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ANALYSIS OF ARMOURED-VEHICLE TRACK LOADS AND STRESSES, WITH CONSIDERATIONS ON ALTERNATIVE TRACK MATERIALS

R.H. Keays, Keays Engineering & Computing Pty Ltd

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track of an armoured-vehicle in service conditions.   The analysis includes the
effects of normal traction, inertia forces from rotation of the individual
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The report looks in some detail at the tracks of the M113 vehicle, and
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1. INTRODUCTION

This report investigates the loads applied to track components on tracked vehicles. It summarizes the forces and their effects, with examples taken from the geometry of the M113 vehicle.

A tracked vehicle runs on road wheels along a track which is carried, laid on the ground, and then picked up by the vehicle. The track is continuous and is assembled from a series of links so it is flexible enough to wrap around and engage a sprocket wheel which transmits the power from the engine to pull the vehicle along the track. The track provides a large surface area to distribute the weight of the vehicle when travelling over soft ground. The leading slope of the track provides a ramp to negotiate obstacles and for traversing uneven ground.

Figure 1 shows the arrangement and basic dimensions of the M113 vehicle which is used as the example in these calculations. Figure 2 shows how the track components are assembled, and Figure 3 shows the detailed geometry of the track and wheels.

The tracks used for military and commercial purposes differ significantly. Commercial vehicles, such as bull-dozers, travel slowly, with few maneuvers on hard surfaces. They require maximum traction on soft or broken surfaces, so each link is provided with blades to key into the soft surface. Military vehicles travel faster (up to about 65 kph), turn frequently, and regularly travel on made roads. To protect the made roads from damage, the military vehicle is provided with rubber pads on the underside of each link. Traction in soft conditions is provided by grouser bars on the sides of each link. To achieve the higher maneuverability required, the military vehicle is provided with a flexible track. This flexibility allows one shoe to rotate a small amount in plan relative to its neighbour, improving the turning ability by an order of magnitude. The rubber bushes between the pins and the shoes give the track this flexibility.

The demands placed on the track by military operations are severe. Typical life expectancy for a track is about 9 000 km of service. For the M113, this represents about 1 000 000 track rotations, placing the design life in the region where fatigue is critical.

This report looks first at the various sources of loads on the track, with an estimate of magnitudes for the M113 vehicle. It then looks briefly at the stresses in the three components (pins, bushes, and shoes) of the existing track. Consideration is given to the replacement of the existing track shoes with ones made from an aluminium alloy/SiC metal matrix composite. This material has the advantage of lighter weight, and it is required that design goals for material properties be established.

2. TRACK LOADS

The track consists of a train of links (called shoes) connected by pins, as shown in Figure 2. At each pin there is a rubber bush between the pin and the boss on the shoe. This bush provides the majority of the tensile flexibility for the train.

The analysis of the loads in the track is approached by considering each of the possible sources in turn, starting with initial tension and simple traction. The internal loads from friction, inertia, and vibration are then considered. Having established the basics, it is then possible to go on to consider the complex loads resulting from uneven ground and turning maneuvers.

2.1 Initial Track Tension

The track is pretensioned onto the wheel system, to provide adequate track guiding under all operating conditions, and to ensure that the track meshes with the drive wheel at all speeds. The exact value of this pretension is probably not known with any degree of accuracy, as the track stretches and the rubber bushes wear with service. Track tension is adjusted by altering the position of the carrier wheel. This is done by the operator at regular intervals.

On the M113, the initial tension is set by adjusting the volume of grease in a ram, which has the effect of setting the position of the carrier wheel. (Some tracked vehicles have a hydraulic ram providing a constant load on the carrier wheel. Analysis of such vehicles requires a slightly different approach.) Adjustment is a simple process, with tension judged by checking the sag in the track over the top between the drive and carrier wheels.

The tension corresponding to this sag can be calculated by considering the track as a uniform cable spanning between two points. This is an approximation, as the cable consists of a series of links connected by rubber bushes with a certain amount of torsional stiffness. (An estimate has been made of the effect of this, indicating an overestimate of the sag of about 10%.) For a cable tied between two points on the same level, as shown in Figure 4, the inelastic sag is given by:-

$$f = \frac{m.g.L^2}{8.H} \qquad (1)$$

where
f = sag
m = mass per unit length
g = gravity

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L = length between supports
H = horizontal load applied at the ends
     (close enough to the cable tension in most cases)

The length of the cable between the two points is given by:-

$$s = L.(1 + (8.f^2) / (3.L^2)) \tag{2}$$

Note that the elastic extension of the cable doesn't appear in these expressions. In applications where the cable is fixed at its ends (such as suspension bridges), further terms must be added to account for the elastic extension. For the track, the initial extension is taken up by adjusting the carrier wheel. In operation, when the tension is higher, the extension is taken up by additional sag between between the drive wheel and the forward road wheels.

For the M113, the dimensions and parameters are as follows:-

m (unit weight)   9.68 kg/0.152 m = 63.7 kg/m
L (across top)   4 m (approximately)

Substituting these values in Equations (1) and (2), gives the values tabulated below:

Tension (H – kN)	Sag (f – mm)	Extra Length (s–L – mm)
5	250	42
10	125	10
15	83	5
20	62	3
30	42	1
40	31	1
50	25	–
60	21	–

At loads below about 10 kN, the sag is such that the track will rest on the top of the intermediate road wheels, and so not form the full catenary.

With the M113, the procedure for setting the initial track tension is to allow the vehicle to roll to a stop in neutral, thus minimizing any differences in the tension. The sag is then checked by ensuring that the track is resting on the top of the middle road wheel, with a gap of about 1/4" at the second wheel from the front. This corresponds to about Owen (1) 12 kN tension suggests a figure of 8% to 10% of the vehicle weight (10-12 kN for the M113) as typical for tracked vehicles. Given the approximations in this analysis, it will be assumed that the initial track tension is 12 kN.

