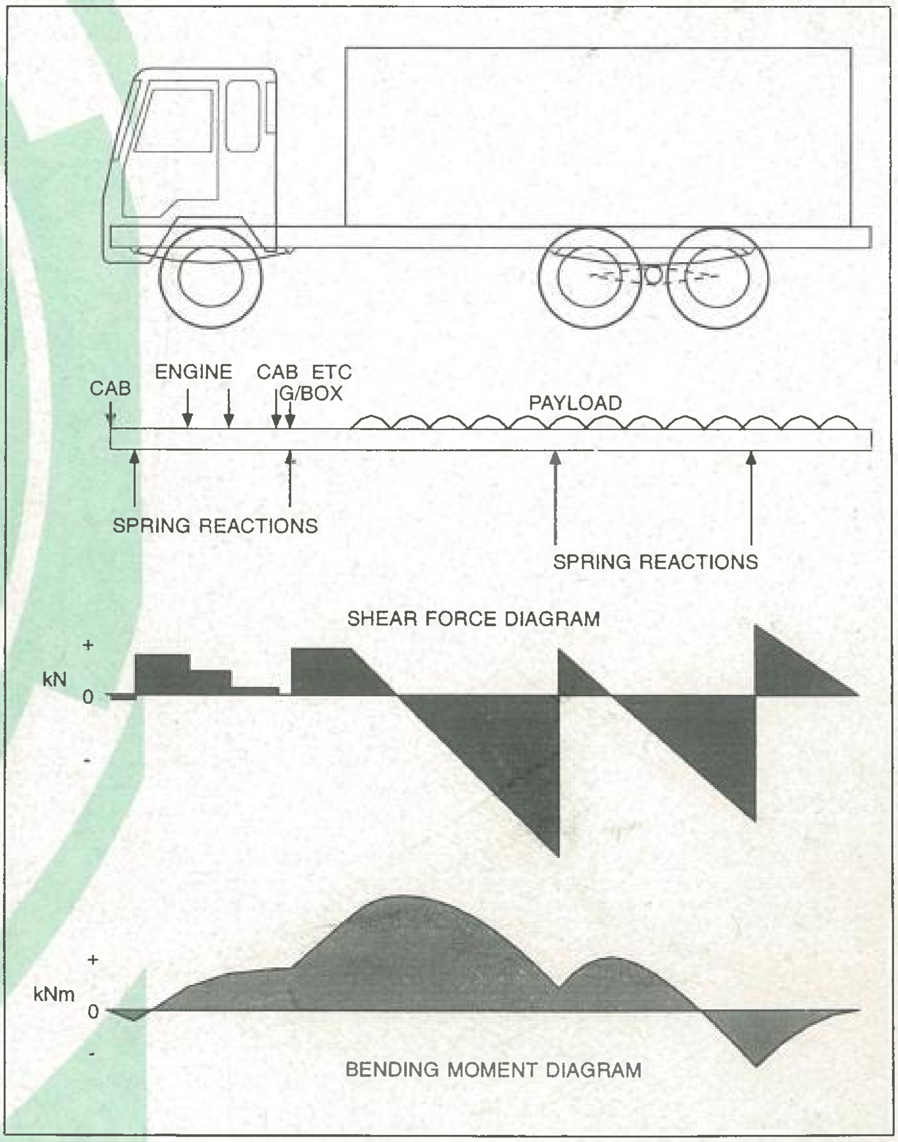
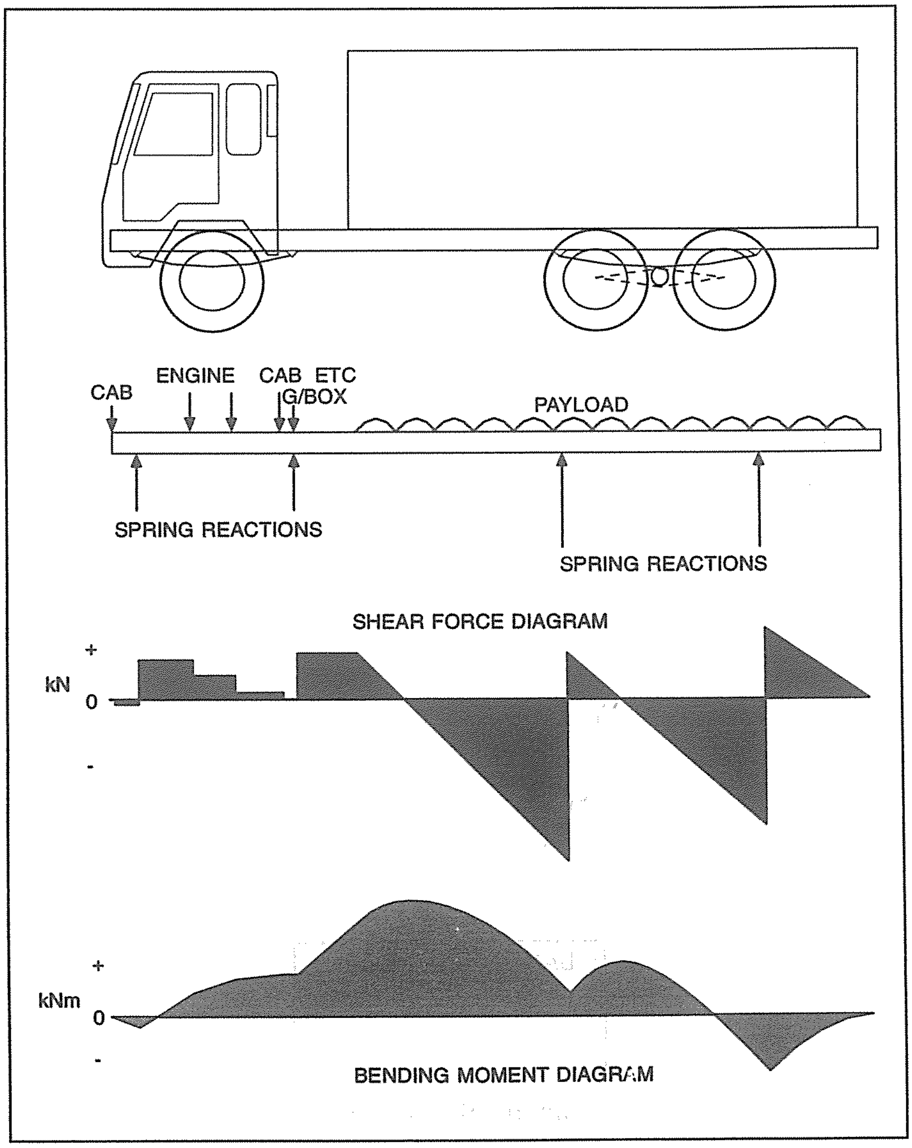
THE NEW ZEALAND INSTITUTE FOR INDUSTRIAL RESEARCH AND DEVELOPMENT

INDUSTRIAL RESEARCH LIMITED

LADDER FRAME CHASSIS DESIGN & MODIFICATION



LADDER FRAME CHASSIS DESIGN & MODIFICATION



This Guide has been prepared by:

Richard Wong
INDUSTRIAL RESEARCH LTD
24 Balfour Road
Parnell
PO Box 2225
Auckland
New Zealand
Phone +64-9-303 4116
Fax +64-9-307 0618

in association with

LAND TRANSPORT
SAFETY AUTHORITY
Pearse House
225 Willis Street
PO Box 27-459
Wellington
New Zealand
Phone +64-4-382 8300
Fax +64-4-385 5699

Industrial Research Limited Report Number 337

ISBN 0-909040-11-7
ISSN 1171-9281

CONTENTS

LADDER FRAME CHASSIS DESIGN & MODIFICATION

1. INTRODUCTION

It is hoped that by publishing this paper some recognition has been made of the difficulties that exist in New Zealand in the area of heavy vehicle design and modification.

In New Zealand there are very few standards or design rules regulating the design of heavy vehicles. It is therefore, to a large extent up to the engineer doing the modification to define what is required in the design.

The paper is not intended to be a set of regulations controlling the modification of heavy truck and trailer ladder frame chassis.

It is the intention of this paper to bring together the most important information which is already available in the literature. It should then be possible to carry out modifications in a thoughtful and thorough manner. The paper should only be used as a set of recommendations to guide the designer through the most important aspects of the chassis modifications.

It must be emphasised that most vehicle chassis produced by the major manufacturers overseas have been subject to very extensive design analysis and testing. It is an extremely expensive process to accurately verify the design of a vehicle chassis which will be used in such a wide variety of environments.

Unfortunately the conditions and the regulations overseas are usually different from New Zealand and it is inevitable that users here seek to make changes to the original specifications to improve the efficiency of operation. It would be unrealistic to expect the modified chassis to go through the same analysis and testing as the original chassis. The facilities to carry them out are not available in New Zealand or the costs are far in excess of the value of the modifications.

2. CHASSIS FUNCTION

What is the purpose of a heavy vehicle chassis?

Obviously the primary function of the chassis is to carry the payload and the body.

As well as this, the chassis retains the alignment of the axles and (in the case of a truck) the driveline.

It must also transmit all the forces due to steering, braking and drive to the mass of the vehicle.

By far the most common type of chassis used in heavy vehicles is the ladder frame chassis. This is type of chassis that is dealt with in this paper. A ladder frame chassis consists of two siderails running the length of the vehicle connected by crossmembers at the ends and at various positions in between.

The siderails provide the strength in bending and the crossmembers retain the relative positions of the siderails and resist local twisting due to offset loading. They can also provide support for the engine, transmission, suspension etc.

Because of the typical characteristics of heavy vehicle suspensions (stiff springs and limited suspension travel) the chassis must also be able to twist or 'weave' to cope with uneven road surfaces. This is especially true of off-highway type vehicles.

3. CHASSIS LOADS

The loads imposed on a chassis in normal service can be divided into four types;

3.1. Vertical Bending. The static weight of the chassis, components (eg. the fuel tank) and payload produce vertical bending, as do the dynamic loading due to bumps during driving. Where components are mounted to the chassis with brackets which offset the mass, extra bending or torsional loads are superimposed on top of the simple vertical loads.

3.2. Torsion. Asymmetric distribution of the payload or components, a twisted ground plane, lateral acceleration and severe manoeuvring all produce torsion of the chassis (see figure 3.1).

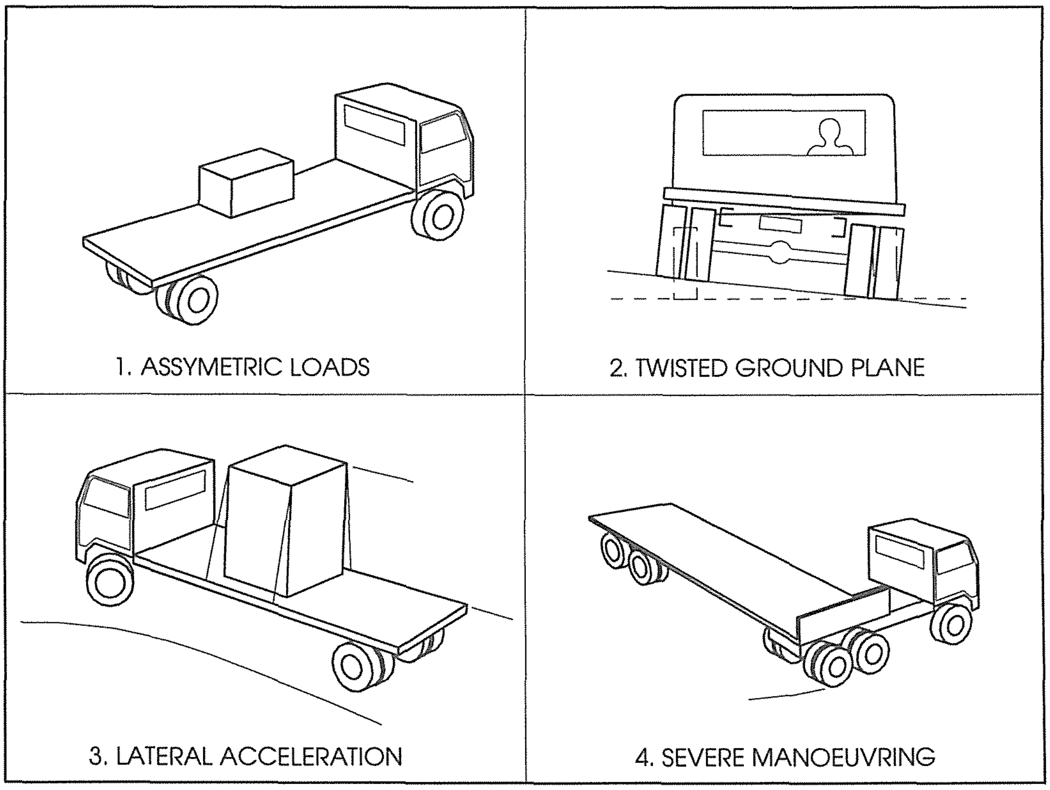


Figure 3.1

3.3. Lateral Bending. This can be caused by sideways steering forces especially during severe manoeuvring at parking speeds (see figure 3.2).

