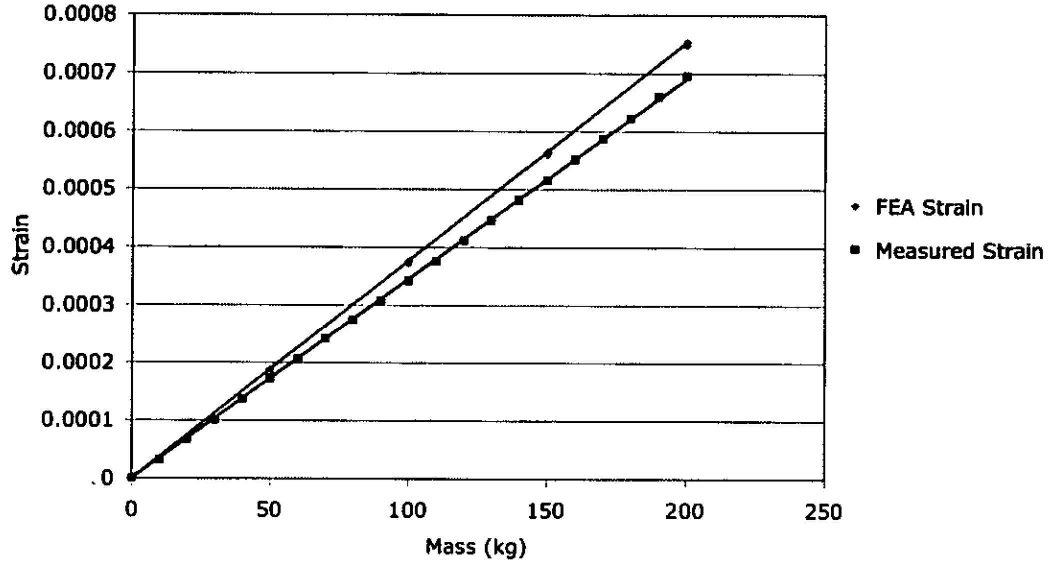
Table 4.11. FEA Results for Vertical Upward Test.

Mass (kg)	Maximum Stress (MPa)	Loaded Deflection (mm)
50	37.667	4.564
100	75.334	9.127
150	113.000	13.691
200	150.670	18.255

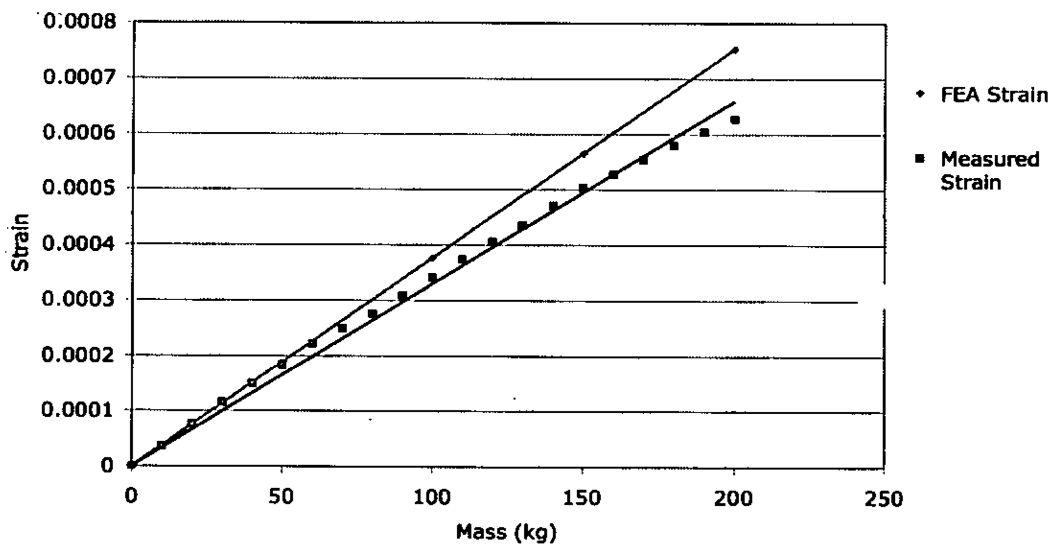
Table 4.12. FEA Results for Longitudinal Test

Mass (kg)	Maximum Stress (MPa)	Loaded Deflection (mm)
50	63.489	3.191
100	126.980	6.383
150	190.470	9.574
200	253.960	12.765

Graph 4.10. Comparison between Measured Stress from Strain Gauge tests and FEA predicted Maximum Stress for the Vertical Downward Test. The lines are linear and diverging. This shows a consistent systematic error between the two measurements. Similar trends are observed in graphs 4.11 and 4.12.



Graph 4.11. Comparison between Measured Stress from Strain Gauge tests and FEA predicted Maximum Stress for the Vertical Upward Test.



Graph 4.12

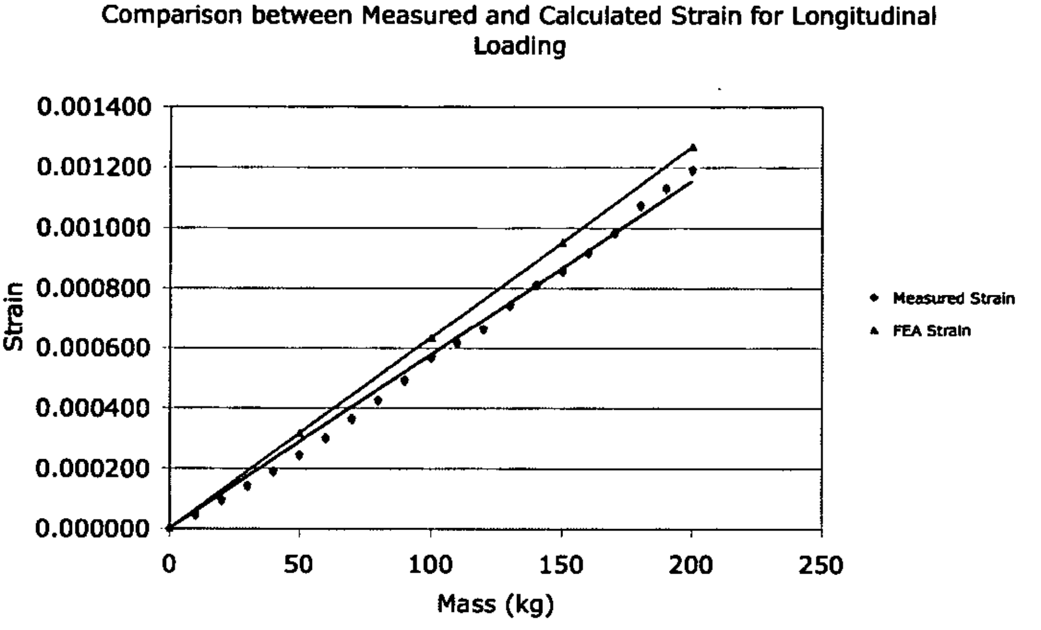


Table 4.13. Checking the Vernier Calliper against other Measuring Devices.

Two Vernier Callipers and 1 DTI used to measure the same object.

	Result 1 (mm)	Result 2 (mm)	Result 3 (mm)
Vernier Calliper	7.99	8.02	8.00
2nd Vernier Calliper	8.02	8.01	7.99
DTI	8.03	8.02	8.01

First Vernier calliper was used to measure in all tests on the rack.

Table 4.14. Average Values for Measurement Tests

Average For Vernier Calliper	8.00mm
Average of all Measuring Devices	8.01mm
Difference Between Averages	0.01mm
Spread of Vernier Calliper Measurements	0.03mm

Table 4.15. FEA Results using different Mesh Densities. Tests Based on Vertical Downward Forces.

Table 4.15a. Standard Mesh Results – Relevance = 0.

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	37.667	4.564
100	75.334	9.127
150	113.000	13.691
200	150.670	18.255

Table 4.15b. Fine Mesh Results – Relevance = 100.

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	39.400	4.404
100	78.801	8.808
150	118.200	13.212
200	157.600	17.616

Table 4.15c. Coarse Mesh Results – Relevance = -100.