2.2 Traction Loads

The track is driven by a sprocket wheel (the drive wheel) located forward of the road wheels (on the M113), as shown in Figure 3. A carrier wheel is provided behind the last road wheel. This serves to guide the track up to the level of the drive wheel, and provides

an arrangement suitable for operation in reverse.   Further small carrier wheels may be provided to support the weight of the track in its travel forward to the drive wheel, but these are not used in the M113.

The weight of the vehicle is supported by the road wheels.   The distribution of the weight depends on the centre of gravity position.   The distribution will change as the vehicle is turning, producing greater loads on the outside track.   If the vehicle is accelerating or decelerating, the fore and aft distribution will be affected.   The tension in the track, acting with the suspension springs of the road wheels, will tend to lift the extreme road wheels, distributing a greater proportion of the total weight to the inner wheels.

The traction transferred at each wheel depends on the vertical load at that wheel and the coefficient of friction developed.   The exact distribution may be important to the detailed analysis of turning performance, but is not greatly significant in straight line motion.   In this case, the tension in the track increases as each wheel passes.   So, for the purpose of this analysis, the effect of the several road wheels can be replaced by a single action.

Figure 5 shows the loads and resulting tensions when a tractive load is applied at the drive wheel.   Ignoring losses at the wheel bearings, the total traction is given by:-

$$T_r = \Sigma F = T1 - T7 \tag{3}$$

When the applied tractive load is small, the effect is to increase T1 and decrease T7.   Analysis of the relative stiffness of the two parts of the track indicates that the majority of the load is absorbed by reducing T7.   The linked chain between the drive wheel and the front road wheel cannot carry a compressive load - it would simply fold out of the line of action.   Thus once T7 reaches zero, any increase in the tractive load is carried by an increase in T1, which is then equal to the applied traction.

Setting the initial tension high enough to prevent T7 reaching zero is not necessary.   The track will still follow the guides because its lateral stiffness is sufficient at the low speeds which are coincident with maximum traction.   Hence the design maximum traction force (without allowance for initial tension) is the basis for the design tension in the track.

The limit on the tractive force is reached either when the motor driving the sprocket reaches its maximum torque, or when the complete track skids across the ground. In the latter case, the traction developed at each wheel will be the weight applied at that wheel multiplied by the coefficient of friction at slip.   Assuming that the coefficient of friction at the several wheels is similar, the summation of forces means that the limiting traction is a function of the total weight on the track multiplied by the average coefficient of friction.

Tests with actual vehicles on bitumen and concrete surfaces (Ref. 2) gave overall Owen (1) friction coefficients in the range 0.75 to 0.94, with an average of 0.84 suggests 0.8 as a good working value.) These tests were carried out with a vehicle mass of about 75% of the combat mass of the M113.   If the motor could provide sufficient torque to slip the wheels at the maximum (combat) mass of 12 200 kg, the resulting differential tension at the driving wheel on each side would be 12 200 x 9.81 x 0.94 /2 = 56 kN per side (at maximum friction coefficient).

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Engineering Development Establishment (EDE) have advised that the maximum draw-bar pull of the M113 is 110 kN, giving a traction force of 55 kN per side. This value will be used in the following analysis.

2.3 Internal Friction Losses

Internal friction losses reduce the proportion of the total applied traction force that actually goes into propelling the vehicle forward. These losses occur at each wheel, and to a lesser extent at the pins and rubber bushes joining the shoes.

As the design traction force has been determined from testing the draw bar force for the vehicle, the friction losses should be added to the calculated equivalent tension in the track, as these must exist and add to the required applied force at the drive wheel.

A brief assessment of the drive system indicates friction forces even on good surfaces will be considerable, possibly of the order of 5% of the vehicle weight. The prime source of friction appears to be rolling friction. Simply put, rolling friction occurs because each road wheel is constantly trying to climb out of a small depression created by elastic deflection of the tyre and road surface. The spring-back of the road and wheel behind the point of contact assists the wheel in moving forward, but this is not 100% efficient, leading to a net retardation force on the wheel.

When travelling in less than perfect conditions such as loose soil and mud, friction forces within the track system will increase considerably. However, the traction achieved will be reduced because the loose soil will provide less resistance to slipping, with the nett effect that the sum of friction plus traction would be unlikely to exceed the hard surface value.

Accurate calculation of the actual rolling friction would be difficult. Hence for the purposes of this initial study, a figure of 5% (on hard ground) will be adopted. There will also be losses at the rear idler wheel, the drive wheel, and in the rubber bushes, but these should not be as large.

Adding this 5% to the peak draw-bar force gives a maximum tension at the drive wheel of 58 kN.

2.4 Inertia Loads

Each track shoe is subjected to significant inertia forces as it circulates. As it passes under the rear road wheel it is picked up from the ground, raised about 800 mm, and propelled forward at twice the vehicle speed. It is then thrown back down on the ground again, where it stops just as the forward road wheel rolls over it. Careful analysis of the inertia forces is required in this instance, because consideration is being given to replacing the existing steel shoes by lighter aluminium alloy metal matrix composite shoes. The lighter shoes mean lower inertia forces, and may result in a reduction in the track pretension required.

Figure 6 shows how the motion of the track can be divided into two components for analysis. The first component is the forward motion of the complete vehicle, whilst the second can be likened to a continuous belt moving round a series of pulleys.

The first component can be dealt with quickly and simply. If the vehicle is moving forward at constant speed there will be no inertia forces from this component as the acceleration is zero. If the vehicle is accelerating or decelerating, there will be a force on each component equal to the element mass by the body acceleration. The peak acceleration is achieved when the tracks slip. However this force is simply a reaction to the applied traction - no additional force is introduced into the system.

(There is a possibility that the snatch load on the track (when the driver "drops the clutch") may be of more significance. However, as the M113 has an automatic transmission, complete with a torque converter, and a drive train with considerable inertia, it is unlikely that this action will result in a load at the drive sprocket that exceeds the load derived for other conditions.)