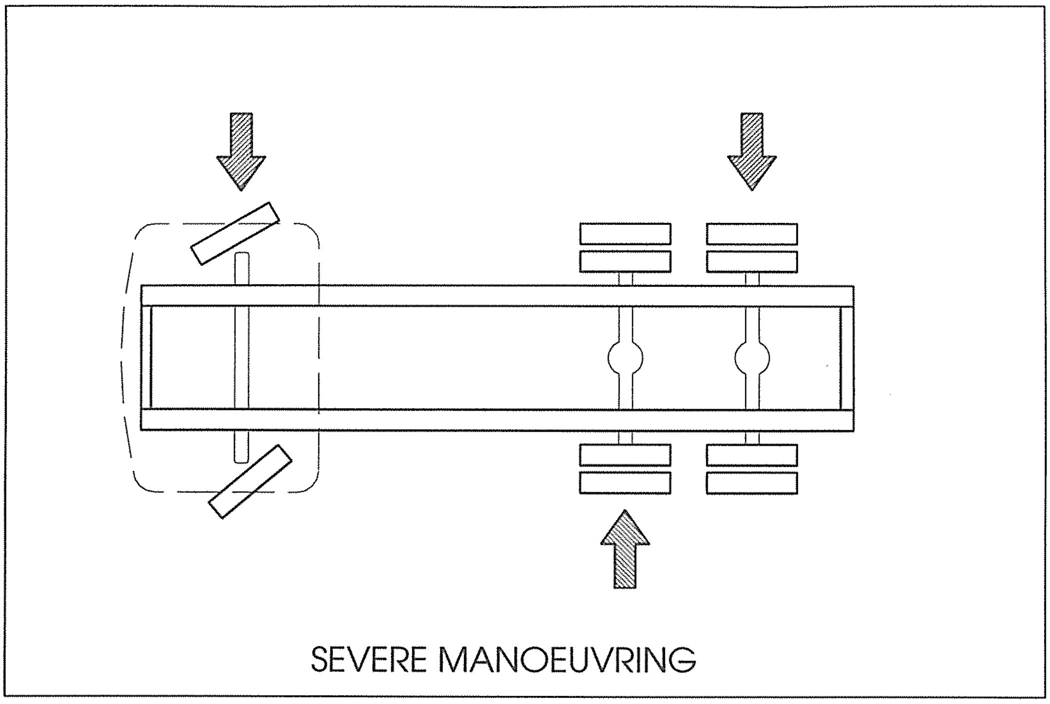


Figure 3.2

3.4. Axial Bending. This is also known as parallelogramming. This is the tendency under some conditions for the siderails to move longitudinally relative to each other. An example of this is the effect of uneven braking forces on the left and right hand sides.

4. FAILURE MODES

There are several different occurrences which can be considered to constitute failure in a chassis.

The most well known is tensile yield. This happens when the tensile stress in the material exceeds the elastic limit. This usually results in permanent deformation of the structure. It is also possible to yield the material in compression or shear. Typically yield occurs as the result of a single event such as the accidental overloading of the chassis.

It is also possible (and relatively common) to cause cracking in a chassis through repeated loading at stress levels much lower than the yield stress. This is called fatigue. It normally occurs in areas of stress concentration such as around bolt holes or sudden changes of cross-section as well as at welded joints.

When a beam fails in compression or bending due to instability of the section this is commonly known as buckling. This can occur in lightweight, thin sections. When minimum weight is the prime consideration and thinner sections are being used it is especially important to check that buckling does not occur.

Although not strictly a failure, excessive deformation (inadequate stiffness) of a chassis is also unacceptable. It can result in damage to bodies or components attached to the chassis or possibly give rise to unwanted dynamic resonances in the chassis which could cause other failures.

5. DESIGN CHECKS

It is assumed in the following that modifications will be made to an existing ladder frame chassis in which the original has been adequately designed in terms of stress and stiffness for its original design payloads.

Manufacturers will often publish their own set of recommendations on the correct procedure for carrying out modifications. The design checks given here are not meant to replace those recommendations. They are additional to the manufacturers recommendations or can be used when manufacturers information is not available.

Typical modifications which are dealt with are; adding or shifting axles, lengthening and shortening chassis.

It is worth noting here that although there are few design rules in New Zealand applicable to the design and modification of heavy vehicle chassis, the Queensland Department of Transport in Australia has a **Code of Practice for Commercial Motor Vehicle Modifications**. This covers a wide range of modifications including the engine, transmission, driveline, suspension etc. It also includes checklists to ensure compliance with Australian Design Rules in Queensland.

5.1 VERTICAL BENDING

The primary check which needs to be made on a chassis is its strength in this mode.

A ladder frame chassis is much like a bridge in that the payload applies a vertical force downward on the chassis and the springs/suspension provide the upward reactions to those forces. The siderails of the frame can therefore be treated as beams in bending. The crossmembers contribute no strength in this mode. (See figure 5.1).

The designer must establish how the major masses are distributed along the chassis and how they are connected to it. The masses of the axles, wheels and tyres do not contribute to the downward forces on the chassis. However the relative positions of these points on the chassis will determine what the reaction forces are.

This information can then be used to draw a **Bending Moment Diagram**. Take a nominal value of 9.81 N/kg (one g). Calculate the bending moment in Nm along the modified chassis. The maximum bending stress in the siderails (at one g) can then be calculated using the second moment of area and the distance from the neutral axis of the upper and lower surface of the appropriate section.

The same procedure should be used to calculate the maximum stress in the unmodified chassis at one g with the maximum allowable payload. The distribution of the payload can have a significant effect on the maximum stress in the chassis. For example point loads produce higher bending moments than distributed loads and loads positioned far away from reaction points produce higher bending moments than loads acting close to the reactions. One should always assume the worst case that can occur in real circumstances.

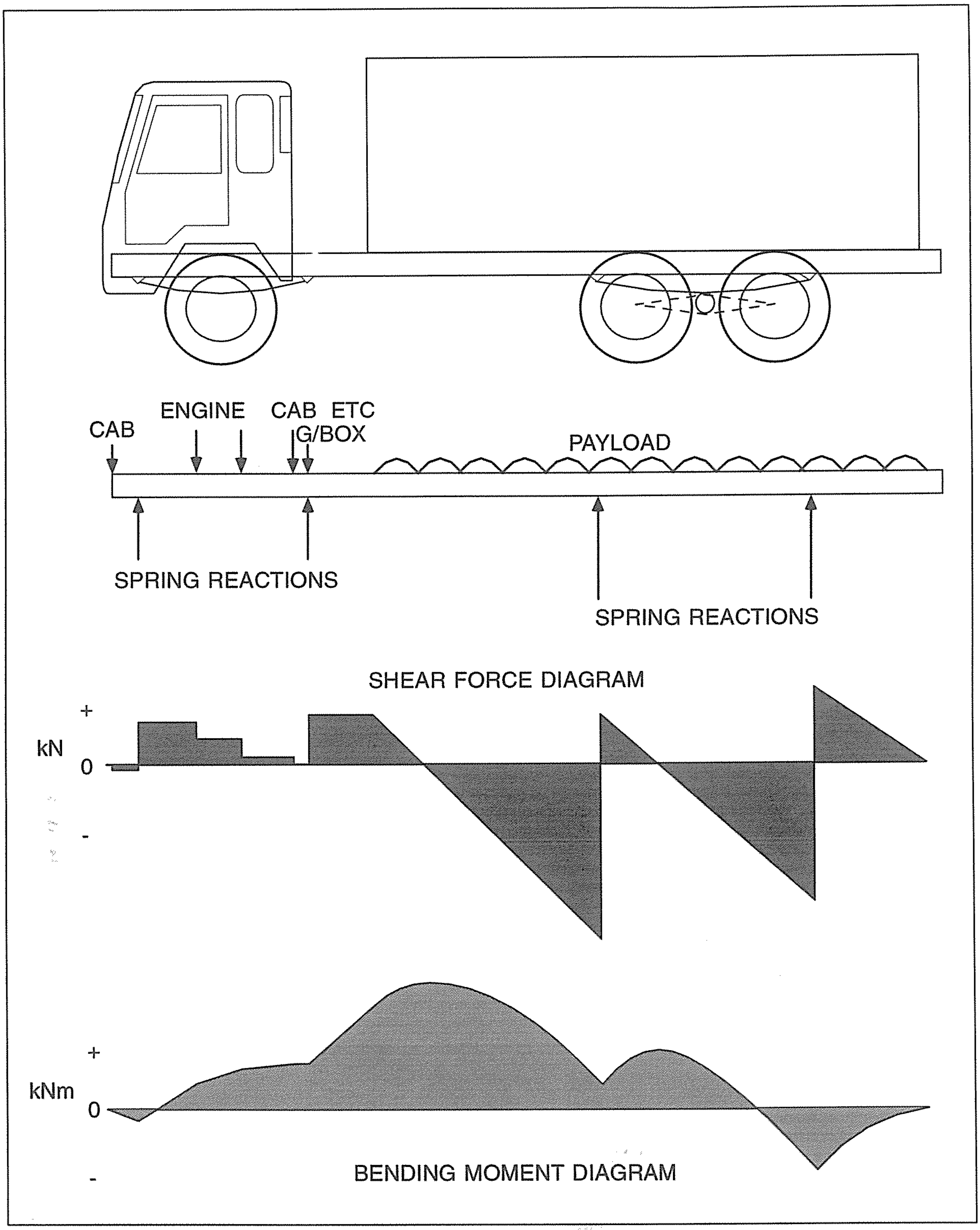


Figure 5.1

The maximum stress in the modified chassis should not normally exceed the maximum stress in the unmodified chassis (for the manufacturers' recommended maximum payload). Where there are no manufacturers' recommendations it is necessary to establish all the relevant material properties of the chassis. If changes are being made which significantly alter the use to which the chassis is being put, and it is necessary to deviate from the manufacturers' recommendations, it is essential to establish all the relevant material properties, and to use these as the appropriate maximum stresses.

When using reinforcements such as extra flitching or channel section attached to the original chassis it is normal to specify material with the same specification (yield strength) as the unmodified chassis. This way the full strength of both the original chassis and the reinforcement metal will be utilised. If the flitching has a much higher yield strength than the unmodified chassis then the chassis is likely to fail long before the reinforcement. If the flitching has a lower yield strength than the original chassis it will be impossible to achieve the same maximum stress as for the unmodified chassis. (These comments apply when the original chassis and the extra flitching both achieve the same magnitude of stress under the critical loading situation).

When no information is available on the specifications (maximum payloads) of the original chassis or if a completely new chassis is being designed, what parameters should be used for the maximum vertical loads?

William Sidelko [1] uses a factor of 1.75 times the static load to allow for dynamic forces and a general "safety" factor of 1.70. Multiplying these together gives a factor of 3.0. The stresses produced in vertical bending using 3.0 times the static load must be less than the yield strength of the material. The maximum dynamic forces experienced by a chassis are likely to vary depending on the type of application. In some cases manufacturers use a factor of 3.4 against yield.

The bending strength of the chassis may also be needed to provide reaction to braking forces on the payload. The centre of gravity of the payload will be at some height above the chassis. A maximum braking deceleration of 1.0g (this is very conservative for typical heavy vehicles) will produce a horizontal force on the payload. This force multiplied by the distance above the chassis gives the moment which must be resisted by the chassis. Of course, when calculating the stresses it is essential to consider how the loads are applied into the chassis.

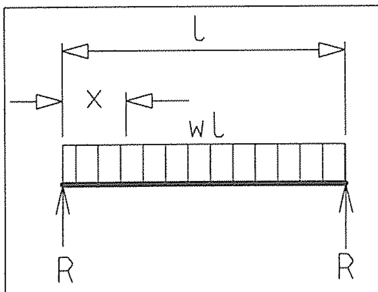
It should be noted here that if the vertical loads are not applied through the "shear centre" of a section then extra torsional stresses will be added to the pure bending stresses. Sections which are symmetrical about the vertical axis such as I beams or rectangular hollow sections have their shear centre at the geometric centre of the section. Channels however have their shear centre some distance to the outside of the web. In these cases it is advantageous to use outrigger brackets to ensure loads are applied through the shear centre. Section 5.4 discusses torsion and figure 5.10 gives the position of the shear centre for a channel.