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	37.637	4.366
100	75.274	8.731
150	112.910	13.097
200	150.550	17.462

Table 4.16 FEA Results using 2mm wall 31.8mm tube instead of 2.7mm wall tube

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	50.475	5.890
100	100.950	11.781
150	151.420	17.672
200	201.900	23.562

Table 4.17 FEA Results using 1.5mm wall 31.8mm tube instead of 2.7mm wall tube

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	75.929	7.761
100	151.860	15.522
150	227.790	23.283
200	303.710	31.000

Table 4.18 FEA Results using 1.0mm wall 31.8mm tube instead of 2.7mm wall tube

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	147.580	11.982
100	295.160	23.964
150	442.750	35.945
200	590.330	47.927

Table 4.19 FEA Results using 3.5mm wall 31.8mm tube instead of 2.7mm wall tube

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	31.427	3.811
100	62.853	7.621
150	94.280	11.432
200	125.710	15.242

Table 4.20 FEA Results using increased 40mm diameter tube with 2.7mm walls.

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	31.000	2.243
100	42.271	4.485
150	63.406	6.728
200	84.542	8.971

Table 4.21 FEA Results using increased 40mm diameter tube with 2mm walls.

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	27.576	3.017
100	55.152	6.033
150	82.728	9.050
200	110.300	12.067

Table 4.22 FEA Results for a small increase in the bending moment created by situating the load further away to simulate a change in position of the centre of gravity of the bicycles. Increase = 75mm.

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	46.789	6.084
100	93.577	12.169
150	140.370	18.253
200	187.150	24.337

Table 4.23 FEA Results for a small decrease in the bending moment created by situating the load closer to the start of the arm to simulate a change in position of the centre of gravity of the bicycles. Decrease = 75mm.

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	27.943	3.136
100	55.885	6.271
150	83.828	9.407
200	111.770	12.543

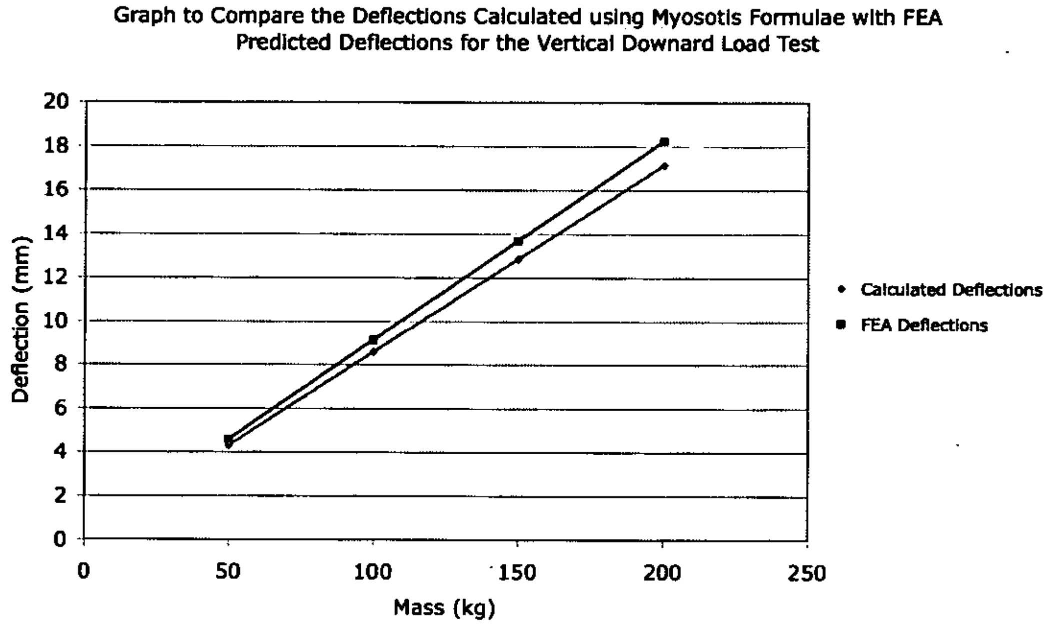
Table 4.24 FEA Results for a change in bend radius to 50mm from 105mm, this moves the centre of the load back towards the car by 27.5mm, reducing the bending moment.

Mass (kg)	Stress (MPa)	Loaded Deflection (mm)
50	30.228	4.014
100	69.455	8.027
150	100.680	12.040
200	138.910	16.053

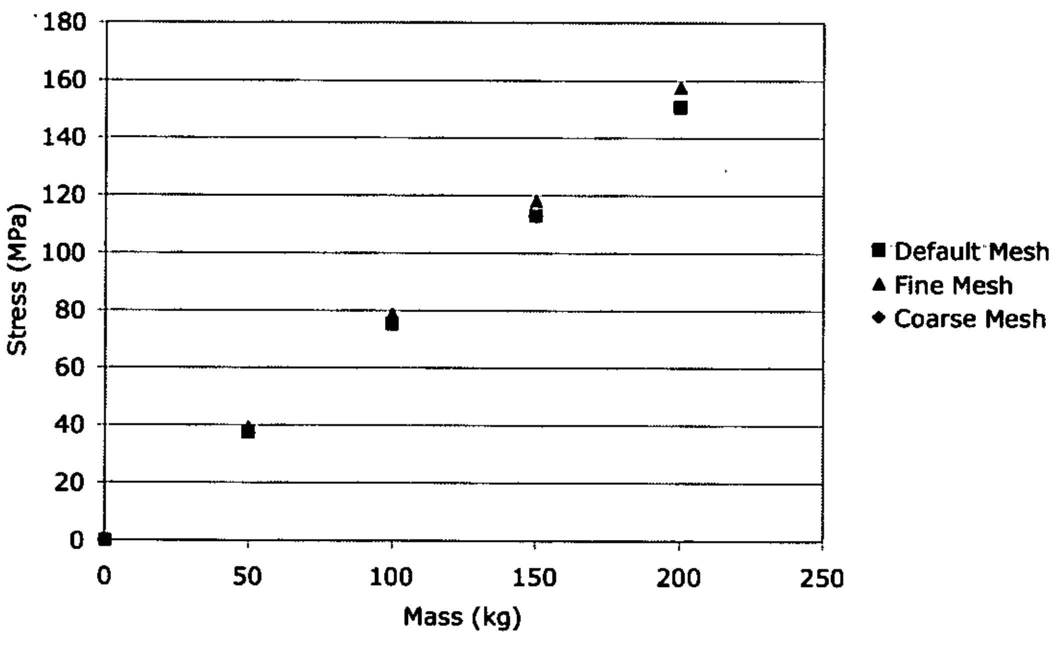
Table 4.25 Compares the deflections predicted by the FEA (default mesh density) with those calculated by using Myosotis Formulae for the vertical downward load test.

Mass (kg)	Calculated (mm)	FEA Deflection (mm)
50	4.298	4.564
100	8.586	9.127
150	12.864	13.691
200	17.152	18.255

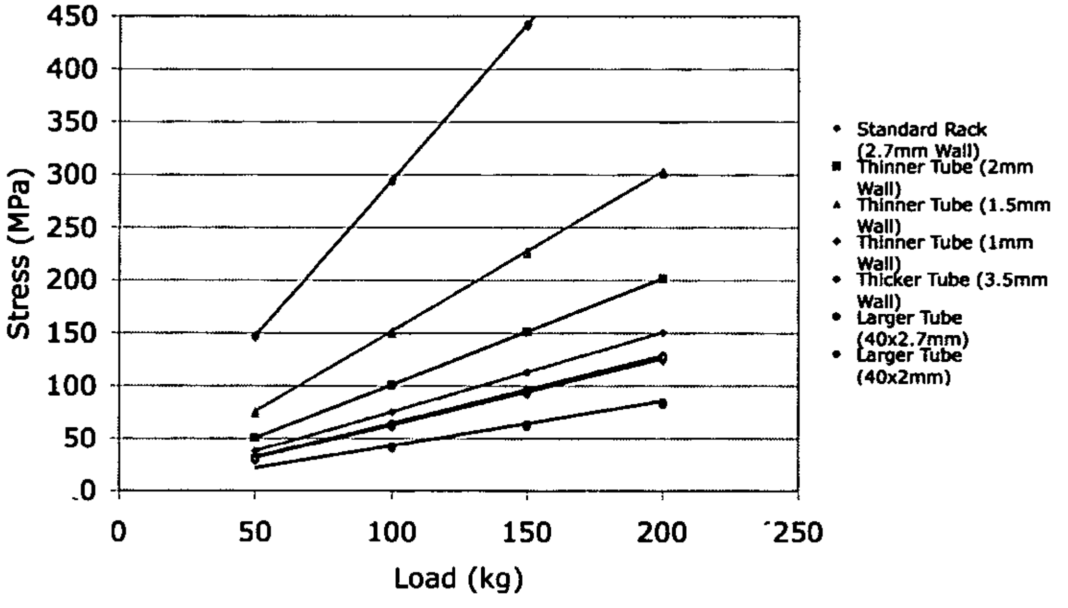
Graph 4.13 Both lines are linear and diverge indicating a consistent systematic error between the two measurements. This error may be used to predict the accuracy of the FEA results for future applications.