The second component of the motion in Figure 6 can be analysed in the manner applicable to a continuous belt round a pulley. Figure 7 shows the derivation of the inertia component of belt tension, without considering the traction component. The result of this analysis is that the tension due to inertia is constant round the full length of the belt, and equals the mass per unit length times the velocity squared. Interestingly, it does not depend on the radius of the pulley.

The track on the M113 has a mass of 9.68 kg per shoe. The spacing of shoe pins is 152 mm giving a unit mass of 63.7 kg/m. The top speed of the vehicle is 40 mph (65 k/h - 18 m/s). The track travels round the wheels at this speed. So the peak inertia tension is $63.7 \times 18^2 = 20.6$ kN.

This inertia tension interacts with the initial belt tension. Consider a belt delivering no power. When the belt is at rest, the initial tension results in a pressure on the pulley wheel faces. As the belt is brought up to speed, the belt remains in contact with the wheels. The centripetal force relieves the radial pressure on the wheel surface. Once the belt speed reaches the level where the inertia tension and initial tension are the same, the pressure on the surface reduces to zero. Increasing the belt speed past this point results in the belt moving out from the wheels, with a tension which must be equal to the inertia tension, as there is no other source of load in the system.

With a simple belt drive, this would result in a loss of traction, and the belt would slow down to a speed where it remained in contact. With a sprocket drive, such as in the vehicle tracks, the drive remains in contact, although over a reduced arc and a catenary forms between wheels. Ultimately the stretch of the belt under the centripetal force will be such that the sprockets no longer engage.

Whether the initial tension should be added to the centripetal tension depends on the system used to set the initial tension. If the initial tension is set by applying a force, such as might be done with a hydraulic actuator supplied with a constant pressure, then the carrier wheel will exert a constant pressure on the surface of the track, resulting in the addition of the two loads. On the M113, adjustment of the initial tension is achieved by altering the volume of grease in a cylinder, effectively applying a fixed displacement, rather than a constant force. In this case, when the inertia tension exceeds the initial tension, the track will move away from the wheel. As the track tension is equal to the initial tension at zero speed, and at the speed at which the track moves away from the wheel, it is then reasonable to assume that it remains at that value through that whole speed range.

The question now arises as to whether this tension should be simply added to the maximum traction force. Typically, an engine and transmission will not produce its maximum output torque at its maximum speed (otherwise it could continue to accelerate). Whilst engine torque on the M113 only drops by 10% between minimum and maximum operating speeds, the available torque at the drive wheel is much reduced, because the vehicle will be in top gear. As the ratio of top to low gear is of the order of 1:4, only a quarter the torque is available.

For the M113, assuming an initial tension of 12 kN (10% of vehicle weight), initial tension will determine forces up to a speed of $(12000/63.7)^{.5} = 13.7$ m/s. At this speed the vehicle would be in an intermediate gear giving about a third of the peak traction (say 20 kN), giving a peak tension of $12 + 20 = 32$ kN. At top speed in top gear, the inertia tension will be 20.6 kN, combined with a quarter of the peak traction (14.5 kN), giving a peak tension of 35.1 kN. From this it can be n that inertia effects combined with traction will not produce a governing load case.

The vehicle's brakes can be operated at any speed, so it is necessary to consider inertia effects in combination with maximum braking. EDE have reported results of deceleration tests carried out with an M113. Results of tests are shown in the following Table.

All–Up Weight (tonnes)	Initial Speed (k/h)	Stopping Distance (m)	Average Deceleration (m/s^2)
10.8	16	3.5	2.8
10.8	32	10.0	4.0
12.8	16	3.5	2.8
12.8	32	10.5	3.8

From these results it would be reasonable to assume a peak braking deceleration of 5.0 m^2, giving a resulting track tension of 30.5 kN at combat weight. Adding this to the 20.6 kN inertia tension gives a total of 51.1 kN – a little less than the peak tractive tension.

It can be concluded from the above analysis that inertia forces, while significant, do not contribute to the governing design cases for the track tension.

2.5 Vibration

The loads from vibration can be considerable. Uncontrolled vibration will absorb energy from the drive system, and increase the noise level, as well as reducing the service life of the track. So it is something to be avoided where possible.

Vibration will be a problem when the frequency of a forcing function is close to one of the natural frequencies of the track system. The forcing functions for the track include:-

(a)   out of balance loads from the road wheels, carrier wheel, and drive wheel (at a frequency corresponding to the revolutions per second of the different wheels),

(b)   impact loading as each successive track shoe is loaded by the sprockets on the drive wheel (at the tooth-passing frequency),

(c)   out of bala..ce loads from irregularities in the track (at the track revolution frequency),

(d)   road surface corrugations (these tend to form at the natural frequency of the suspension of the vehicle/speed combination most commonly using the road), and

(e)   road surface irregularities (the impact from these can feed energy into a coupling between two natural modes of vibration, if these are similar).

With a tracked system, there are two important types of vibration. The first of these is the vertical motion of the wheel suspension system. The natural frequency of this is usually set to between 1 and 2 cycles per second, to give a reasonably comfortable ride over bumps and isolated obstructions.

The other important type of vibration is the lateral motion of the track between points of support. This acts as a catenary, as previously described when considering the setting of initial tension.

The first mode natural period of vibration of a catenary is:-

$$f_1 = \frac{C}{2.PI.L} \tag{4}$$

where $C = (H/m)^{.5}$ = velocity of propagation of waves, and H, m, L are as previously described (Fig.4).

If the cable is moving (as is the case with a chain or the vehicle track), the natural frequency changes to:-

$$f_1 = \frac{C}{2.PI.L} (1 - (V/C)^2) \tag{5}$$

where V is the track velocity relative to the drive wheel (equal to vehicle velocity).

If the velocity of the track could reach the velocity of propagation, the natural frequency would reduce to zero, resulting in a smooth, vibration-free travel. However, as the speed increases, inertia tension plays a part, placing a lower limit on the value of C, such that V can never reach C.