5.2 VERTICAL BENDING DEFLECTION

The maximum vertical deflection allowable in a ladder frame chassis is not expressly defined by any engineering standards or regulations. However the available literature suggests that a maximum of 10 - 12.7 mm difference between the bare chassis and the fully laden chassis be used. This corresponds to a fundamental resonant frequency of around 5 Hz. Greater vertical deflection under static load would result in a lower resonant frequency, which would begin to approach the bounce mode of most heavy vehicle suspensions. This could lead to severe dynamic problems which could also result in structural damage to the vehicle.

The load distribution along a vehicle chassis may be quite complex but it can often be idealised into combinations of quite simple cases. When this is done and if the chassis rails have a constant section the deflections as calculated in table 5.1 and table 5.2 can be added together to give the total deflection under the maximum loading. For more complex loading situations the designer is referred to a number of other texts [2-4].

If it can be shown that the unmodified chassis has a maximum vertical deflection of greater than 12.7 mm for its original design load then this greater value should be acceptable for the modified chassis.

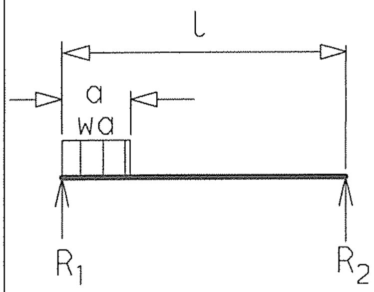


$$\text{Total Load} = wl$$

$$R = \frac{wl}{2}$$

$$\Delta_{max} \text{ (at center)} = \frac{5wl^4}{384\,EI}$$

$$\Delta_x = \frac{wx}{24EI}(l^3 - 2lx^2 + x^3)$$

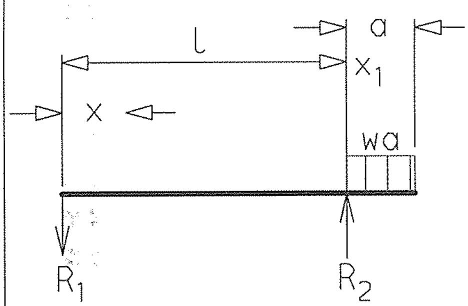


$$R_1 = \frac{wb}{2l}(2l-a)$$

$$R_2 = \frac{wa^2}{2l}$$

$$\Delta_x \text{ (when } x<a) = \frac{wx}{24EIl}\left(a^2(2l-a)^2 - 2ax^2(2l-a) + lx^3\right)$$

$$\Delta_x \text{ (when } x>a) = \frac{wa^2(l-x)}{24EIl}(4xl - 2x^2 - a^2)$$



$$R_1 = \frac{wa^2}{2l}$$

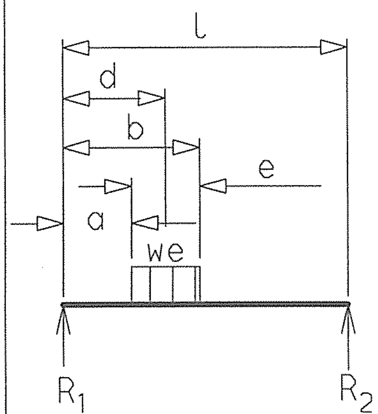
$$R_2 = \frac{wa}{2l}(2l+a)$$

$$\Delta_{max}\left(\text{btwn supports at } x = \frac{l}{\sqrt{3}}\right) = 0.03208\,\frac{wa^2 l^2}{EI}$$

$$\Delta_{max}\left(\text{for overhang at } x_1 = a\right) = \frac{wa^3}{24EI}(4l+3a)$$

$$\Delta_x \text{ (between supports)} = \frac{wa^2 x}{12EIl}(l^2 - x^2)$$

$$\Delta_{x_1} \text{ (for overhang)} = \frac{wx_1}{24EI}(4a^2 l + 6a^2 x_1 - 4ax_1^2 + x_1^3)$$



$$d = \frac{(a+b)}{2}$$

$$R_1 = \frac{wed}{2l}$$

$$R_2 = \frac{we}{2l}(a+b)$$

$$\Delta_x \text{ (when } x \leqslant a) = \frac{1}{48EI}\left\{8R_1(x^3 - l^2 x) + wex\left[\frac{8d^3}{l} - \frac{2be^2}{l} + \frac{e^3}{l} + 2e^2\right]\right\}$$

$$\Delta_x \text{ (when } a<x<b) = \frac{1}{48EI}\left\{8R_1(x^3 - l^2 x) + wex\left[\frac{8d^3}{l} - \frac{2be^2}{l} + \frac{e^3}{l} + 2e^2\right] - 2w(x-a)^4\right\}$$

Table 5.1

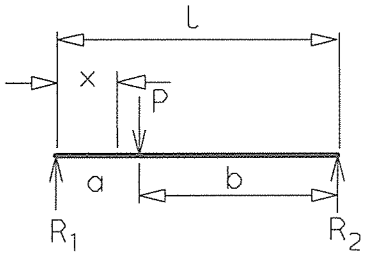
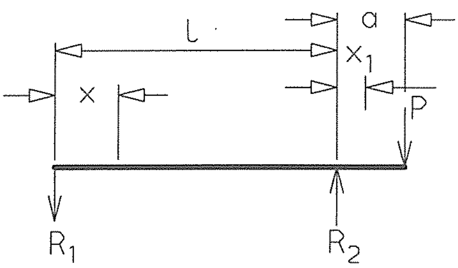
	$R_1 = \frac{Pb}{L}$ $R_2 = \frac{Pa}{L}$ $\Delta_{max}\left(\text{at } x = \sqrt{\frac{a(a+2b)}{3}} \text{ when } a<b\right) = \frac{Pab(a+2b)\sqrt{3a(a+2b)}}{27EIL}$ $\Delta_x \text{ (when } x<a) = \frac{Pbx}{6EIL}(L^2-b^2-x^2)$
	$R_1 = \frac{Pa}{L}$ $R_2 = \frac{P}{L}(L+a)$ $\Delta_{max}\left(\text{btwn supports at } x=\frac{L}{\sqrt{3}}\right) = 0.06415\frac{PaL^2}{EI}$ $\Delta_{max}(\text{for overhang at } x_1=a) = \frac{Pa^2}{3EI}(L+a)$ $\Delta_x \text{ (between supports)} = \frac{Pax}{6EIL}(L^2-x^2)$ $\Delta_{x_1} \text{(for overhang)} = \frac{Px_1}{6EI}(2aL+3ax_1-x_1^2)$

Table 5.2

5.3 BUCKLING

In a ladder frame chassis there are two types of buckling that can possibly occur, local compressive buckling or lateral buckling due to bending.

5.3.1 Local Buckling

Local buckling in ladder frame chassis is dependant on the b/t ratio (ie. the width divided by the thickness) of the webs and flanges which make up the chassis members. It is unusual to change the thickness or the overall dimensions of the original chassis in heavy vehicle modifications so it is unlikely that local buckling will be a problem in the main chassis rails. However it is common to add or modify flitching or reinforcing sections to the original chassis rails. In terms of weight it will normally be most effective to use wide thin sections to obtain the desired section modulus. This type of section is more likely to fail in buckling so it is important for the designer to ensure that those plates which are in compression will not buckle locally.

Figure 5.2 illustrates the two modes of local compressive buckling that can possibly occur in a chassis rail. This shows that long plate elements which are free along one edge buckle quite differently from a plate which is simply supported on all four edges. The flanges of channels, angles and I beams are typical of plates with a free edge. Any face of a rectangular hollow section behaves like a plate which is simply supported on all four edges (figure 5.3).

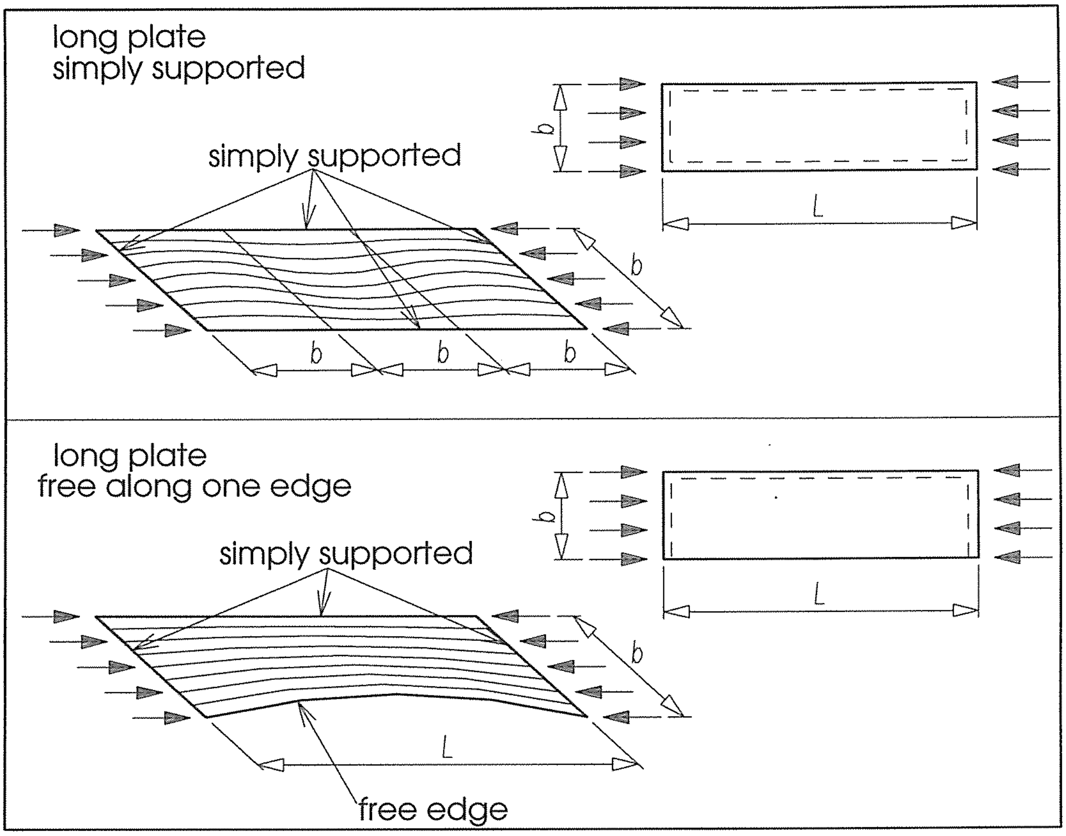


Figure 5.2

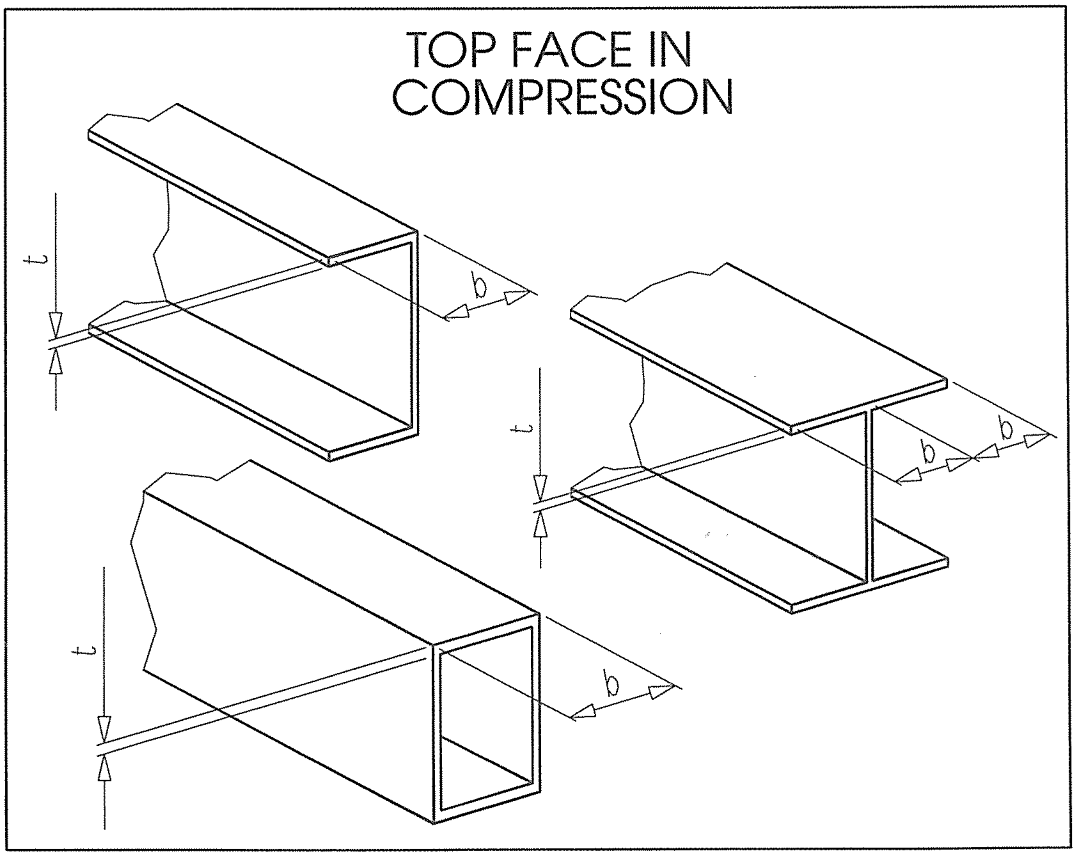


Figure 5.3

The earlier check for vertical bending strength will have ensured that stresses will not exceed yield. In fact they will not exceed the maximum stresses in the unmodified chassis. So if the b/t ratios for the reinforcing flitches are equal to or less than the b/t for the original chassis rail at the same place, local buckling should not occur. Note that this comparison is only valid if the buckling mode for the reinforcement is the same as for the chassis rail. For example it is true when comparing the flange of an angle flitch or a channel to the flange or a channel type chassis rail. It is not valid to use the same b/t ratio for an angle flitch when the chassis has a rectangular hollow section.