Graph 4.14 Shows the difference between the Default (medium) Coarse and Fine Mesh settings for FEA analysis. The results show close correlation between the default and coarse mesh with a larger gap to the fine mesh.



Graph 4.15 Compares the effects of changing the wall thickness and tube diameter. These changes were intended to reduce stress and weight in the bicycle rack. Changes were made by changing parameters in the CAD model of the bicycle rack.



Graph 4.16 Compares the effects of changing the bending moment to simulate changes to the Centre of Gravity position. This was achieved by changing the design of the rack, using a tighter bend to move the load, and by simulating different positions of load. This may be due to user preferences or the shape of the bicycles carried.

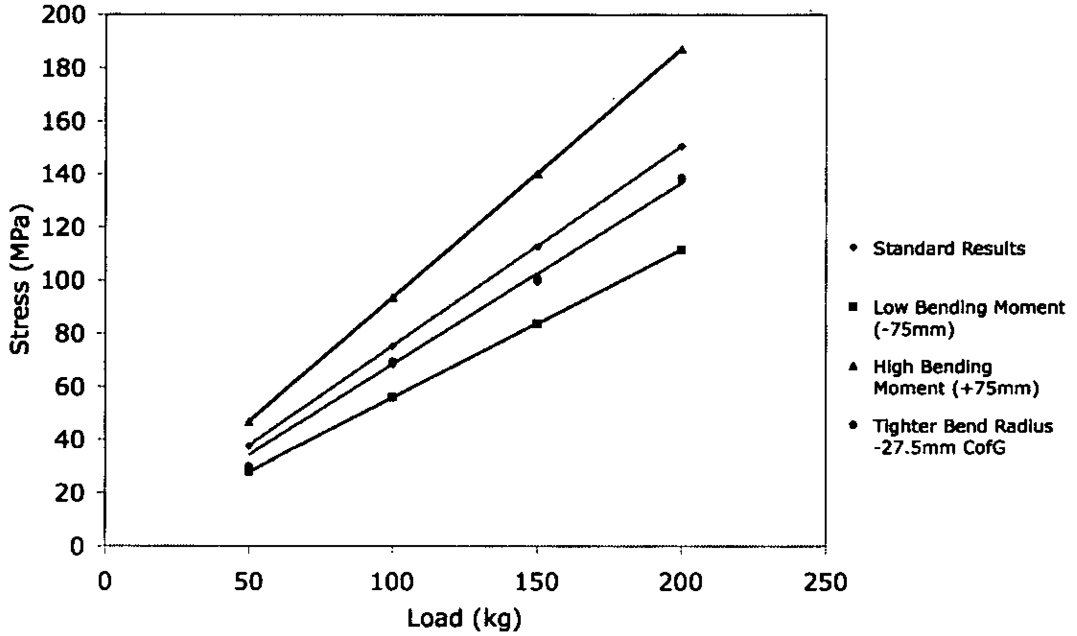


Table 4.26. Measurements of the Tested Rack Arms.

Tube diameter is the diameter of the bicycle rack arm tube measured at two places to check for roundness.

		Tube Diameter	Lug Width	Bottom Hole Diameter
Rack 1	Right Arm	31.75-31.95	12.00	5.31
	Left Arm	31.9-32.05	12.06	5.20
Rack 2	Right Arm	31.8-31.9	12.03	5.21
	Left Arm	31.75-31.95	12.08	5.42
Rack 3	Right Arm	31.8-32.0	12.01	5.41
	Left Arm	31.83-31.95	12.06	5.31

All dimensions in mm.

Chapter 5

Discussion

5. Discussion

5.1 Interpreting the Deflection Tests

The residual deflections recorded are all within the 20mm allowed by the standard. The maximum residual deflection was 1.8mm recorded on the longitudinal test as shown in results table 4.3. This is only 9% of the allowable deflection. The Pendle TBM 3 rack easily satisfies the requirements of the deflection test part of the draft ISO 15263.4 standard.

For residual deflection to occur due to the arms being deformed then the stress in the arms must exceed the yield point of the material. The yield strength for the material is 300 MPa. The maximum stress recorded during the longitudinal test was 197 MPa with a 169 kg load and is 56.3% of the yield stress of the material.

The residual deflection is not due to yielding of the material, it was observed that the rack arms move within the mounting block sockets. Figure 5.1 illustrates the mounting of the rack arms in the mounting block sockets. All deflections count towards the residual deflections that the draft ISO 15263.4 standard is concerned with, whether they are due to yielding or not.

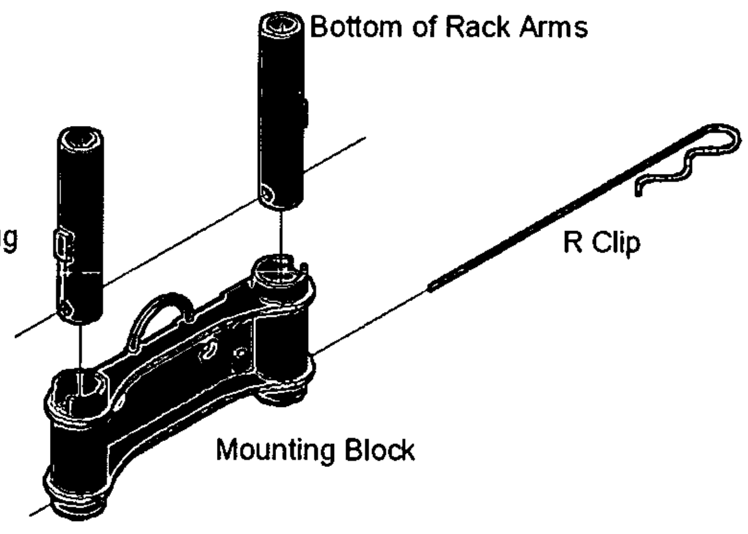


Fig 5.1. Mounting block assembly. Arm tubes are inserted into the mounting block; the lugs locate into slots on the sockets. The R Clip is inserted through the entire assembly.

Table 4.19 records measurements that were taken after testing. The measurements show that there is 0.5mm clearance between the holes and the R clip. There is 1.5mm clearance between the lugs and the mounting block. The clearances are required for ease of assembly, they also reduce damage to the painted surfaces from regular assembly.

The movement between the tube and the mounting block is responsible for less than 10% of the 20mm that is allowed in the draft ISO 15263.4 standard. The maximum residual displacement is low compared to the loaded displacement for this test, which was 30.06mm. The residual deflection is 6% of the maximum loaded deflection, making it small enough to be considered acceptable for safe performance of the rack.

Using Myosotis formulae (page 27) the downward vertical deflection was predicted at 12.5mm. The value of deflection under load measured in the tests was 13.55mm. The movement of the rack relative to the mounting block may explain the 1.05mm difference between these values. Some of this difference is due to movement in the mounting block and some will be due to flex in the apparatus as well as any measurement errors. The calculations do not account for any flex in the supports.

Graphs 4.1-4.6 illustrate the results of the deflection tests showing the loaded and residual deflections. The graphs show results for all three measurements taken with each of the three racks. The graphs show agreement between measurements for the three measurements on a particular rack, but larger differences between the different racks. The small difference for measurements on the same rack may be attributable to random errors in the measurement. The larger difference between measurements of different racks are more likely systematic due to differences in the test apparatus setup or manufacturing differences in the racks.

Taking an average reduced the small random errors of up to 0.46mm between measurements on the same rack. The most practical measuring method was to use a digital Vernier calliper. It is difficult to consistently place the calliper in line with the direction of displacement so multiple measurements were taken so that the mean average value may be obtained.