There are catenaries in the track between each successive wheel, but the only important one is that over the top between the carrier and drive wheels. This has a length of about 4 m. Application of Equations (4) and (5) gives natural frequencies in the range 0.5 cps to 1.2 cps for the M113.

These are just the first mode frequencies. With catenaries, it is easy to develop the second and third modes, and these could also be excited by the forcing functions. As the forcing functions can have a wide range of frequencies, it is likely that vibration will be induced at particular speeds and surface conditions. As with out-of-balance vibrations on

cars, the driver will sense the vibration and adjust his speed to avoid the problem areas, or to accelerate through them quickly.

The forcing functions described at the beginning of this section are generally overall functions, producing global effects. One, however, can have local effects, and requires some further study. This is the impact of the drive wheel teeth on the shoes as they pass over the drive wheel.

The track shoes are connected together as a chain, and so consideration must be given to the basic action of a chain on a sprocket wheel. Because the links of a chain pass around the sprocket as a series of chords - instead of a continuous arc - the kinematic action of the chain drive differs from that of a belt drive. The chordal action of the chain is illustrated in Figure 8.

When the input shaft, joined to the driving sprocket, rotates with constant velocity, the height of the approaching chain varies from a minimum to a maximum value for each chord-engagement period, as shown in Figure 8. Expressed in terms of the sprocket dimensions shown in Figure 8, the variation in the height of the chain centreline ranges from

$$r_{min} = r.cos\ l(180/N)$$

to
$$r_{max} = r$$

where
r = pitch radius in metres
N = no. of teeth on driving sprocket.

With the 10-tooth sprocket on the M113, and 6" pitch for the track links, the pitch radius is 242 mm, and the minimum radius is 230 mm. The 12 mm difference is a source of vibration and stresses. This motion is a cycloidal one, with a cusp as each successive pin intercepts the drive wheel. At the peak speed of 18 m/s, the sprockets strike the track every 0.008 seconds. The sprocket lifts the track as it passes, but it does not fall back down immediately. Gravity, with some assistance from the track tension, dictates the rate of descent of the track. Under gravity, it takes 0.05 seconds to fall 12 mm. Thus before the track can fall appreciably, the next sprocket is forcing it up again, avoiding the high impact loads associated with the cusp on the cycloidal motion. Further analysis of this effect is not warranted at this stage. Analysis of the bearing of the sprocket on the track shoe will be considered in detail later.

2.6 Loads from Uneven Ground Conditions

The design case for uneven ground conditions reduces to the simple case of a sharp edged obstacle. Typically this will be a normal concrete kerb as the vehicle moves from a made road into open country. Less frequently, the vehicle will have to cross man-made barriers such as spikes and steel plates.

As the vehicle traverses such objects, it will come to a point of balance, where the entire weight of one side of the vehicle is carried by the sharp edges object. Figures 9 and 10 show two possible geometries for the M113. Figure 9 shows a single shoe spanning between two road wheels. This becomes the design case for bending of the shoe between its ends. The vertical force, W, is reacted by the road wheels, and there is no additional tension in the track. Figure 10 shows the pin boss wedged between two road wheels. Once again, no additional tension results.

The case is feasible if the driver can apply sufficient force to turn the vehicle through a minimum radius turn at road speed. If he does this, and mounts a concrete curb in the process, the outside track will carry the full vehicle weight. In practice, the track will probably destroy the curb.

A similar result occurs if the vehicle is traversing a slope. The grade of the slope that can be traversed is limited by the ratio of centre of gravity height to track width, as shown on Figure 12. This case is less severe than the turning case, as the vehicle mass is not supplemented by any centripetal effects.

As the track tension was dictated by the skid load with half the weight applied to each side, doubling the weight on one side might increase the tension. However, in the M113 the transmission includes a differential. Locking the inside track to go round a corner results in the outside track going twice as fast for the same engine revolutions. This is akin to changing to a higher gear, resulting in only half the torque at the drive wheel. Thus if the engine has only just sufficient torque to skid the fully loaded vehicle on level ground, the torque supplied to one side will not exceed half this, and the design limit is not changed.

The other design case was the bending load on the track shoe resulting from negotiating a sharp object. Here the complete weight of the vehicle is taken by the sharp object, as shown in Figure 13. Note the lateral load in this instance. The track shoe must be able to transfer this force to the adjacent road wheels, via the guide horns.

A further case arises from this. During steering and sliding, the side of the track may strike an obstacle which will apply a concentrated lateral load at a point between the road wheels. The magnitude of this load is the same as that in Figure 13, as shown in Figure 14. In this instance the track between the wheels will be lifted by the obstacle until it is in contact with the road wheels. (The geometry is similar to Figures 9 and 10.) Thus it reduces to the case shown at Figure 13.

2.8 Design Loads

Each of the loads considered above can be expected to occur in service, although the shoe bending cases would not be expected to occur often, as these require the combination of a turn at maximum speed coincident with striking a sharp-edged object.

As with all structural design, a factor of safety must be applied. Owen (1) suggests 2.5 on material yield. This factor takes into account material variability, tolerances, and inaccuracies in load estimates. It is recommended that this factor of 2.5 be applied to the basic case of track tension.

For the combined cases of turning and sharp objects, a reduction to two thirds this (1.67) is recommended. The reduced factor is applicable, as this case requires the simultaneous occurrence of two low-probability events.

The ultimate load cases can then be summarized as:-

1.   Track Tension alone (used in design of pins, rubber bushes, track shoe bosses, and drive wheel sprockets), with a factor of safety of 2.5:-

$$T = 58 \text{ kN (working) or } 145 \text{ kN (ultimate)}$$

If the wheels were not there to support the ends of the shoes, considerable tensions (possibly in excess of the traction tension) would be developed. To achieve these geometries, the track and suspension system must have sufficient flexibility at the design tension to deform around the obstacle.