The above comparison is very simple to make but it will always be very conservative. The chassis rails of heavy vehicles are seldom designed to approach b/t ratios which will lead to local buckling. It will therefore be possible to use reinforcement with b/t ratios much higher than the original chassis rails. This should allow lighter and more efficient flitching. Figure 5.4 shows b/t ratios which theoretically allow yielding to occur before local elastic buckling. These values are still conservative and may be used to check chassis modifications.

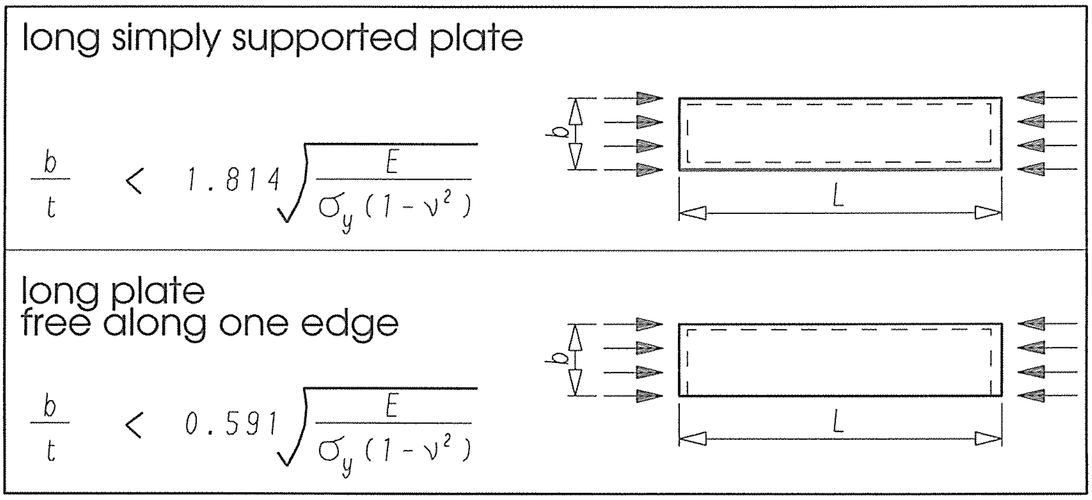


Figure 5.4

5.3.2 Lateral Instability

Long narrow beams (such as chassis rails with great distance between cross members) are subject to instability called elastic flexural-torsional buckling. This effect is apparent with sections which have comparatively low torsional stiffness such as open sections (eg. I sections and channels). Figure 5.5 illustrates this mode of failure.

In almost all chassis modifications it will be unnecessary to do a full analysis of the critical stress in lateral buckling. The design checks above will ensure the maximum stresses will be kept lower than the original chassis, and the original chassis section will be retained. Therefore it will be sufficient to ensure that the distance between crossmembers (ie. the distance between supports) is less than or equal to the distance on the original chassis.

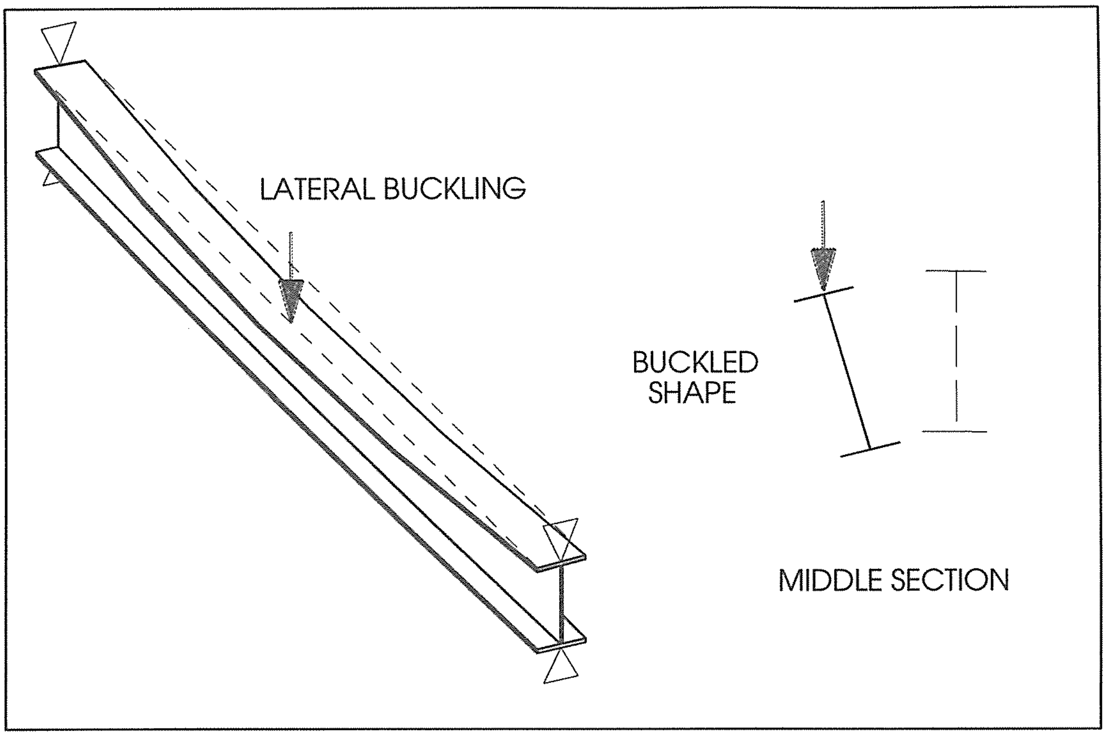


Figure 5.5

Lateral buckling is a complex problem. Take the relatively simple case of a beam where there is bending through the shear centre axis and where the ends are simply supported and free to warp but not twist. The maximum stress before lateral buckling is affected by the distance between supports, the warping coefficient, the section modulus in the x direction, the second moment of area in the y direction, the torsional constant of the section and the Young's modulus and shear modulus of the material.

The Australian Standard AS 1250 uses the following criteria to check for lateral buckling in doubly symmetric I beams and symmetric channels subjected to the above conditions.

The maximum allowable bending stress

$$F_b = 0.66 \frac{\sigma_{ob}}{\left[1+\left(\frac{\sigma_{ob}}{\sigma_y}\right)^2\right]^{0.5}}$$

where σ_{ob} = the Critical Bending Stress

and σ_y = the Yield Stress

Now the critical bending stress for the above can be calculated from

$$\sigma_{ob} = \pi\alpha_m \frac{\bar{y}}{I_{xx} l}\left[EI_{yy}\left(GJ + \pi^2 E\frac{C_w}{l^2}\right)\right]^{0.5}$$

where α_m is a factor which varies according to the bending moment distribution and is equal to a maximum of 1.0 for a constant bending moment along the beam.

Other values of α_m are shown in table 5.3.

E is Young's modulus which is about 200 GPa for steel and G is the shear modulus which is about 78 GPa for steel. C_W is the warping coefficient of the section and l is the distance between the supports. C_W for I sections and for symmetric channels is given in figure 5.10.

For a thorough analysis of lateral buckling the references [5-6] at the end of this paper are recommended.

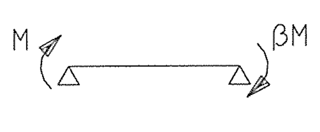
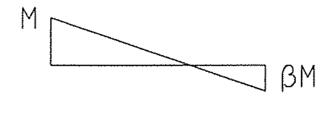
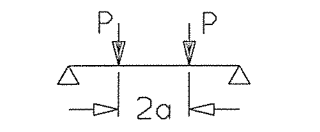
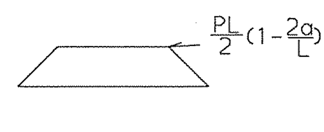
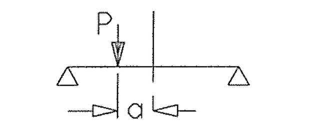
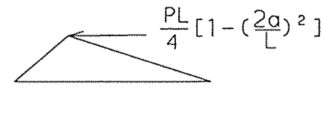
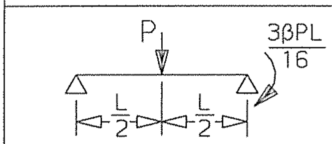
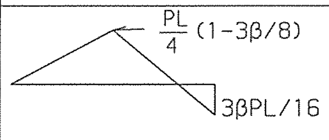
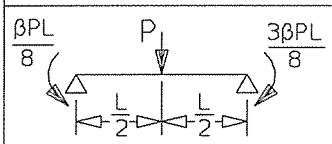
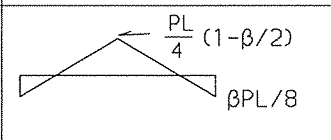
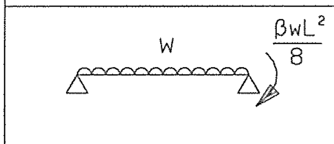
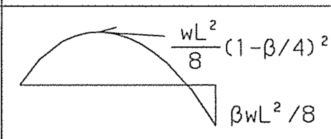
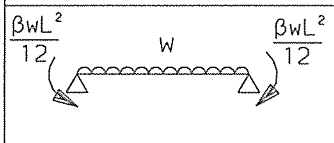
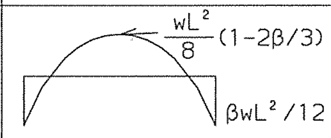
Beam segment	Moment distribution	α_m	Range
		$1.75+1.05\beta+0.3\beta^2$ 2.5	$-1\leq\beta\leq0.6$ $0.6<\beta\leq1$
		$1.0+0.35(1-2a/L)^2$	$0\leq\frac{2a}{L}\leq1$
		$1.35+0.4(2a/L)^2$	$0\leq\frac{2a}{L}\leq1$
		$1.35+0.15\beta$ $-1.2+3.0\beta$	$0\leq\beta<0.9$ $0.9\leq\beta\leq1$
		$1.35+0.36\beta$	$0\leq\beta\leq1$
		$1.13+0.10\beta$ $-1.25+3.5\beta$	$0\leq\beta\leq0.7$ $0.7\leq\beta\leq1$
		$1.13+0.12\beta$ $-2.38+4.8\beta$	$0\leq\beta\leq0.75$ $0.75\leq\beta\leq1$

Table 5.3

5.4 TORSION

As discussed there are basically four conditions which lead to torsion in a ladder frame chassis (figure 3.1).

1) Asymmetric load: This causes deflection in the chassis and the suspension. The amount depends on the relative stiffness of the front and rear suspension as well as the position of the load. A conventional chassis has a very high torsional stiffness in this mode (where torque is applied half way along the wheelbase).

2) Twisted ground plane: Here the suspension takes some of the twist. The torque in the chassis is approximately proportional to it's torsional stiffness.

3) Lateral Acceleration: The centre of gravity height of the vehicle is unlikely to coincide with the height of the chassis. This means that when the vehicle is cornering the lateral acceleration produces a torque on the chassis (figure 5.6). The lateral acceleration also produces roll (side sway) of the payload and chassis which in turn increases the torque on the chassis ie. the effect is non-linear.