The maximum difference was 0.46mm in the measurement of the second rack on the Longitudinal Forward deflection test. This value is 1.53% of the average reading of 30.06mm and is within three standard deviations of the mean. Most measurements were within 0.15mm of each other.

Between tests none of the equipment was changed except the test rack. The larger errors between different racks are systematic errors due to the variations between the three racks. The application of the load will vary slightly between racks. The point of application was measured using a steel tape with a resolution of 0.5mm. The error would be of the order of 0.5 mm in 200mm or 0.25%. The residual deflections are due to the movement of the rack arms in the mounting block. The differences between the racks could be due to the differences in the size and shape of the tube or the position of the lug that locates it in the mounting block or a difference in the material.

Table 4.19 shows the difference between the dimensions of the three racks. The measurement of the diameter of the tube 90 degrees apart show that there is a 0.2mm difference in the diameters because the tube is slightly oval. There is a difference of 0.25mm between the different racks. The orientation of the oval will be different for each rack when the tube is bent, this will change the size of the clearance between the tube and the socket. Table 4.19 also shows differences of 0.08mm in the lug width and 0.21mm difference in the diameter of the bottom hole for the R clip. The clearance on the R clip will allow differing amounts of movement as the arm rotates slightly in the socket.

5.2 Strain Gauge Results

The strain gauge indicates the strain from which the stress may be calculated. Bentley (1995) and Measurements Group (1992) describe installing gauges with particular emphasis on accurate alignment and adhesion.

The gauges were installed on the arm at a point just above the socket of the mounting block. Bending moment calculations showed this to be the maximum stress point. The gauges were installed on the front and back of the arms to measure stress in the vertical and longitudinal tests. The longitudinal test showed the greatest deflection and highest predicted stress. The results confirm that the highest stress was recorded on the longitudinal test. Table 4.9 shows that stress was measured at 238 MPa for a 200kg load. The stress is significantly below the 300MPa yield point of the material so the material will not yield.

Graphs 4.7 – 4.9 plot the strain against load for the vertical and longitudinal tests. The points plotted fall close to the straight line with some small random errors. This may be due to errors in the measurement of the applied loads, temperature effects or float in the output reading.

Dally and Riley (1991) discuss the effects of hysteresis which is the difference in the energy to load and unload a test piece. This may lead to a systematic error and a float in the zero position. There was a small error in the return to zero on the tests, but hysteresis would normally lead to a small systematic error across a wide range of results.

5.3 Finite Element Analysis

It is well known, see for instance Berk (1988) and Burnett (1988), that FEA can only be used as guidance and not considered absolutely accurate. This is due to the complexities of a solution, the ability of the software and the assumptions and approximations of boundary conditions that must be made. These statements were made 19 years ago; the same is said in many more modern studies of FEA such as McMahon and Browne (1998). As computer processors become more powerful and affordable then it is easier to gain access to better solutions. Rooney and Steadman (1993) stated that as power increases then the range of available solutions and the accuracy will increase, because processing time will come down making more complex solutions or meshes with smaller elements possible on the average PC. Conversely simple solutions may become more accurate because they may be split into smaller elements.

Another requirement for good quality FEA results is creating a boundary condition that will accurately reflect the physical situation. Even then as stated by Berk (1988) the solution is only predictable in theory. There may be certain situations that are beyond the foresight of the operator that may lead to the software producing an incorrect solution.

When producing the FEA results for the bicycle rack, it was observed that changes to the boundary conditions and the mesh density made significant differences to the results. The results showed differences such as recording a higher deflection than the physical tests. This may be due to only modelling the rack and mounting block. Creating a model of the towball and test bench may reduce these errors, although the point of restraint will always be infinitely stiff, so it should be moved to a point where the real world deflections become so small as to be insignificant, such as the test bench or the floor.

The same difference is seen when comparing the FEA deflections to predicted deflections calculated using Myosotis formulae, as shown in Table 4.25 and Graph 4.13. The deflections from the FEA are consistently 6.4% higher than those that were calculated by the Myosotis formulae. The difference between Calculated and FEA predicted stresses was slightly lower at 4.6%. Whilst the FEA assumes stiff restraints the model does include the mounting block. The software models some deflection in the mounting block, which will contribute to maximum deflections at the ends of the rack arms.

To create the exact boundary conditions seen here an FEA package with the ability to utilise node separation as used by Yahiaoui et al (2001) to model cracks would be required. This would allow accurate modelling of the interface between the rack arms and the mounting block as well as the interface between the block and the towball. In this FEA the main shortcoming was the inability to model the fit between the mounting block and the arms that causes the small residual deflections.

Changing the boundary conditions made the most significant differences. The first tests used a model of one arm that had a surface split where the socket would hold it. This was used as the constraint position to simulate the block. A load was applied at the centre of the arm at the same point as the load was applied during the physical tests. This produced results that differed substantially from those obtained using calculated estimates and the strain gauges.

The model, illustrated by Figure 5.2 has two arms linked by a solid object to simulate the mounting block. The load was applied to the point created on top of the beam at the same position as the mass used in the deflection tests. This was a more representative model of the situation and the results were much closer to those obtained through physical tests.

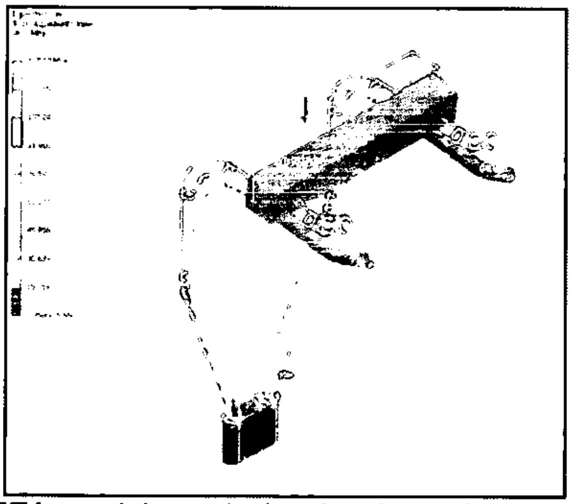


Figure 5.2 Final FEA model used showing here the vertical load test. Red indicates high stress, blue indicates low stress.

The calculated stress results correlate with the strain gauge measurements, a minimum difference of 2.5% (see table 4.10-4.12) was achieved from the measured stress obtained on the vertical downward test. The software cannot predict residual deflection, but it can predict the deflection under load. The deflection results are subject to a greater error than those for stress because the calculations must take into account material properties. Since stress is equal to force divided by area, it will be the same for a particular shape regardless of the material. The deflection will depend on the stiffness of the material. The FEA must use accurate material properties that are subject to manufacturing tolerances in the metal. It will also require further steps in the calculation process where errors may be compounded. Ray (1997) found when comparing FEA results to experimental data in crash testing that material properties must be made equal in the software to the actual properties of the material used in production, taking into account manufacturing tolerances. Changes in material properties may occur due to working and heat effects. For a truly accurate analysis these may have to be accounted for in the software.

Graphs 4.10-4.12 show a comparison between the strain results from Strain Gauges and FEA. The graphs show both straight lines diverge at the same rate indicating that there is a consistent error between them. If the error is consistent between FEA and the Strain Gauge results then it is systematic and can be corrected or a tolerance allowed for this inaccuracy during processing.

It is known that in FEA an increase in mesh density will lead to an increase in accuracy. Burnett (1988), Rooney and Steadman (1993) and McMahon and Browne (1998) discuss these characteristics.

In this case, the vertical downward test was used to compare the effects of mesh density. The same test was carried out with Medium (default) density, the coarsest density allowed and the finest. Tables 4.15 a, b and c show the results of changing the mesh density. The measured stress from the strain gauge tests produced a result of 139 MPa. The results calculated using simple bending theory produced a value of 144 MPa.

The results in tables 4.15 a,b and c show that the accuracy compared to the strain gauge result decreased with a finer mesh. The most accurate results were with the coarsest mesh, the opposite of the expected result. Berghini and Bettini (2001) utilised a variable mesh to obtain an accuracy of 0.1%. The variable mesh consists of smaller elements at predicted higher stress points. They were also using a derivative of ANSYS but not the version built into Inventor as used in this research. Their software was the full stand alone ANSYS application.