The primary source of flexibility in the track is the road wheel suspension system, which retracts under the combined effects of vehicle weight and track tension, providing the additional length needed to achieve the desired geometry. Exact details of the suspension system were not available, but estimates indicate a total of about 150mm of "extra" track would be achieved in these situations. In addition to this, elastic deflection of the track rubber bushes and reduction of catenary sag each contribute about 10 mm additional length of track available.

The arrangement shown in Figure 9 requires 93 mm of "extra" track, which makes it easily accommodated. Figure 10, however, needed 185 mm of "extra" track, 15 mm in excess of the total available. In this instance, the vertical components of the track tension, T, on either side of the object react the applied load, W, in the relationship:-

$$W = 2 . T . \sin(A) \tag{6}$$

At 55°, a tension of only 36 kN would be required to react the weight of one side of the vehicle (60 kN). With a track tension of 58 kN, the required angle of the tracks would be 31°, requiring much less "extra" track.

The above indicates that the flexibility of the track suspension system plays an important part in ensuring that the track tensions induced by crossing sharp objects are not excessive. It has been estimated that the M113 suspension has been designed to ensure that these tensions do not exceed the primary traction tensions.

2.7 Turning Maneuver Loads

The tracked vehicle negotiates corners by skidding the tracks across the ground. Significant forces are generated under the road wheels, but between them the ground pressure is much less and so the force required to skid these elements reduced.

The forces generated at each wheel are limited by the local skid force which is proportional to the wheel load and the friction coefficient.

If the vehicle is negotiating a sharp object whilst attempting to turn, this sharp object will become the centre of the pivot. With the critical shoe fixed in contact with the road wheels (as shown on Figure 9), the tension in the track is not affected, although torsional loads will be introduced into that pad.

If the vehicle is moving at speed, the centripetal force moves the effective centre of gravity towards the track on the outside of the curve. At the limit, the combined gravity and centripetal force vectors will pass through the outside track. Beyond this, the vehicle is unstable and rolls over. Figure 11 illustrates this. This places the full weight of the vehicle on the outside track, in addition to the horizontal load from the centripetal effect. Combining this with a sharp object under the outside track produces a most severe case. This might be considered as the ultimate load case.

2.   Longitudinal and transverse bending of track shoes, combined with track tension, with a factor of safety of 1.67:-

	T	=	58 kN (working)	or	97 kN (ultimate)
	W	=	120 kN	or	200 kN
and	H	=	134 kN	or	224 kN

The arrangement and point of application of these loads is shown in Figure 15.

The above load combinations cover the ultimate strength when new. Consideration must also be given to the effects of fatigue and wear on the track. Most of the time, loads will be considerably less than the design loads. For wear at the sprockets and fatigue of the pins and shoes, a load marginally in excess of the initial tension should be considered.

For the fatigue of the pins and shoes, the load would cycle from the peak back to almost zero (as it leaves the front of the drive wheel) with each revolution of the track. With 63 shoes each 152 mm long, the track length is 9.576 m. With a design life of 9 000 km, this represents about 10^6 cycles.

For fatigue testing of pins, rubber bushes, track shoe bosses, and drive wheel sprockets, a tension load cycling from zero to somewhere in the range of 15 to 20 kN would be appropriate.

For fatigue testing track transverse and longitudinal bending, the transverse load, W, would be much less. In normal operation, the weight of the vehicle is divided between the 10 road wheels. Further it is unlikely that the track will strike a sharp object on each revolution. Thus not only is the load reduced by a factor of 10, the number of cycles is also reduced. As fatigue strength is rarely less than 10% of the yield strength, it would be reasonable not to conduct fatigue testing for transverse or longitudinal bending, and rely on the ultimate load tests for proof of safe operation in service.

Reference 1 suggests different load cases, in general less severe than the above. It assumes a maximum friction coefficient of 0.8, giving a track tension of 48 kN. It also distributes the vertical and lateral forces between a number of wheels, reducing these loads to about 40% of the above.

3. STRESS ANALYSIS

To provide a basis for rational assessment of the alternative material, a preliminary stress analysis of the existing M113 track has been carried out. This analysis includes checks of the rubber bushes and pins, primarily to assess whether the calculated loads are of the correct magnitude.

3.1 Rubber Bushes

The arrangement of the track at each pin is shown in Figure 2. The track pin is an octagonal bar. The rubber bushes are stretched onto a cylindrical steel former with an

octagonal bore. On assembly the rubber bushes are compressed into the boss on the track shoe. Torsional movement of the rubber is resisted by the friction forces developed during stretching and compression.

As the shoe passes round the various wheels, the rubber is deformed in torsion. To minimize the rotation of the rubber, the shoes are pre-set at about half the maximum angle before inserting the rubber bush. Shear stresses due to this torsion are significant, but not calculated here as the torsion is a function of wheel geometry, not track tension.

Figure 16 shows the dimensions and calculations for the stress analysis of the rubber bushes. This is a simple analysis, as is normally adopted for such components. Lindley (3) provided the physical properties assumed in the analysis.

The bearing stress on the rubber bush at the inner pin is of the order of 16 MPa, which is about as much as 70 shore rubber could be expected to sustain as a working load.

The radial stiffness of the rubber bush is in the direction corresponding to the axial stiffness of the track assembly. The calculated figure is about 0.14 mm deflection per link (at 58 kN load).

When rubber is subjected to cyclic loading, there is a loss of energy through internal hysteresis. This energy is dissipated as heat. Fawcett and Robertson (4) reported tests on Scorpion tracks in a stationary rig. It reported a rise in temperature of only 4°C after one hour of operation. This test was carried out without developing traction, so the track tension was low. However, it indicates that the temperature rise in the rubber is not a major consideration.

3.2 Pins

Figure 17 shows the dimensions assumed for the preliminary analysis of the pins. The analysis is approximate, to give an indication of the order of magnitude of the stresses.