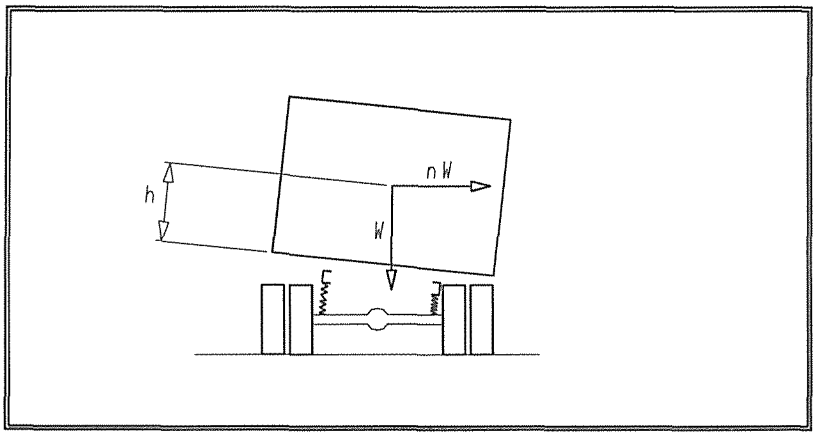


Figure 5.6

4) Severe Manoeuvring: (See figure 3.1 and 3.2) In all multi-axled groups which are not steered, severe manoeuvring produces large tyre scrubbing forces which cause large torques to exist between the front and rear mounting points of the suspension for the group (figure 5.7). It also produces parallelogramming which has to be resisted by the joints between the longitudinals and the crossmembers.

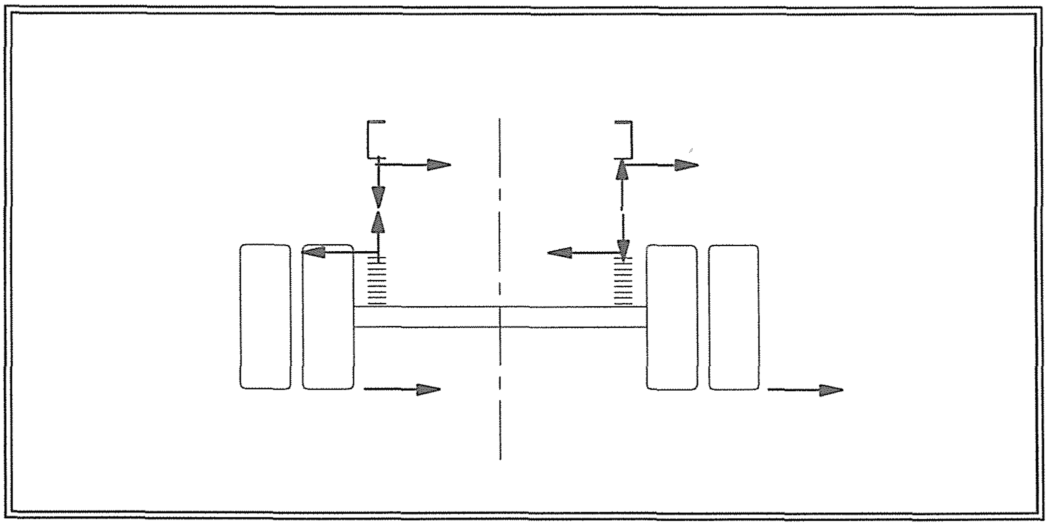
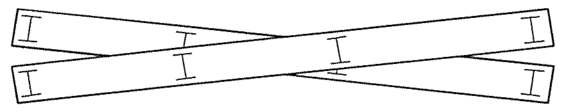


Figure 5.7

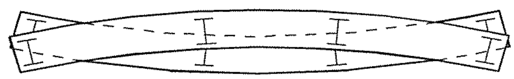
For cases 2 and 4 (see figure 5.8) the longitudinals are so stiff in bending the crossmembers are all twisted through almost the same angle so the torque is easily calculated.

For cases 1 and 3 the torque is largely resisted by differential bending of the longitudinals (see figure 5.9) and relatively little torque is produced by the crossmembers.



cross members twisted equally

Figure 5.8



differential bending of longitudinals

Figure 5.9

Pure torsion (where plane sections remain plane) in an elastic member is given by the following formula:

$$\text{torque } T = \frac{G\theta}{l}$$

$$\text{and shear stress } \tau = \frac{rT}{J}$$

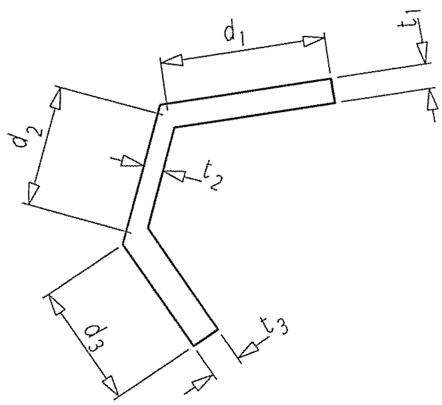
Some torsion constants (J) for thin walled sections are given in figure 5.10.

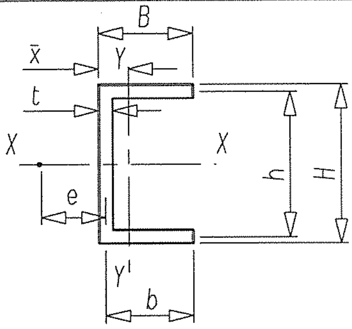
In general for thin walled open sections (ends free to warp):

$$T = \frac{1}{3}\sum\left(t^3 d\right)\frac{G\theta}{l}$$

$$\left[\tau = 2G\frac{\theta}{l}x\right]$$

$$\tau_{max} = \frac{3T t_{max}}{\sum t^3 d}$$





$$I_{xx} = \frac{BD^3 - (B-t)(D-2t)^3}{12} \qquad A = \text{area}$$

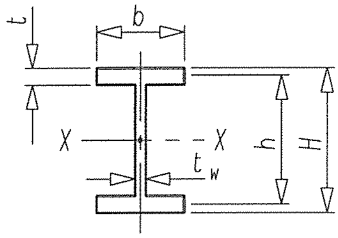
$$\bar{x} = \frac{Ht + 2(B-t)(B+t)}{2(H+2(B-t))}$$

$$I_{yy} = \frac{H}{3}B^3 - \frac{(B-t)^3}{3}(H-2t) - A(B-\bar{x})^2$$

$$J = \frac{(2B + H-2t)t^3}{3}$$

$$e = \frac{3b^2}{h+6b}$$

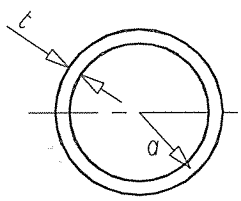
$$C_w = \frac{h^2b^3t}{12} \cdot \frac{2h+3b}{h+6b} \qquad K = \frac{t^3}{3}(h+2b)$$



$$I_{xx} = \frac{bH^3 - (b-t_w)(H-2t)^3}{12}$$

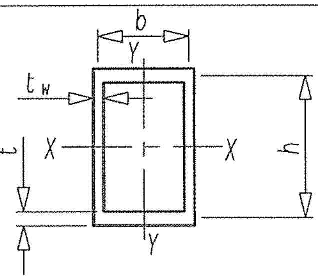
$$I_{yy} = \frac{bt^3}{6} + \frac{t_w^3(H-2t)}{12} \qquad K = \frac{1}{3}(2t^3b + t_w^3h)$$

$$J = \frac{2bt^3 + (H-2t)t_w^3}{3} \qquad C_w = \frac{h^2b^3t}{24}$$



$$I_{xx} = I_{yy} = \frac{\pi}{64}((2a+t)^4 - (2a-t)^4)$$

$$J = 2\pi a^3 t$$



$$I_{xx} = \frac{(b+t_w)(h+t)^3 - (b-t_w)(h-t)^3}{12}$$

$$I_{yy} = \frac{(b+t_w)^3(h+t) - (b-t_w)^3(h-t)}{12}$$

$$J = \frac{2b^2h^2t_wt}{bt + ht_w}$$

Figure 5.10 Selected Section Properties

For thin walled sections where the ends are restrained from warping as shown in figure 5.11:

$$\theta = \frac{T_o}{C_w E \beta^3}[\beta l - Tanh \beta l]$$

$$\theta''(\text{max.}) = \frac{T_o}{C_w E \beta} Tanh \frac{\beta l}{2}$$

$$\beta = \left(\frac{KG}{C_w E}\right)^{0.5}$$

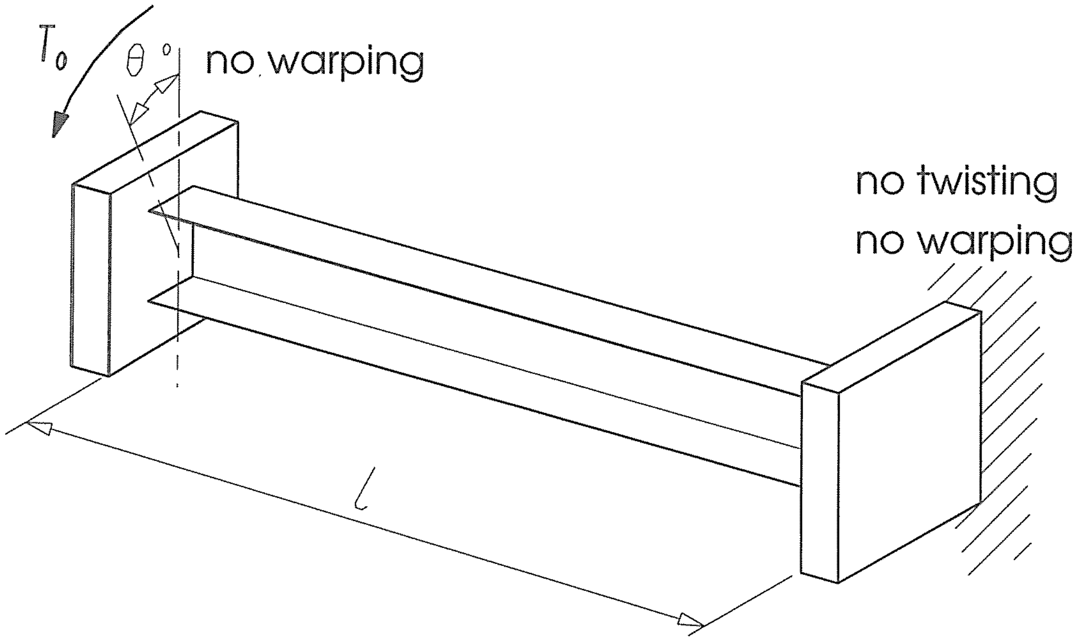


Figure 5.11

For the channel section in figure 5.10, the maximum bending stress due to warping occurs at the open edge of the flanges and is;

$$\sigma_{\max} = \frac{hb}{2}\frac{h+3b}{h+6b}E\theta''$$

and the maximum shear stress due to warping is;

$$\tau_{\max} = \frac{hb^2}{4}\left(\frac{h+3b}{h+6b}\right)^2 E\theta''$$

at a distance from the same edges of;

$$b\frac{h+3b}{h+6b}$$

For the I section in figure 5.10, the maximum bending stress due to warping occurs at the edges of the flanges and is;

$$\sigma_{\max} = \frac{\text{hb}}{4}E\theta''$$

and the maximum shear stress due to warping is;

$$\tau_{\max} = -\frac{hb^2}{16}E\theta''$$

at the intersection of the webs and flanges.

P.W. Sharman [7] suggests that the design criteria in torsion should be that the yield stress should not be exceeded for the following cases:

1) Asymmetric Loading: Half the maximum payload over half the width of the deck.
2) Twisted Ground Plane: 150 mm elevation under one side of the rear suspension or a 4.3 degree twist of the ground plane (whichever is greater).
3) Cornering Lateral Acceleration: 0.3g (3 N/kg) on half the payload with a centre of gravity 1.5 metres above the deck.
4) Side Forces from a Slow Manoeuvre: For a trailer, assume a centre of rotation at the centre of the four rear wheels and a maximum coefficient of friction of 1.0 and apply a side force 150 mm below the twist axis of the chassis. Calculate the vertical forces from the suspension layout.

According to Sharman the maximum stresses should be calculated by combining the effects of one and four together and then combining the effects of two and three together.

Note that it is necessary to consider the stiffness of the vehicle suspension as well as the torsional stiffness of the chassis when analysing the effects of a twisted ground plane. Obviously if the suspension is very soft in comparison with the torsional stiffness of the chassis most of the deflection will be taken up by the suspension and there will be little stress in the chassis. When a modification is being carried out on a chassis any changes which tend to reduce the twist per unit length in this mode will reduce the stress in the chassis. For example an increase in wheel base will have this effect.

Since torsion in cases 1 and 3 is largely resisted by bending in the longitudinals the checks that were carried out in 5.1 for vertical bending should ensure that modifications should be adequate in this mode of torsion.

The simplifications described in reference to figures 5.8 and 5.9 will enable the designer to get an estimate of the stresses in the chassis. However it should be noted that the true stress distribution in these situations is very complex. This is especially so because of the effects of warping at the ends of the crossmembers. It is possible to use techniques such as finite element analysis to calculate stresses due to torsion more accurately.

5.5 LATERAL BENDING

The greatest lateral bending stresses occur as a result of severe manoeuvring as shown in figures 3.1 and 3.2. As has already been stated in 5.4 side forces from a slow manoeuvre produce torsion in the chassis. These same forces produce lateral bending and some of the same assumptions can be made; that for a trailer the centre of rotation is at the centre of the four rear wheels and a maximum coefficient of friction of 1.0 can be used between the tyres and the ground.