The strain gauge is taking an average over it's surface area, the stress could be concentrated over an area that is smaller than that measured by the strain gauge. This area may be similar in size to the elements in the coarse mesh causing the result for the coarser mesh to be closer to the value measured by the strain gauge. Alternatively the gauge may not be at the correct location. The FEA showed the maximum slightly higher than the point measured. Further testing of other racks with gauges in different positions could prove this.

Other factors in the geometry of the model may affect the result obtained. As Burnett (1988) stated FEA gives an approximate value based on the boundary conditions given. It cannot cover any unpredicted occurrences that will manifest in physical test regardless of user input. Berk (1988) says that the software is predictable if all boundary conditions are known.

The FEA software was tested in one situation here. A wider range of problems should be used to evaluate the software more comprehensively. Results should be generated for a wide variety of problems, using various mesh densities so that conclusions on the accuracy and reliability of the results may be drawn. A correction factor may be developed to improve accuracy as demonstrated by Cristofolini and Viceconti (2000). Eventually FEA work can be used with an acceptable level of accuracy and repeatability to make a meaningful input into the design process. The experiences here of changing boundary conditions to give more realistic results are one example of the learning process required. The initial tests demonstrated that improving boundary conditions and the model produces more realistic results.

5.4 Predicting Failure

The purpose of the draft ISO 15263.4 standard is to prevent failure in use. In terms of the requirements of ISO 15263.4 the Pendle TBM 3 rack would pass the deflection test part of the standard and be deemed safe. The residual deflections are within the allowed limits.

Lewis and Samuel (1989) discuss failure and determine that an acceptable life for a component should be defined by considering the likely use of the component and the mechanics and materials involved. Dym and Little (2004) categorise failures due to wear or service time. According to Lewis and Samuel (1989) in service failures are attributable to any conditions including fatigue.

Fatigue failure could occur to a bicycle rack because the loads placed upon it are cyclic due to road surfaces and movement of the car. These forces fit the descriptions given by Lewis and Samuel (1989) of cyclic forces that will lead to fatigue failures.

In order to predict statistical fatigue failure the endurance limit or fatigue strength of the steel must be known. Suresh (1991) states that for steel the fatigue strength is between 0.35 and 0.5 times the UTS of the material although Shigley (2004) presents experimental data showing it to lie between 0.35 and 0.6 times the UTS. Most work on fatigue concludes that the fatigue life should be determined for the particular material involved by experimental means rather than estimation. Suresh (1991) shows that surface finishes and treatments can make a significant difference to similar materials.

Osgood (1970) chronicles the development of fatigue prediction, Fig 5.3 illustrates the three main relationships between endurance limit and the mean stress. An early method was the Gerber relationship, a parabolic line drawn on a graph of endurance limit against the Tensile strength. The relationship was developed from experimental results. Suresh (1991) considers this method generally accurate for most ductile alloys, but less accurate for brittle ones. Goodman proposed a relationship based on experimental results for brittle alloys that was more conservative than Gerber's. The Goodman relationship is a straight line under the Gerber parabola. Suresh (1991) considers Gerber conservative for ductile materials, but accurate for brittle ones. Soderberg suggested another relationship, drawing a straight line from the endurance limit to the yield strength. Suresh (1991) and Osgood (1970) consider this to be conservative, but more reliable for safety critical parts. The Soderberg relationship therefore would be most suitable for a bicycle rack.

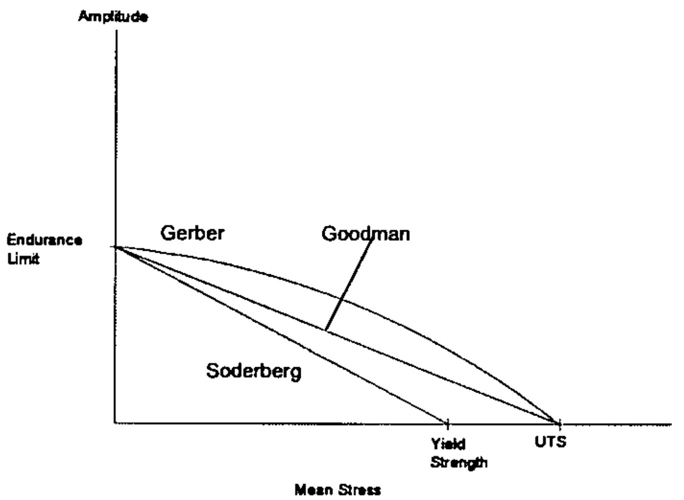


Figure 5.3 An illustration of the Gerber, Goodman and Soderberg relationships.

When using the Soderberg relationship as illustrated by Figure5.3 a point would be plotted with the alternating stress on the y-axis and the mean stress on the x-axis. If the point sits below Soderberg's line then the rack will not fail in fatigue under normal conditions. This method is dependent on knowing the endurance limit of the material as well as the maximum and mean stress values.

It was not possible to determine the fatigue strength during this work due to time restraints, Suresh (1991) recommends completing a rotating beam test on a sample of material. So the most conservative estimate must be used which according to Shigley (2004) and Suresh (1991) would be 0.35 UTS, the UTS is 400MPa so the estimated fatigue strength is 140 MPa.

The mean load is the weight of the bicycles on the rack while the car is standing still which is 45kg equating to a stress of 35 MPa according to Table 4.7. The maximum stress amplitude is likely to be much lower than the 3.7 times the mass of the bicycles recorded in the deflection tests. Higher stresses are caused during braking and would be longitudinal. Braking at 9.8 ms^{-2} would equate to a 100-0 Kph time of 2.8 seconds. This is consistent with most manufacturers Figures and independent magazine road tests. Table 4.9 shows that this would be 49 MPa.

Plotting these points produces the Soderberg analysis as illustrated by Figure 5.4. Indicating that the TBM 3 rack sits well below the line and would not be subject to fatigue failure under these conditions. Conditions may occur outside those expected so a factor of safety must be applied to this result. Reducing the endurance limit and yield strength by a factor of 2 would show a factor of safety of 2 as seen in Figure 5.5.

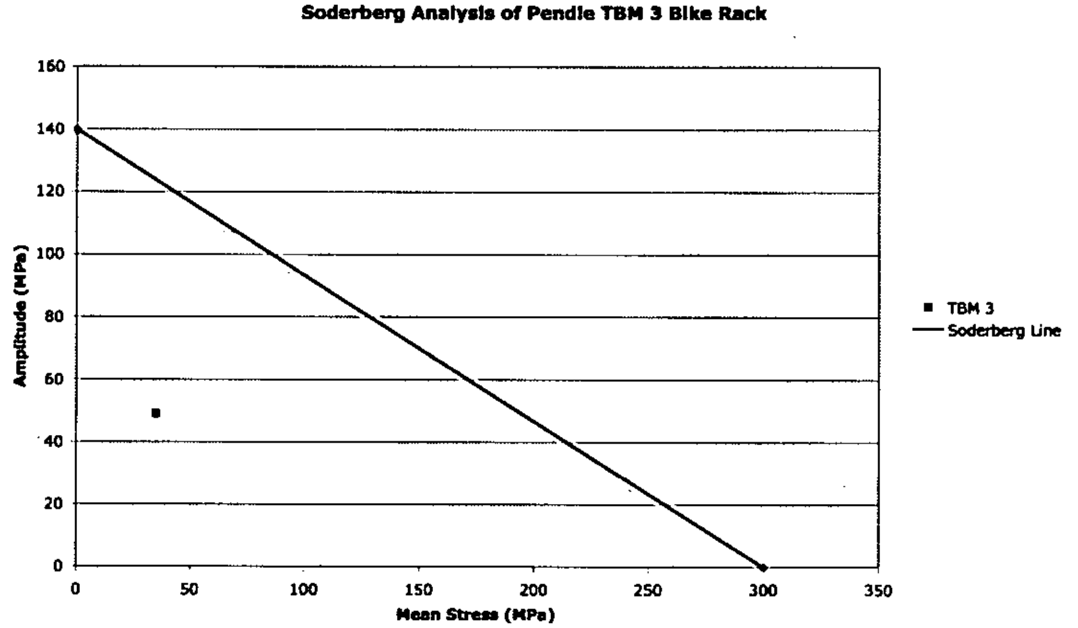


Figure 5.4. Soderberg Analysis of a Pendle TBM 3 rack. The point plotted sits a long way below the line indicating that the rack would not fail in fatigue under these circumstances.