The pins act as beams on an elastic foundation, with the foundation stiffness corresponding to the radial stiffness (per unit length) of the rubber bush and the sleeve. For the preliminary analysis, it has been assumed that the reaction from the rubber and sleeve is a constant pressure. It has also been assumed that there is a 5mm gap between sleeve segments. Both these assumptions are conservative. In the middle of the bosses, where the sleeve supports the pin, it has been assumed that the two components act in composite. Analysis gives a peak bending stress at 58 kN (working) load of about 900 MPa. The transverse shear stress at this load was 69 MPa.

A range of heat treatable steels is allowed for the pin with the quenched and tempered hardness to be in the range 285 to 331 Brinell. This represents an ultimate strength in the range 950 to 1100 MPa and indicates that the pin has a minimal factor of safety, although there would be a small reserve of plastic moment capacity. The shear stress of only 69 MPa indicates that a shear failure is unlikely.

Fatigue failures of pins in service have occurred, so an assessment of the fatigue strength has been carried out. The rubber bushes and sleeves provide a virtually notch-free

environment, so a stress concentration factor of about 1.0 can be assumed for the fatigue analysis. This will only be so while the bushes remain in good condition.

The load spectrum depends on the operation of the vehicle. With straight line motion, there is a load cycle applied to each pin as it passes across the drive wheel. This load cycle is a reduction in the tension in the pin equal to the traction applied to the track by the motor. Thus the alternating stress (as normally defined in fatigue calculations) is half the change in stress resulting from the traction. The Mean Stress depends on the preload and inertia tension in the track. At zero speed and maximum traction (as discussed earlier), the minimum stress will be zero, giving a mean stress equal to the alternating stress. With 15 kN load (the assumed working load described earlier), this is 900 x (15/58)/2 = 116 MPa. Reference 5 suggests a fatigue endurance limit for rotating bending of smooth specimens of 68 000 psi (470 MPa). To adjust this for the mean stress, use a modified Goodman Diagram Mann (5) (x=1), as described on p. 56. This reduces the allowed alternating stress for 10^7 cycles to about 350 MPa. This indicates the pins should be able to sustain regular loads up to about 45 kN, without suffering fatigue failures.

This shows that properly heat treated pins, with no initial surface defects should last as long as the rubber bushes. Once the rubber bushes start to deteriorate at the extremities, the moment applied to the pin will increase resulting in higher stresses and reduced fatigue life.

3.3 Track Shoes

The general arrangement of the M113 shoe, Part Number 11646782, is shown in Figure 18. The shoe is forged from 1345H or 4140H steel to Mil-S-13048, normalized, then quenched and tempered to a hardness of R_c 25-35 (corresponding to a UTS of 850 to 1130 MPa). Contact areas on the guide horn and sprocket holes are then flame hardened to R_c 50 (minimum).

A preliminary stress analysis has been carried out for this arrangement using the design loads summarized in Section 2.8, above. Geometric properties at the critical section have been scaled from original drawings of the shoes.

The analysis of the shoe under direct track tension is shown in Figure 19. The peak stress occurs at the bosses for the rubber bushes. The thickness of the boss is allowed a wide tolerance (between 3.4 and 6.1 mm); an average of 5 mm has been assumed for analysis. The peak working stress is about 69 MPa, less than 10% of ultimate.

Calculation of stresses in the shoe for bending between pins is shown in Figure 20. Here the critical section is mid-way between the two pins. The analysis assumes that the load is distributed uniformly over the rubber road pad. The loading comes from the "turning at speed / crossing an object" case, where the full weight of the vehicle is taken on a single track shoe. The peak working stress is 204 MPa, about 24% of ultimate.

Stresses in the shoe for the transverse bending case are shown in Figure 21. The loading is the same as that for Figure 20, but concentrated on the outer pins and grousers. The critical section is at the holes for the drive sockets, where the geometry of the drive does not permit additional material. The peak bending stress calculated was 437 MPa, to which must be added 89 MPa of compression from the side load, giving a maximum stress of

526 MPa.   With a UTS of 850 MPa, this corresponds to a factor of safety on ultimate strength of 1.60 (compared to a suggested figure of 1.67 for this case.)

Analysis of the load on the guide horn is shown in Figure 22.   This assumes that all the horizontal load from the maximum turning case is resisted at a single horn.   Assuming dry conditions, the majority of load is taken by friction at the road wheel tyre surface, leaving 28% of the horizontal load to be resisted by horn bending.   The calculated stress for this case gives a factor of safety of 2.9.   Given the assumptions inherent in the transfer by friction, a factor of not less than 2.5 would be applicable here.

Figure 23 shows how the load from track tension is distributed between the sprockets on the drive wheel.   This is not conservative, as it assumes uniform distribution. If the vehicle is working in muddy conditions, particles of gravel will be trapped between the sprockets and shoes, resulting in an uneven distribution.   This indicates a bearing pressure in the range 32 to 128 MPa, which is well within the capacity of the steel.   (Note that this analysis does not include a calculation of the bending stresses induced in the boss behind the sprocket.   These may also be important.).

Although there are some conservative and approximate aspects to this analysis, it is clear that the critical case is transverse bending of the shoe, as illustrated in Figure 21. The peak working stress of 526 MPa, indicates that a material with a UTS of the order of 850 MPa is appropriate for the track shoe.   Basic tension and drive loads are not significant compared to the bending actions caused by crossing sharp objects, such as kerbs and traps.

4.   INVESTIGATION OF ALTERNATIVE MATERIAL

It has been suggested that the steel track shoes be replaced by a lighter weight aluminium-based metal matrix composite material.   The two tracks on the M113 account for about 10% of the weight of the vehicle, so the anticipated two thirds reduction in track weight would result in a significant improvement in performance and payload.   The lighter weight tracks would also result in reduced inertia loads, although the above analysis has indicated that these are not critical in the design of the track shoe for strength.