Looking at the chassis in plan the ladder frame is basically the same as the type of girder known as a Vierendeel Girder. Simple bending calculations cannot be used to calculate the stresses in this mode. This is a statically indeterminate structure but a simple static analysis can be adopted.

In general the statically determinate analysis is not suitable for Vierendeels with:

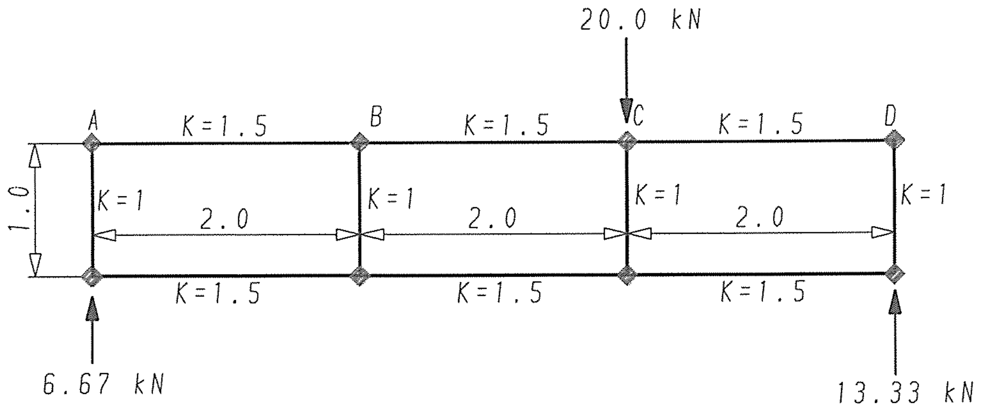
(a) inclined members

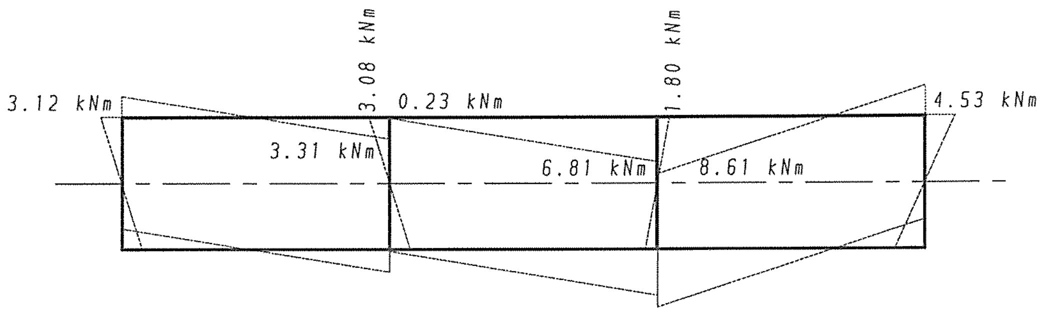
(b) longitudinal members with radically varying stiffness

(c) crossmembers of variable depth or with loads applied away from the node points (connections).

An example of this analysis technique is given with figure 5.12.

Figure 5.12 Plan of Simple Chassis





FINAL BENDING MOMENTS

THREE BAY VIERENDEEL GIRDER

	Cross-members				Longitudinals		
	A	B	C	D	A-B	B-C	C-D
Length	1.00	1.00	1.00	1.00	2.00	2.00	2.00
Stiffness	1.0	1.0	1.0	1.0	1.5	1.5	1.5
Vertical Force	-6.67	0.00	20.00	-13.33			

	A-B	B-C	C-D
SHEAR	6.67 kN	6.67 kN	-13.33 kN
F.E.M.'s	$\frac{-6.67 \times 2.0}{2 \times 2}$	$\frac{-6.67 \times 2.0}{2 \times 2}$	$\frac{13.33 \times 2.0}{2 \times 2}$
	= -3.33 kNm	= -3.33 kNm	= 6.67 kNm

		A	A		B			C		D	D
1	DF	0.571	0.429	0.300	0.400	0.300	0.300	0.400	0.300	0.429	0.571
2	FEM		-3.33	-3.33		-3.33	-3.33		6.67	6.67	
3	Dist.	1.90	1.43	2.00	2.67	2.00	-1.00	-1.33	-1.00	-2.86	-3.81
4	CO		-2.00	-1.43		1.00	-2.00		2.86	1.00	
5	Dist.	1.14	0.86	0.13	0.17	0.13	-0.26	-0.34	-0.26	-0.43	-0.57
6	CO		-0.13	-0.86		0.26	-0.13		0.43	0.26	
7	Dist.	0.07	0.06	0.18	0.24	0.18	-0.09	-0.12	-0.09	-0.11	-0.15
8	Final Moments	3.12	-3.12	-3.31	3.08	0.23	-6.81	-1.80	8.61	4.53	-4.53

1) The stiffness (in the horizontal plane) and lengths of the members are used to calculate the distribution factors (D.F.).

2) Shear forces are then used to determine the fixed end moments (F.E.M.s) at the ends of the chords ie. the points on the longitudinals where they connect to the crossmembers. This is equal to the shear force in the longitudinal times half the length between the crossmembers.

3) The distribution factors are then multiplied by the sum of the F.E.M.s at the joint to give the distribution (Dist.).

4) The carry over (C.O.) is then taken as minus the value of the distribution at the opposite end of the chord.

5) The distribution is again worked out, this time using this C.O. moment.

6) The C.O. for the above distribution is taken as the minus value of this distribution at the opposite end of the chord.

7) The third distribution is calculated by multiplying the D.F. by the above C.O. moment.

8) The final moment is now taken as the sum of the moments calculated above.

Shear in the crossmembers is obtained by dividing the moment at the joints by half their length. The axial force in the longitudinals is obtained by summing the shears in the crossmembers. The maximum stresses can now be calculated from the bending moments, axial forces and the section properties of the chassis members.

The above method of analysis is taken from "The Steel Designers Manual" [8] where the Vierendeel girder has rigid welded connections between all the members of the strucure. In practice a heavy vehicle chassis does not have such rigid connections between the crossmembers and the side rails. So it can be seen that the crossmembers will not in fact carry as much bending moment as the analysis indicates. If it is assumed that these connections carry a negligible moment then the side rails would provide all the lateral bending stiffness of the chassis and a simple bending calculation could be made. Half the bending moment could be assumed to be taken in each of the two side rails. The real situation must be somewhere between these two extremes.

If it can be shown that the maximum stresses due to lateral bending moment in the modified chassis are less than those of the original chassis then the modifications should be acceptable.

If the stresses are higher then it should be sufficient to ensure that the yield strength is not exceeded in the longitudinal members of the modified chassis.

5.6 AXIAL BENDING

As has already been stated the main cause of axial bending (or parallelograming) is uneven braking forces between the left and right hand sides of the vehicle. If for example the left hand wheels of the vehicle were on wet grass and the right hand wheels were on dry tar seal the respective coefficients of friction could be as low as 0.2 on the grass and as high as 0.8 on the tar seal. This will produce a difference in the force into the left hand side of the chassis versus the right hand side and thus result in a bending moment which must be resisted by the connections between the crossmembers and the siderails. It is unlikely that the stresses due to axial bending are especially high but it is important to realise that these forces exist when considering the connections between the crossmembers and the siderails.

6. FATIGUE

It is not sufficient to design a chassis using the yield or ultimate strength of the chassis material or it's stiffness as the only criteria.

In service a heavy vehicle chassis might never be subjected to the stresses or deflections close to those described in the previous sections but if the effects of fatigue were not considered in the original design, details such as welds or holes could easily result in a cracked or broken chassis. It is particularly important to consider these effects in a modified chassis because it is so common to cut and weld sections and drill extra fixing holes in the chassis members.

It should be noted that wherever welding is used the ultimate strength or the yield strength of the parent material does not give any indication of the weld's fatigue strength. In most instances it is not an advantage to select a very high strength steel for a chassis because the fatigue strength of any welds will not be any higher than for a material with a lower ultimate strength . It is very important to understand that welding has a serious effect on the fatigue strength of the chassis. The permissible stress in fatigue is independent of the steel specification after welding.

6.1 FATIGUE LIFE

It is common in engineering to define what is known as an Endurance Limit for most ferrous metals. This is usually understood to mean the alternating stress which can be applied indefinitely to the metal without causing failure. In theory a steel chassis could be designed so that the endurance limit was never exceeded and it would have an infinite life. In practice this would usually mean the chassis would be too heavy.

Non ferrous metals such as aluminium do not display this property ie. there is no level of stress below which the metal will not eventually fail if cycled enough times. For these materials a figure is usually quoted which equates to the stress which will cause failure after a specific number of cycles, typically 500 million cycles. Similarly, welds in steel structures have no true Endurance Limit. The fatigue properties of a material are usually shown in the form of a graph plotting stress against the number of cycles to failure (see figure 6.1). This is known as an SN curve.

As can be seen, if the material is subjected to relatively high alternating stresses it will survive far fewer cycles than if it is subjected to relatively low alternating stresses.

Vehicle chassis have a finite life. It is not necessary to design the chassis to take an infinite number of cycles of any level of stress. It is sufficient to ensure that the chassis will only fail after it has survived all the likely load fluctuations and the number of cycles it will receive in it's expected lifetime.

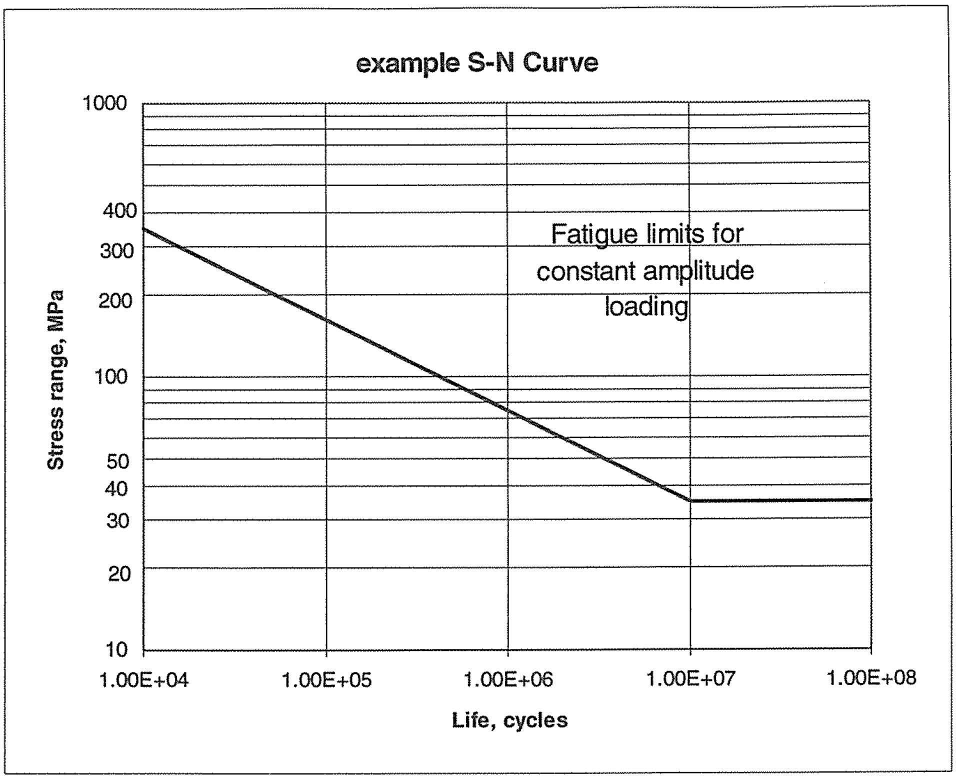


Figure 6.1

6.3 LOAD HISTORY

If an estimate can be made of the loads that a vehicle chassis is likely to be subjected to in it's lifetime, then it will be possible to design or check the design to ensure that it is adequate in terms of fatigue.

Welds are the principal initiators of fatigue cracks in vehicle chassis. Norman Simmons [9] has examined the cracking of welded structures and makes some estimates on the fatigue type loads on heavy vehicle structures.

The numbers he quotes are for European roads. They are only examples but they provide the basic parameters which could be modified for local conditions. The vehicle is assumed to cover 100,000 miles a year for ten years - a million miles life.

1) Vertical dynamic loads. Plus or minus 0.25g loads are relatively rare. So a total stress range of 0.5 of the static stress twice per mile ie. 2 million cycles is reasonable.

2) Normal cornering loads. On average sideways accelerations of 0.1g are typical. He assumes a fluctuation level of plus and minus 0.1g with a life of 2 million cycles.