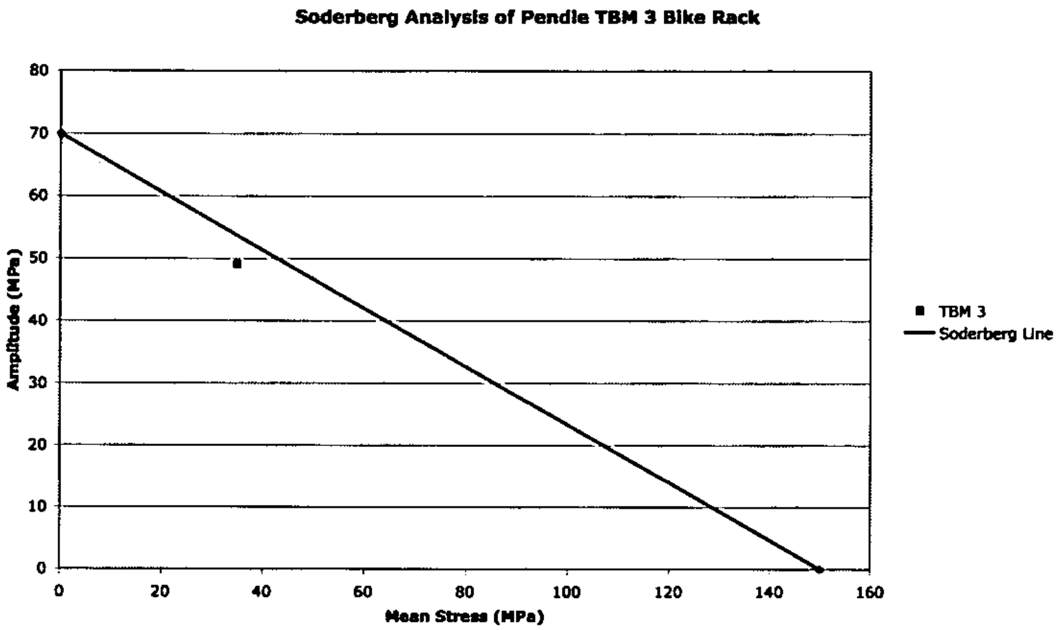


Figure 5.5 Soderberg Analysis of Pendle TBM 3 rack, fatigue and yield strengths halved still shows that the point plotted for the rack would be under the line indicating no failure.

When the factor of safety is doubled the TBM 3 rack is still below the line. Indicating a factor of safety greater than 2 (measured as 2.4). This is only an estimate, it is likely that the fatigue strength of the material is higher than the estimate used here so the factor of safety may be greater than shown.

If the rack is designed to minimize fatigue failure it may still be subject to failure because of corrosion. Corrosion in the material will lead to an increase in stress. The maximum stress in this rack is just after the bend at the bottom of the arms. Hummel (1997) states that there is a higher chance of corrosion on a bend in a tube than the straight sections. The rack arms are powder coated to protect the steel, but they may easily become scratched or worn around the maximum stress point due to repeated assembly.

The draft ISO 15263.4 standard includes a salt spray test. This should be conducted on a used rack so that corrosion risk at the points of wear can be identified. Corrosion is an important consideration on this rack because it will likely be used in wet weather where it may be affected by road spray and cold weather when the roads are salted. Mud and water may also come from bicycles as they are put back on the rack after a ride.

Ideally further analysis may look at the nature of road spray, for example which parts of the rack catch the most spray. Ideally a car would be filmed travelling through various depths of water to determine the shape of the spray and where it is most likely to contact the rack.

5.5 Changing the Design

An FEA was performed to consider the potential for changing the tube wall thickness to 2mm from 2.7mm. This is a typical design situation that FEA may be applied to. Apart from the wall thickness, the rest of the design was unchanged. The benefits of changing wall thickness are a weight saving of 1.1 kg or 22% and a cost saving of 35% on the raw material. A reduction in weight will mean lower loads on the tow bar, which is a source of failure outside. The bicycle rack itself. The results, table 4.16 and graph 4.14 show that there would be an increase in maximum stress of 51.23 MPa or 34%. Kaye and Heller (2001) used a similar technique to achieve a 27% reduction in hoop stress and an un-quantified weight saving in a bulkhead from an F/A 18 fighter jet airframe.

The maximum stress of 201.9 MPa is still significantly below the yield strength of 300 MPa, but there will be a reduction in the safety factor. The FEA has allowed this test to be performed relatively quickly and cheaply so that the decision may be made to manufacture a prototype and test it further. Without the FEA the expense of making and testing a prototype would have been incurred without knowing whether it may have been worthwhile.

Further analyses considered how thin the walls could be whilst retaining adequate strength, using larger thinner tubes to reach a point of similar strength with reduced weight and using tighter bends to reduce bending moments. He results are shown in Tables 4.16-4.24 and Graphs 4.14,4.15.

When the wall is reduced below 1.5mm the maximum stress exceeds the UTS of the material making the design unsafe. At 1.5mm wall thickness the maximum stress just exceeds the yield strength. A prototype could be produced and tested to determine the residual deflection if the yield strength is just exceeded. The weight saving would be a significant 42%.

The results tables 4.20 and 4.21 show that it is possible to use a 40mm tube with 2mm walls to obtain a reduction in stress and deflection with a slightly lower weight. Whilst this may seem an ideal solution, it is practically less ideal. The mounting block would have to be redesigned to fit the larger tube and the tube would have to be custom rolled because this size is not available off the shelf in ERW3 material. Both factors would increase costs significantly making the decision no longer purely an engineering matter.

Table 4.24 shows that reducing the bend radius to 50mm to reduce the bending moment can reduce the stress and deflection without making changes to the material, this is illustrated in fig 5.6. This modification could be implemented easily with minimal changes to cost of manufacture. A prototype should be made and tested to confirm the FEA results with a view to making production changes.

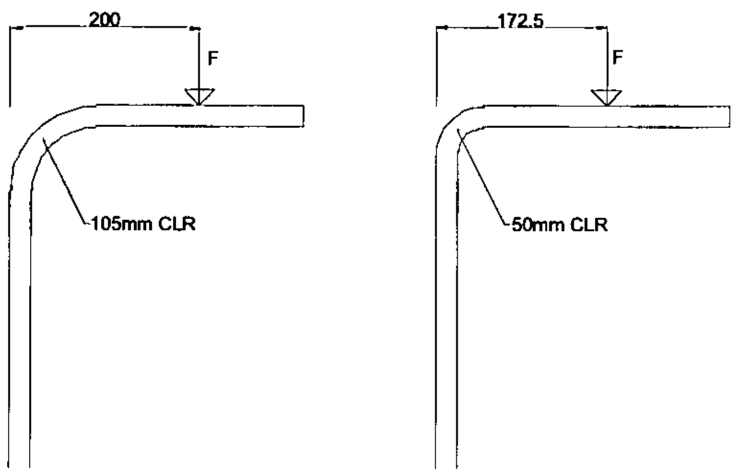


Figure 5.6 Using a tighter bend radius to reduce the bending moment. If the load is distributed evenly along the support arm then the Centre of Gravity will be nearer to the start of the arm.

5.6 Interpretations of the Standard

Further tests were simulated using FEA to consider the effects of misinterpreting the deflection tests or trying to influence the results through interpretation. It was felt the main area that this could occur was in the positioning of the load.

In this research the deflection tests were interpreted so that the load could be placed where the centre of gravity of the bicycles would be, rather than using 3 bicycles, for reasons of practicality. It was also felt that the type of bicycles used might influence the results. For example bicycles with narrow handlebars that fit closely together will create a smaller bending moment than ones that are far apart. Heavy bicycles may be used at the start of the arm with light bicycles used at the ends so they total the required amount, but effectively act with a smaller moment. For that reason the load was applied in the centre of the bicycle support to simulate a distribution evenly over the length of the arm.

The results in tables 4.17 and 4.18 show the effects of moving the load 75mm in either direction of the centre of the support arm. They show a significant difference in the stress. Moving the load further away results in a 36.48 MPa (24%) increase, which may cause a borderline rack to fail. Moving the load closer results in a 38.9 MPa (25%) reduction, which may allow the borderline rack to pass. Clearly the ISO description of deflection tests must minimise the possibility of this happening. In the case of the Pendle TBM3 it would have passed in any situation but other racks may not have such large margins of safety. The differences are illustrated in Graph 4.15.