Transverse bending at the sprocket holes as the vehicle crosses uneven ground appears to be the critical design case, as this case resulted in factored stresses of the order of the minimum tensile strength of the existing steel track shoe.   It is not practical to add material at this location - the geometry of the slot is determined by the shape of the sprocket wheel and the pin bosses.   Therefore, in this area, at least, the tensile strength of the substitute material should be of the same order - at least 800 MPa, and preferably 900 MPa.

Wear and strength at the guide horn appears important from the initial assessment.   The guide horn on the steel forging is specified as hardened to Rc 45-55. Assuming this requirement is critical to the wear of the existing horn, the substitute material should match the durability of the steel.

As the guide horn must pass between existing wheels, it is not possible to increase dimensions to accommodate an increased wear rate.   However, it could be that the existing shoes survive several design lives, with only the pins and bushes requiring replacement at

9000 km intervals. In which case, the replacement shoes might be accepted on the basis that they are scrapped at the end of the 9000 km.

Fatigue of the pin bosses may be critical. The peak stress in the boss was of the order of 69 MPa. At this level of stress, fatigue failures in normal aluminium alloys would not be expected. However, it would be necessary to establish the fatigue properties of the particular substitution proposed. Note that the pin bosses are located at the extremities of the forging die (or casting mold), and have a relatively thin section. This may produce unacceptable defects, or lead to a high rejection rate, even in a ferrous material.

Reference 6 provides some information on Silicon Carbide whisker-reinforced Metal Matrix Composites, which is the type of material proposed for the substitution. Aluminium alloy 2124-T6 with 20% by volume SiC whiskers is quoted as having a UTS of 856 MPa, which would be satisfactory. Correct alignment of the whiskers appears critical to the strength achieved, with transverse strength about two thirds longitudinal strength. Properties of SiC particulate reinforced composites are not yet at the desired level.

Other considerations, such as fatigue properties and wear resistance, have not been addressed as yet. These will be important, but the indications from general literature are that this material will have marginally satisfactory properties.

5. CONCLUSIONS

This report gives some guidance on the design loads for tracked vehicles. It amplifies some of the conclusions of Owen (1) and illustrates that design loads for tracks can be calculated from simple analysis.

Inertia loads from operation at high speeds are significant, but do not produce loads in excess of the maximum vehicle traction loads.

Crossing of obstacles, especially when turning, produces the critical load combinations. Bending of track shoes (between the pins and across the width) as the vehicle crosses uneven ground with hard, sharp obstacles produces stresses in excess of the ultimate of the proposed substitute material.

Replacement of the existing steel track shoe by a lighter weight aluminium shoe would result in a significant reduction of the vehicle weight, thereby improving vehicle performance. Preliminary assessment indicates that a silicon carbon whisker-reinforced metal matrix composite could have sufficient strength for the application, although a considerable design effort will be involved in developing the manufacturing techniques to achieve this strength in the critical parts of the shoe.

6. REFERENCES

1. Owen, J. (June 1981). Guidelines for the design of combat vehicle tracks. Ottawa, Canada: DEW Engineering and Development Ltd.

2. —— Report No AFV 1/85, Project Waler, Australian Army, Engineering Development Establishment, Proving Ground, Monegeetta, April, 1985.

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4. Fawcett, J.N., and Robertson, B. (June 1981). Dynamic behaviour of tank tracks. Final report, M.o.D. Contract ref 2043/057 MVEE.

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6. American Society of Metals (1967). Engineering materials handbook, volume 1: Composites. Ohio: ASM International.

7. Hong, R. and Heathcock, C.J. (1988). Squeeze cast aluminium based materials. In The Material Wealth of the Nation, Proceedings of the Bicentennial Conference '88, Institute of Metals and Materials Australasia, Sydney, 2, 394.

PLAN

4521

4429

END ELEVATION

ELEVATION

C.G.

965

1981

END ELEVATION

381 1778 381

2540
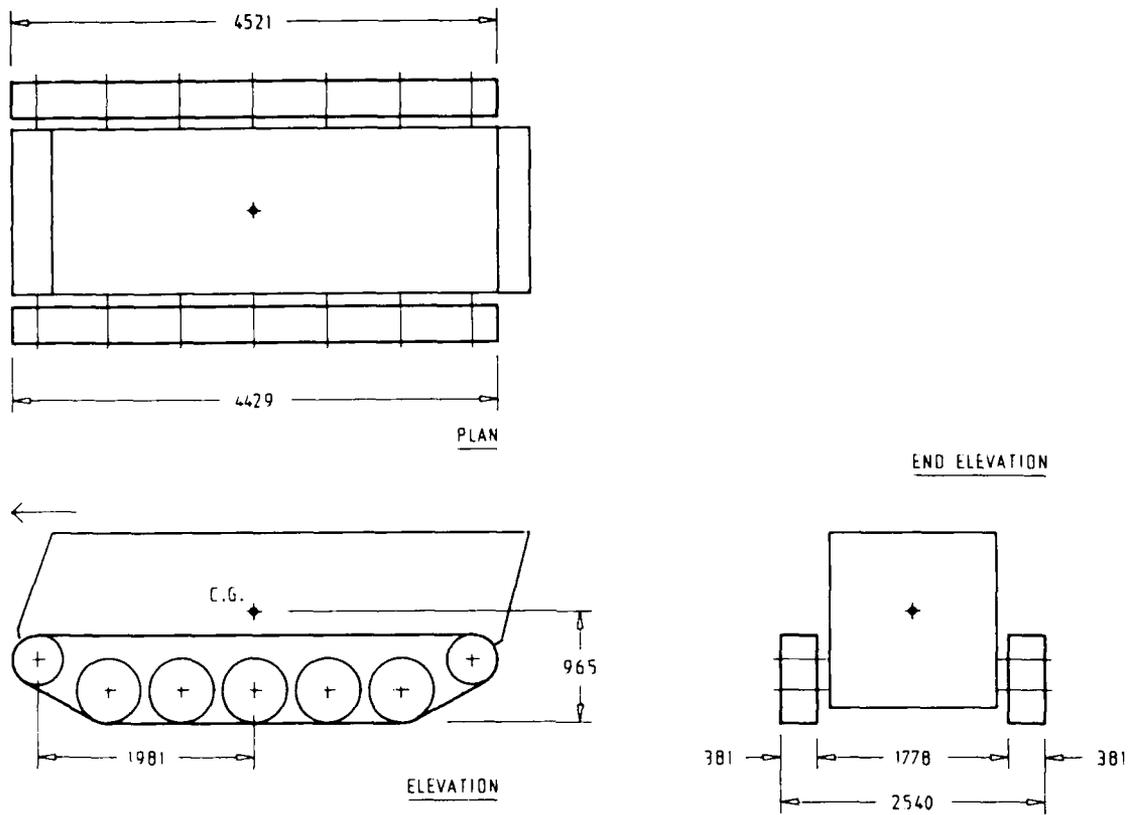
Figure 1   General arrangement.   M113 armoured vehicle.