3) Scrub turns (tight low speed manoeuvres with multi axle groups). Side drag forces above 0.8g are occasional. 100,000 cycles is assumed.

4) Braking and acceleration. Most of these give rise to longitudinal accelerations of less than 0.1g and are unlikley to more than once in two miles. A fluctuation level of plus and minus 0.1g and 500,000 cycles is assumed.

Studies done by the Australian Road Research Board [10] suggest that the vertical dynamic loads should be significantly different to the example given above. The ARRB have made many measurements of the dynamic wheel forces in axle group suspensions of heavy vehicles. These measurements have been concerned mainly with the effects of different two and three axle heavy vehicle suspension groups on road damage. However the results do seem applicable to the vertical dynamic loads on the chassis.

The principle characteristic which was evaluated is known as the Dynamic Load Coefficient (DLC). This is determined by measuring the vertical wheel force going into an axle travelling on a given road at a fixed speed over some period of time. A summation is then done of the amplitude of the forces versus the number of occurences. The distribution of these forces is found to be close to a random normal distribution, with a mean value equal to the static load on the wheel.

DLC is the standard deviation of wheel force distribution divided by the overall mean force:

$$DLC = \bar{s} = s / \bar{Z}$$

where

s	=	standard deviation of wheel force distribution (kN), and
$\bar{Z}$	=	overall mean wheel force (kN)

Using the DLC's measured from these heavy vehicles, and assuming that most of the significant wheel forces occur at frequencies approximately equal to the natural frequency (the bounce mode) of the suspension, it is possible to determine a reasonable distribution of load range and number of cycles.

An estimate was made using this technique, making the following assumptions;

1. a chassis life of 1.6 million km
2. two thirds of the distance is travelled with the vehicle fully laden
3. the distance travelled is spread equally between smooth, medium and rough roads
4. the DLC's for the suspension were 0.10, 0.15, and 0.20 for the smooth, medium and rough roads, respectively. These were typical values taken from the ARRB study, at speeds of 80 km/hr for the smooth roads, 70 km/hr for the medium roads and 60 km/hr for the rough roads. Note that there are wide variations in DLC depending on the suspension type.
5. the natural frequency for the suspension was 3.0 Hz. Softer suspensions will have lower natural frequencies for a given payload, and a proportionately lower number of cycles.

The results, using this approach can be summarised as, vertical dynamic loads of;

6.5 million cycles	at a stress range of	0.6 times the static stress
6 million cycles	at a stress range of	0.8 times the static stress
1 million cycles	at a stress range of	1.2 times the static stress
180,000 cycles	at a stress range of	1.6 times the static stress

In practice these stress ranges are probably very conservative (ie. these are maximums). The calculations assume that the forces are produced with the vehicle purely in bounce mode. In fact some of the forces produce pitching motions, which would generate lower stresses in the chassis. If we assume that 50% of the forces produce stresses that are only 75% of these maximums then a good approximation would be;

6 million cycles	at a stress range of	0.6 times the static stress
3.5 million cycles	at a stress range of	0.9 times the static stress*
600,000 cycles	at a stress range of	1.2 times the static stress
90,000 cycles	at a stress range of	1.6 times the static stress

* the effects of the stress range at 0.8 times the static stress have been incorporated into this range to make the list easier to use.

6.4 WELDED JOINTS

There are a number of standards which are available which can be used to check the design of welded steel connections. The British Standard BS 5400 part 10 or the Australian Standard AS 1250 both deal with the fatigue aspects of welding details in steel structures.

Table B1 from **AS 1250** categorises four loading conditions.

Loading condition	Number of loading cycles
1	From 20,000 to 100,000
2	From 100,000 to 500,000
3	From 500,000 to 2,000,000
4	Over 2,000,000

Also given is table B3 which describes Maximum Permissible Stress Range.

Maximum Permissible Stress Range

	Loading condition	Loading condition	Loading condition	Loading condition
Category	1	2	3	4
A	410	245	165	165
B	310	185	120	110
C	220	130	85	65
D	185	110	65	45
E	140	85	55	30
F	100	80	60	55

The maximum permissible stress range describes in megapascals the maximum range which is acceptable in a particular weld. The loading condition (1 to 4) is the same as described in table B1. The categories A to F describe different welding details. For example, complete penetration butt welds of parts of similar cross section ground flush, with grinding in the direction of applied stress is category B. This would be the best way of joining two sections of siderail with a weld. As can be seen 2 million cycles corresponds to loading condtion 3 and table B3 shows that the weld (B) has an allowable stress range of 120 Mpa. So for vertical dynamic loads a value of 0.5 times the static stress needs to produce less than 120 Mpa.

Note that one of the most common welding details; fillet welded connections with base metal at the junction of axially loaded members with fillet welded end connections, falls into category E. To survive over 2 million cycles the maximum allowable stress range is a mere 30 Mpa.

BS 5400 part 10 defines various welding details and splits them into categories B, C, D, E, F, F2, G, W and S. The SN curves for each of these categories is plotted on a graph. With this information it is possible, using Miner's Rule to determine a "life" for a particular weld.

For example, a category W weld (a fillet weld) is subjected to 2 million cycles at a stress range of 25 MPa and 500,000 cycles at a stress range of 53 MPa.

In BS5400 part 10 the SN curve (at mean minus two standard deviations) shows that the life at 25 MPa would be approximately 10 million cycles (see figure 6.2). The example weld uses 2 million, or 20% of these cycles. At 53 MPa the life is approximately 1 million cycles. The example weld uses 500,000, or 50% of these cycles.

Therefore the total number of cycles that the weld has been subjected to represents
20% + 50% = 70% of the life of the weld, which would be acceptable.

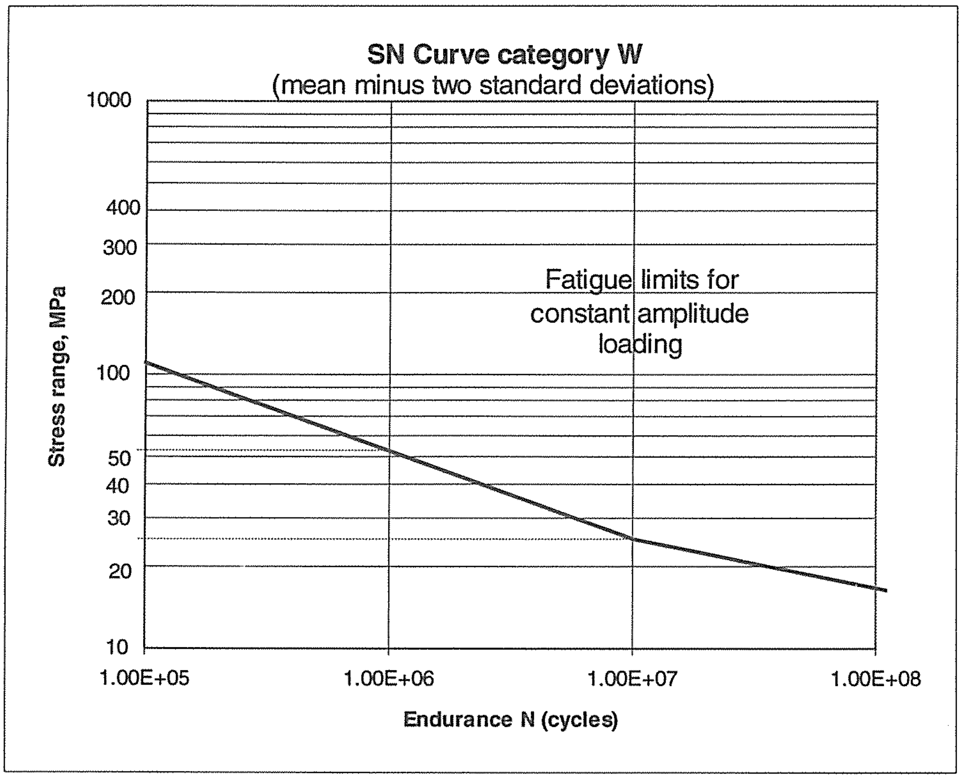


Figure 6.2

7. MANUFACTURER'S RECOMMENDATIONS

Most heavy vehicle manufacturers do not recommend the modification of their chassis but they will usually provide information on the correct ways of doing so - which is an implicit admission that modifications are sometimes necessary. Different manufacturers' advice on modifying chassis varies somewhat but the engineering principles behind their recommendations are the same. This section makes mention of the most common suggestions.

When cutting a chassis, the frame should not be cut in the region of

- points of load application

- control rod mounts and suspension mounts

- transmission mounts

- profile changes (frame offsets, tapering frame sections)

The material used in frame extensions must match the standard chassis frame in size and quality. Extensions or reductions in length are normally only carried out in the rear overhang of a chassis.

7.1 DRILLED HOLES

Holes produce a stress concentration in any stressed member. They are a likely starting point for yielding or fatigue cracking. It is common for manufacturers to require all holes to be finished after drilling and also to have chamfers on both faces.

In general holes should be kept away from the most highly stressed part of the frame. Normally holes should not be drilled in the upper and lower flanges of the side rails or crossmembers. An exception to this might be the extreme tail ends of the siderails where the stresses are normally very low.

For holes in the web it is not recommended for them be within 20% of the frame height from the upper and lower flanges or close to perpendicular bends of the side members.

Holes should also be spaced so they are pitched at least 4.5 times the diameter apart and they should be a minimum of aproximately two diameters away from any edge.

They should not be drilled close to welded sections of the same member.

7.2 WELDING

As has already been stated welds are the most serious initiators for fatigue cracks in vehicle structures. Welding is a technique which requires much specialised knowledge and skills which only come from the proper training and practical experience. It is beyond the scope of this paper to advise on detailed aspects of welding. The following comments only refer to the more general aspects with respect to vehicle chassis.

It is usually recommended that the welding material must show at least the same yield point and tensile strength as the steel to be welded.

Always weld from bottom to top. Preheating of the parent metal is recommended. In particular note that high strength steels, as are more and more commonly used in heavy vehicle chassis, often have specific preheat requirements. Most welding codes provide a method to determine the minimum preheating temperature considered as safe for particular grades and thicknesses of steels eg AS 1554.5 1989.

Welding onto the outside of the top or bottom flanges of the sidemembers is usually prohibited and welds are not recommended in, or close to edges or bending radii. In general they should not be used in areas of stress concentration.

Where butt welds are used to join or extend sections of chassis they must have their surfaces ground flush. The same applies when using weld to plug drill holes.

Welded chassis extensions or joins are always accompanied by extra channel or angle reinforcement and if welds are used to attach these to the sidemembers they should not finish close to the ends of the reinforcement (see figure 7.1).

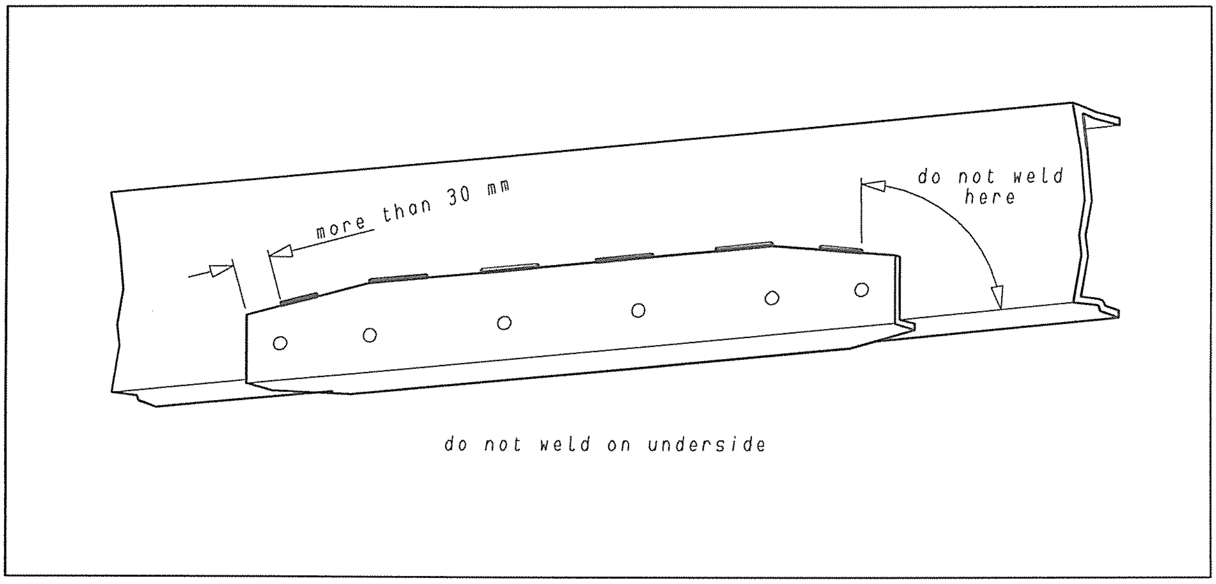


Figure 7.1

7.3 SUBFRAMES & REINFORCEMENT

Chassis reinforcements usually take the form of angle or channel bolted, riveted or welded to the inside of the original chassis rails. Subframes are usually channel sections which sit on the top of the original chassis running over much of the total length. As has already been stated the yield strength and the ultimate strength of the reinforcing members and subframes should at least be equal to the standard chassis.