A potential method of making the rack stronger and lighter is to use butted tubing. This is a method of manufacturing commonly used in bicycle frames. The tube walls are machined to vary in thickness. A thick walled tube is machined away in low stress areas to reduce unnecessary weight. It would be a very costly production method, but large weight savings can be made. For example, the maximum stress point could be 3mm thick whereas the lower stress areas such as the bicycle supports may be only 1.5mm thick. A rack manufactured in this way would require extensive testing to ascertain the effects on the lightened areas, especially with regards to corrosion and fatigue.

5.7 Developing the Draft ISO Standard

This research has shown that fatigue failures must be taken into account when asking whether a rack is safe or not. It could be argued that, due to the initial research such as that by the DTI (2001) which concentrated on Strap On rack failures, that the deflection tests are designed to test Strap On racks. The nature of the straps and hooks making them more susceptible to residual displacements at lower loads. It is, therefore, suggested that the draft ISO 15263.4 standard should be split between tow bar racks and strap on racks. The sections of dynamic tests and requirements for labels and warnings should be the same for both types, but the deflection tests should have different criteria to pass for different types of rack.

For a tow bar mounted rack such as the Pendle TBM 3, the results of the deflection tests show that yielding should not be acceptable. The draft standard should determine that under certain conditions the rack would not be subject to fatigue failure. To determine that no yielding occurs, the deflection under load and the residual deflections, should be measured. A residual deflection within a certain percentage of the loaded deflection must be allowed to account for movement in the assembly of the rack. A maximum deflection should also be stated. The allowed deflections must be reconsidered because there is a large difference between the 1.8 mm residual deflections measured in the deflection tests on the Pendle rack and the 20 mm allowed by the draft ISO15263.4 standard.

Measuring the strain in the rack with strain gauges would make it easy to determine whether the material has yielded or not, but it requires skills and knowledge that limit who can test the rack or provide basis for debate about the validity of a set of results.

The size of the loads applied should be subject to further research, determining the accelerations a rack will be subjected to during typical cornering and braking situations as well as those due to the road surface. The loads should be multiplied by a factor of safety determined from the Soderberg graph so that if no yielding were to occur under these multiplied loads, then fatigue is not likely under normal conditions. Figure 5.6 illustrates the structure of the proposed changes to the draft standard.

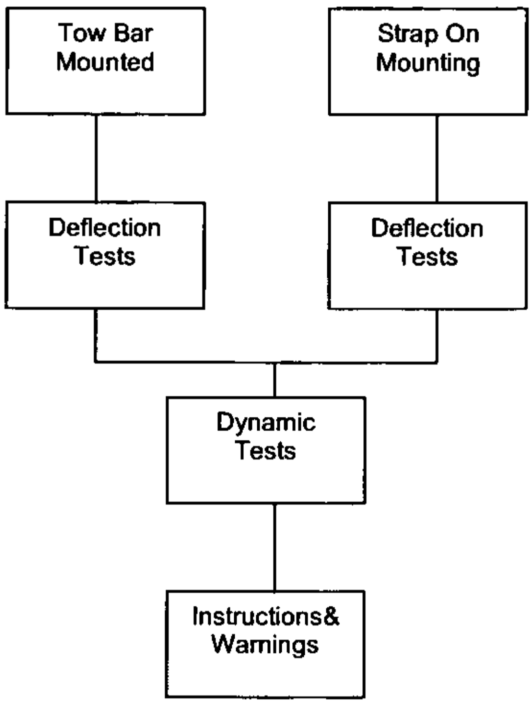


Figure 5.7 Proposed Structure for the draft ISO 15263.4 Standard.

The dynamic tests would concentrate on the racks ability to hold the bicycles in place. The warnings and instructions section would be similar with the addition of warnings about preventing failures due to wear and corrosion.

The ISO 15263.4 standard should ask 2 questions:

1. Is the rack safe enough to use with minimum risk of an in service failure?
2. Does the manufacturer inform the user how to use the rack safely without risk to others or themselves?

The second part is just as important as the first because the conclusions of DTI (2001), the first major research into bicycle rack safety, concluded that the vast majority of racks examined were safe enough, but not enough was done to make sure they were being used safely.

Alternatively, a new approach may be considered. The voting summaries show that the first 2 drafts of ISO 15263.4 failed because there were disagreements over the deflection tests. The 2004 draft featured the reduction in residual deflection from 50mm to 20mm. In the 2006 voting the BSI voted against the standard because they felt it was too strict. In line with the DTI (2001) report conclusions that most racks are safe, but not enough is done to educate customers then a draft standard may be written including only the requirements for adequate warnings on the product and instructions. (Similar standards are already in place, for example BS EN 61310 (1995) which covers the safety of machinery and includes standard symbols and warnings that should be placed on industrial machinery for workshop use.)

This standard should have a scope related to providing warnings and educating users for safe use. It should fulfil the DTI (2001) recommendations which led to the creation of the draft ISO 15263.4 standard. A standard of this type may be easier to get through the voting processes and be easier and cheaper to assess. Sample products and instructions may be sent to an independent body for assessment. Bicycle rack manufacturers could obtain ISO 9001 quality management certification, which many already have and include design and development in their scope. The ISO 9001 approval logo could then be added to the "on product" information. This would ensure that products were designed and tested in an appropriate manner that is suited to their particular design of rack.

Chapter 6

Conclusions

6 Conclusions

6.1 Deflection Tests

The Pendle TBM 3 rack would pass the deflection tests section of the draft ISO 15263.4 standard. The residual deflections are less than 9% of the 20mm limit allowed by the draft standard.

The material does not yield, maximum stress was 197 MPa during the longitudinal test, the yield strength of the material is 300 MPa.

It has been demonstrated that the arms of the rack move within the mounting block under load, this movement causes the residual deflection.

Small random errors in the measurement of the deflections were caused by the nature of the measurements and accounted for by repeating and averaging the measurements.

Systematic errors between racks may be caused by changes during the setup or manufacturing differences in the rack or the material.

There are small differences in the rack arms and mounting blocks within manufacturing tolerances.

6.2 Strain Gauge Measurements and Finite Element Analysis

There were small random errors in the measurements, but the results correlate well to a straight line.

FEA results include systematic errors that were as low as 2.5%.

Changing the boundary conditions has a significant effect on the results. There is a certain amount of trial and error in perfecting the boundary conditions that improves with experience.

Small systematic errors between the strain gauge measurements, FEA and calculated predictions are shown in the results.

Changing the mesh density had a small effect on the results of FEA, the effect may be more significant with more complex models.

FEA software should be evaluated over a wider range of problems; a correction factor or tolerance may be determinable.

When comparing the FEA results to calculated predictions of stress and deflection, the stress results are more accurate. There is a 4.6% difference in stress and a 6.4% difference in deflections. This may be due to assumptions about constraints, modelling of constraints or the nature of calculating stress and deflection and their dependence on material properties.

6.3 Predicting Failure

A bicycle rack is a potential case for fatigue failure, Soderberg analysis gives a conservative estimate of the likelihood of fatigue failure. The factor of safety in the estimate performed is greater than 2.

To improve the estimate, experimental values for the fatigue strength and the stress amplitude should be determined.

It is likely that the rack would be affected by corrosion. Work should be done to identify where the corrosion would affect and what the corrosion would do to the maximum stress in the material.

Reductions in weight, stress, deflection and cost may be achieved by changing the bend radius, tube diameter and wall thickness. The most cost effective to implement are changing the bend radius and slight reductions in wall thickness, perhaps to 2mm.

6.4 Developing the Draft ISO 15263.4 Standard

The draft ISO 15263.4 standard does not meet it's aims. The variety of bicycle rack designs means that a singe solution may not be applicable to all types of rack.

It would be advantageous to split the deflection test section of the standard between tow bar mounted and strap on bicycle racks.

The standard should answer 2 questions; is the rack safe enough and does the manufacturer ensure that the rack is used in a safe manner.

Due to the failure of previous draft standards it may be easier to create a standard that fits the findings of the DTI (2001) research that concentrates on promoting safe usage through instructions and on-product warnings.

Manufacturers with ISO 9001 approval for design and development could include this in their literature, showing that they have approved design and testing systems implemented to produce consistently safe products.

Chapter 7

Further Work

7. Further Research

7.1 Improving Fatigue Estimates

Further research should be undertaken to determine the fatigue strength of the material used in the Pendle TBM 3 rack and the stress amplitude so that a more reliable estimate of fatigue life may be made using the Soderberg method. Suresh (1991) suggests that a sample of material may be tested using the rotating beam method to determine the fatigue strength of a sample piece of material.

Lewis and Samuel (1989) suggest testing a complete component to obtain the specific fatigue strength for the component, which incorporates the effects of manufacturing processes on the material. The stress amplitude may be obtained by recording the accelerations on a rack using a 3 axis accelerometer and data logging software whilst a car is driven over various surfaces such as cobbles and brakes hard from high speed.

A test rig could be designed to simulate the life cycle on a rack by pulling on a rack in different directions to simulate cornering, braking and other forces. This could be similar to the test rigs used in the furniture industry to simulate repeated patterns of sitting on a chair or leaning back on it.

Further investigation into the effects of corrosion should be made. The main points of consideration are; the parts of the rack most likely to corrode and the effects of that corrosion on the stresses and fatigue life.

7.2 Other Types of Rack

This research should be repeated with the strap on type racks, with the aim of determining the draft ISO 15263.4 standards suitability for testing them. The work on strap on racks will focus on the straps and hooks as well as their interface to the vehicle. There are many complications in testing a Strap On type rack; the type of fibre and weave of the straps will influence the strength. There is an added complication in the way the rack loads the tailgate of the car, different cars having different designs of tailgate.

7.3 Developing Standards

To further develop future drafts for ISO 15263.4, research may analyse existing standards for labelling and warnings on products if the standard were to be developed in this direction. An example of an existing standard in this area is BS EN 61310 (1995), which covers safety of machinery.

After a relevant review it may be wise to completely re-draft the ISO 15263.4 standard or propose a new one. The ISO 15263.4 standard may have a legacy of opposition with some parties. The opportunity for all involved to contribute to a new standard learning from the lessons of the ISO 15263.4 draft could produce a better standard that satisfies the aim of guaranteeing a minimum level of safety from bicycle racks.

References

References

AutoDesk ,2007. AutoDesk Inventor User Guides – Stress Analysis

Auto Express, 1998. Product Test. 24th June 1998, pp 68-70.

Beghini, M., Betini, L., 2000. Analytical Expressions of the Influence Functions for Accuracy and Versatility Improvement in the Hole Drilling Method. *Journal Of Strain Analysis for Engineering Design,* 35 (2).

Bentley, J.,1995. Measurement Systems 3rd Edition. Harlow: Longman. pp 3-6, 113-135. ISBN : 0582237793

Berk, A., 1988. Computer Aided Design and Analysis for Engineers. Oxford: BSP Professional Books. ISBN: 063202058x

Bolton, W., 1998. *Mechanical Science Second Edition.* Oxford: Blackwell Science Ltd. pp 219. ISBN: 0632049146

Bolton, W., 1991. Instrumentation and Process Measurements. London: Longman Scientific. ISBN: 0582068088

Brady, G., Clauser, H. and Vaccari, J., 1997. *Materials Handbook Fourteenth Edition.* McGraw-Hill. New York. pp 915-917. ISBN: 0070070849

Burnett, D., 1988. Finite Element Analysis. New Jersey: Addison Wesley Publishing. ISBN: 0201108062

Clinton, K., 1995. *Cycle Racks Can Lead to Trouble*. Birmingham: Royal Society for the Prevention of Accidents (RoSPA).

Clough, R. W., Wilson, E. W., 1999. Early Finite Element Research at Berkeley [Online]. Berkley: University of California. Available at: http://Edwilson.org/my%20webs/history/fe-history.pdf [September 2007]

Consumer Safety Unit, 2001. *The Safety of Rear Mounted Cycle Carriers*. London: Department of Trade and Industry.

Cristofolini, L., Viceconti, M., 2000. Development and Validation of a Technique for Strain Measurement Inside Polymethal Methacrylate. *Journal Of Strain Analysis for Engineering Design*, 35 (1).

Dally, J., and Riley, W., 1991. Experimental Stress Analysis. McGraw Hill. USA. ISBN: 0070152187

Department of Trade and Industry (DTI). 2000. *Leave Danger Behind, 3 Steps to the Safe Use of Rear Cycle Carriers*. London: Her Majesty's Stationery Office.

De Souza, F., Bevan, N., 1990. The Use of Guidelines in Menu Interface Design – Evaluation of a Draft Standard. *Proceedings of IFIP INTERACT'90: Human-Computer Interaction.*

Dym, C. and Little, P., 2004. Engineering Design. New Jersey: John Wily and Sons. ISBN: 0471256870

Hartog, J.P.D., 1962. *Advanced Strength of Materials.* New York - Courier Dover Publications. pp 85-87. ISBN: 0486607550

Hummel, R., 1997. Understanding Materials Science. New York: Springer. ISBN: 0387209395

International Organization for Standardization (ISO). 2003. ISO/TC 22/SC 14. *Road Vehicles – rear load carrier devices Part 4: Bicycle Carriers 15263.*

International Organization for Standardization (ISO). 2006. ISO/TC 22/SC 14. *Road Vehicles – rear load carrier devices Part 4: Bicycle Carriers 15263.*

Intel. 2007. Moore's Law – Made Real by Intel Innovation. www.intel.com/technology/mooreslaw. [Accessed June 2007].

Johnson, A., and Sherwin, K., 1996. *Foundations of Mechanical Engineering.* Cheltenham - Nelson Thornes. ISBN: 0748764232

Kaye, R., Heller, M., 2000. Investigation of shape optimization for the design of life extension options for an F/A 18 airframe FS470 bulkhead. *Journal Of Strain Analysis for Engineering Design.* 35 (6).

Kirkup, L,. 1994. Experimental Methods - An Introduction to the Analysis and Presentation of Data. Queensland: Wiley and Sons. ISBN: 0471335797

Lewis, W., Samuel, A., 1989. Fundamentals of Engineering Design. Australia: Prentice Hall. ISBN: 0724804765

MBUK, 1999. Grouptest: Which Rack. Mountain Biking UK Summer 1999 Edition. pp 91-94.

Measurements Group. 1992. Student Manual for Strain Gage Technology. (s.l.) Measurements Group Inc.

McMahon, C., and Browne, J., 1998. *CAD CAM Principles, Practice and Manufacturing Management.* Harlow: Prentice Hall. pp 188-197. ISBN: 0201178192

Osgood, C., 1970. Fatigue Design. New York: John Wiley and Sons. pp 405. ISBN: 0471657115

Practical Caravan, 2003. Product Test: Pedaller on the Roof. December 2003, pp 72-74.

Ray, M., 1988. Engineering Experimentation : Ideas Techniques and Presentation. London: McGraw Hill Book Company.

Ray, M. H., 1997. The Use of Finite Element Analysis in Roadside Hardware Design. *International Journal of Crash Worthiness*, 2 (4).

Rees, D., 2003. Mechanics of Solids and Structures. London: Imperial College Press. ISBN: 1860942180

Rooney, J. and Steadman, P., 1993. Principles of Computer Aided Design. London: Pitman Publishing. ISBN: 1857282221

RS. 1997. Strain Gauges and Load Cells – RS Data Sheet. RS. Corby. Issued March 1997.

Shigley, J., 2004. *Mechanical Engineering Design International Edition.* New York: McGraw-Hill. pp. 323-328. ISBN: 0071232702

Singletrack, 2005. Through the Grinder. September 2005, Issue 23. pp 42.

Suresh, S., 1991. Fatigue of Materials. Cambridge: Cambride University Press. pp 275. ISBN: 0521437636

Woodward, C., 1972. BSI – The Story of British Standards. London: British Standards Institution. ISBN: 0580075915

What Car, 2003. Carry on Cycling. What Car Magazine May 2003. pp 18-21.

Yahiaoui, K., Moffat, D.G., Moreton, D. N., 2000. Piping Elbow with Cracks Part 1: A Parametric Study of the influence of Crack Size on Limit Loads due to the Pressure and Opening Bending. *Journal Of Strain Analysis for Engineering Design*, 35 (1)