It is most important to ensure a minimal stress concentration at the ends. This is achieved by tapering the ends of the reinforcement section or providing a cut out. The sharp edge in contact with the chassis rails can also be removed (see figure 7.2). The reinforcement or subframe should also end well away from other stress concentrations such as changes in section, the edges of crossmember connections, points of load application, suspension mounts etc.

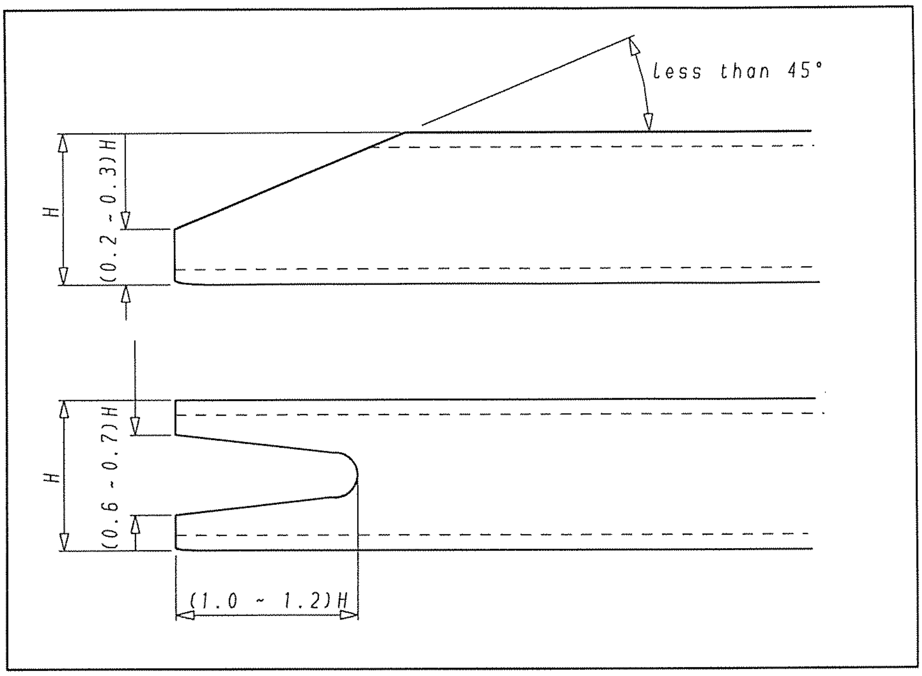


Figure 7.2

There are numerous acceptable methods of attaching subframes to chassis rails, all of them ensure that the subframe channel sits squarely on the chassis rail and that the web of the subframe is in line vertically with the web of the chassis rail. U-bolts can be used to clamp the two together. It is important to ensure that the U-bolt does not distort the flanges of either member.

Mounting brackets can also be used. These are bolted to the web of the subframe and the chassis rail (see figure 7.3).

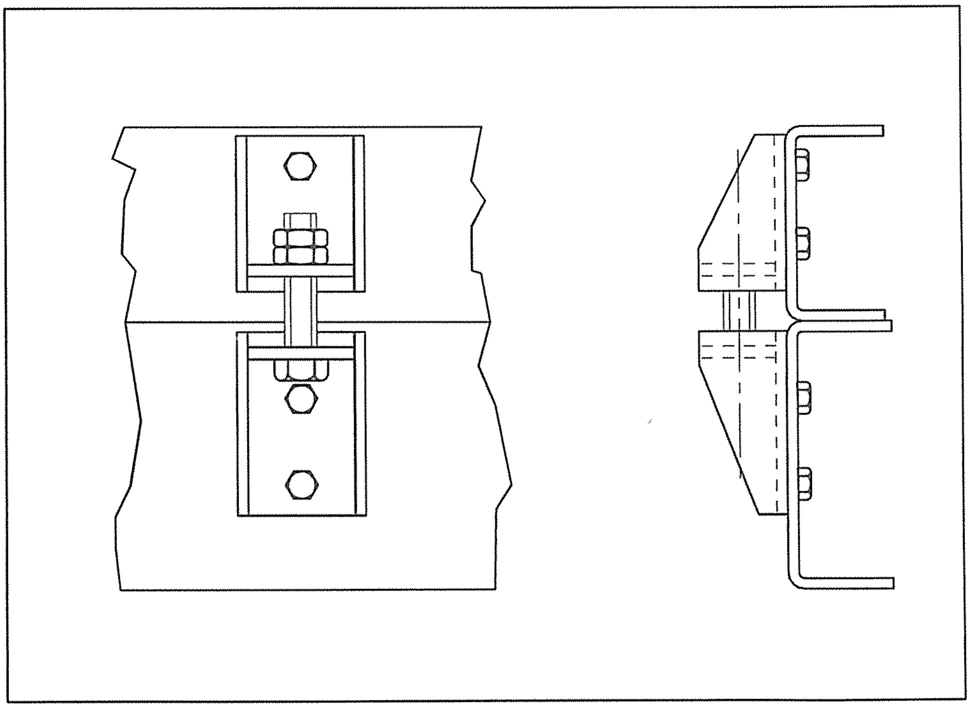


Figure 7.3

Locating plates or guide plates are sometimes used. These are flat plates which are welded or bolted to the web of the subframe and bolted to the web of the chassis rail (see figure 7.4).

They are not recommended when the chassis is subject to severe racking or twisting eg. vehicles with high centres of gravity or off-road type applications.

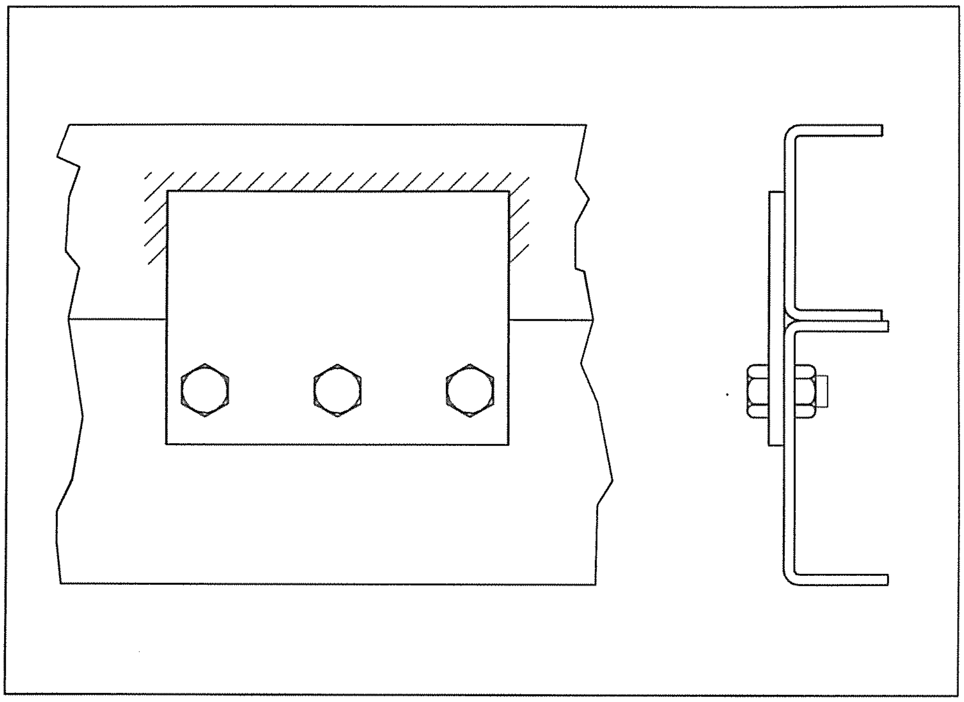


Figure 7.4

In all cases there must also be some mechanism to prevent relative longitudinal movement between subframe and chassis rail, such as angled mounting brackets or bolted guide plates (figure 7.5).

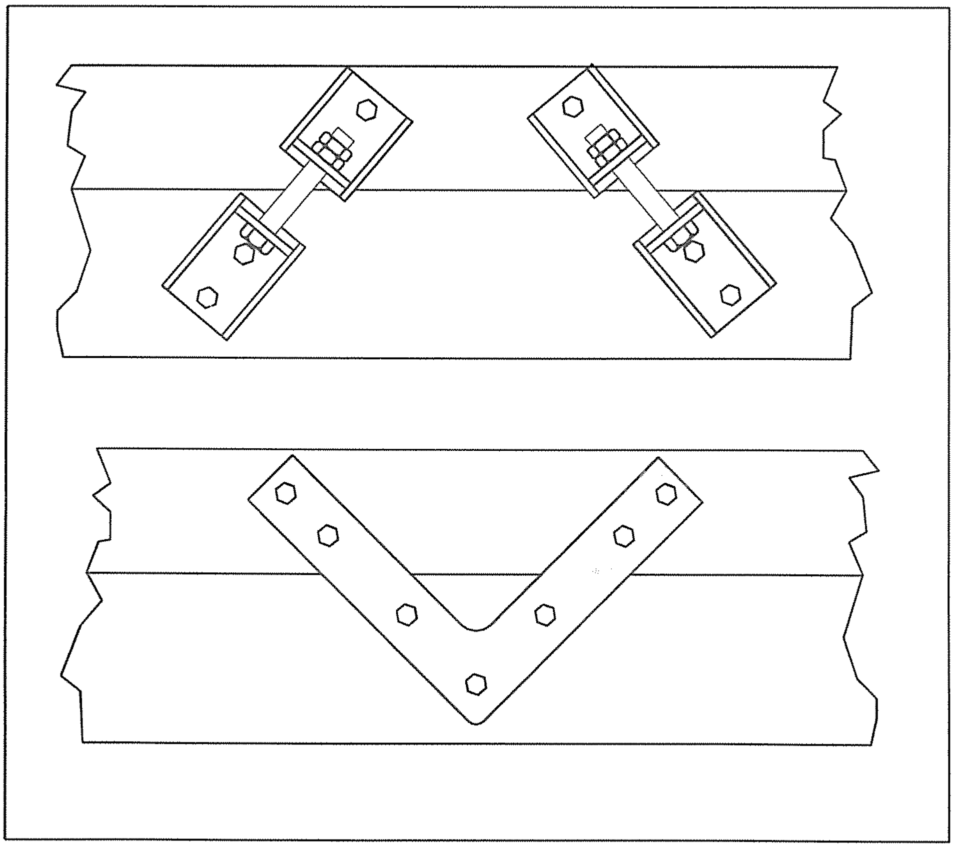


Figure 7.5

REFERENCES

1. Sidelko, William J., An Objective Approach to Highway Truck Frame Design, SAE paper SP276
2. Young, Warren C., Roark's Formulas for Stress & Strain sixth edition, Mc Graw Hill, 1989
3. Hsu, Teng H., Stress and Strain Data Handbook, Gulf Publishing, 1986
4. AISC Manual of Steel Construction
5. Trahair, N.S., and Bradford, M.A., Design of Steel Structures, Chapman Hall, 1988
6. Anderson, J.M. and Trahair, N.S., Stability of Monosymmetric Beams and Cantilevers, Journal of the STRUCTURAL DIVISION Proceedings of the American Society of Civil Engineers, January 1972
7. Sharman, P.W., Torsion of Chassis Frames, IMechE Automotive Engineer, October/November 1976
8. Steel Designers' Manual, Prepared for the Contsructional Steel Research and Development Organisation, Granada Technical Books
9. Simmons, Norman, Transport Engineer, January 1991
10. Sweatman, P.F., A Study of Dynamic Wheel Forces in Axle Group Suspensions of Heavy Vehicles, Special Report No. 27, 1983, Australian Road Research Board

INDUSTRIAL RESEARCH LIMITED

AUCKLAND
Brooke House
24 Balfour Road
Parnell
PO Box 2225
Auckland
New Zealand
Phone +64-9-303 4116
Fax +64-9-307 0618

PALMERSTON NORTH
Fitzherbert West
Private Bag
Palmerston North
New Zealand
Phone +64-6-351 7230
Fax +64-6-351 7239

WELLINGTON
Gracefield Research Centre
Gracefield Road
PO Box 31-310
Lower Hutt
New Zealand
Phone +64-4-569 0000
Fax +64-4-566 6004

CHRISTCHURCH
175 Roydvale Avenue
Bishopdale
PO Box 20-028
Christchurch
New Zealand
Phone +64-3-358 9189
Fax +64-3-358 9506