Single Bush
2 off

Guide Horn

Connecting Pin
with nuts

Multiple Bush
3 off

Road Wheel Pad

Track Shoe

Track Pad

Sprocket Drive Slots
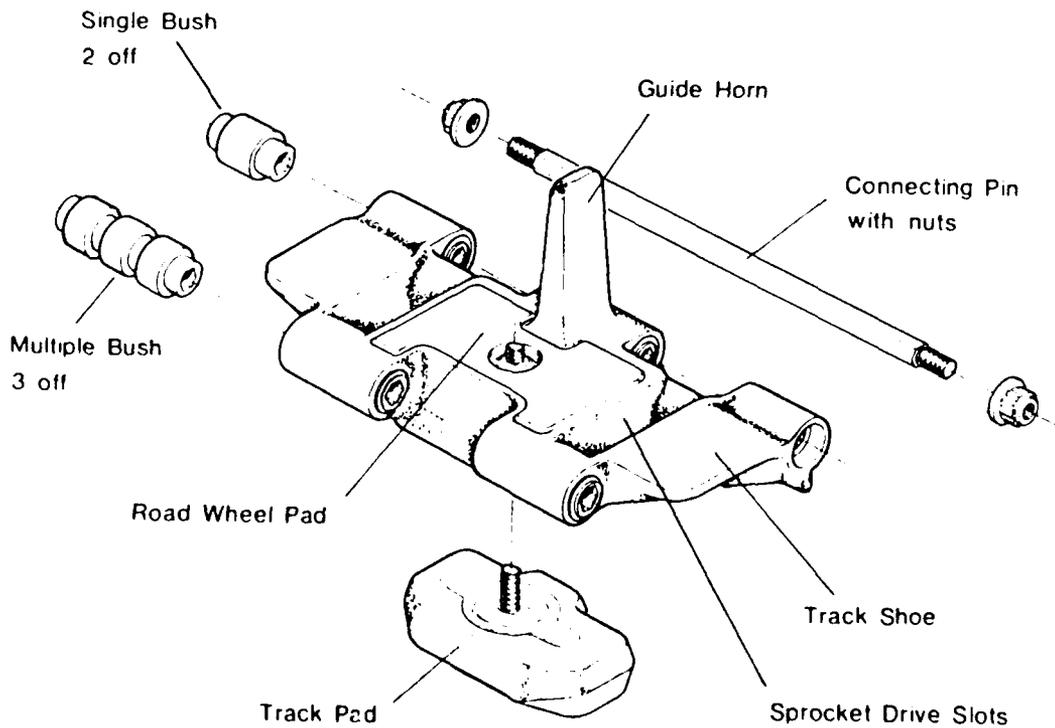
Figure 2   M113 A1 APC - track shoe.
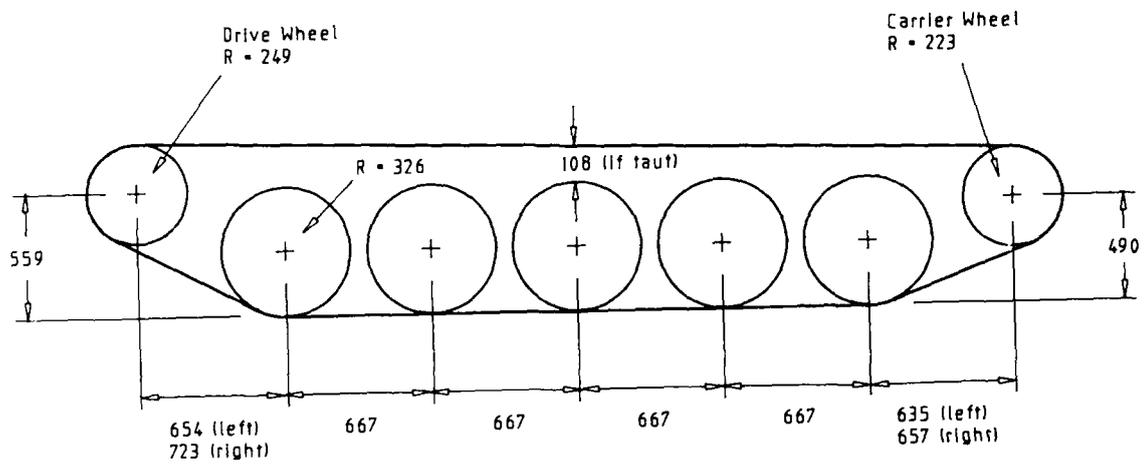
Figure 3   Dimensions of track.
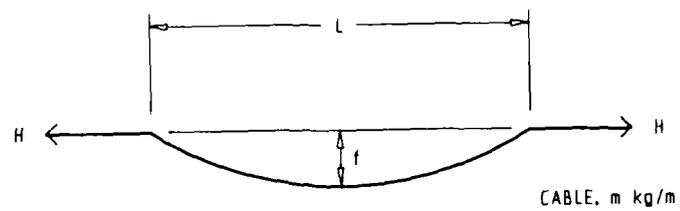
Figure 4   Sag of catenary cable.
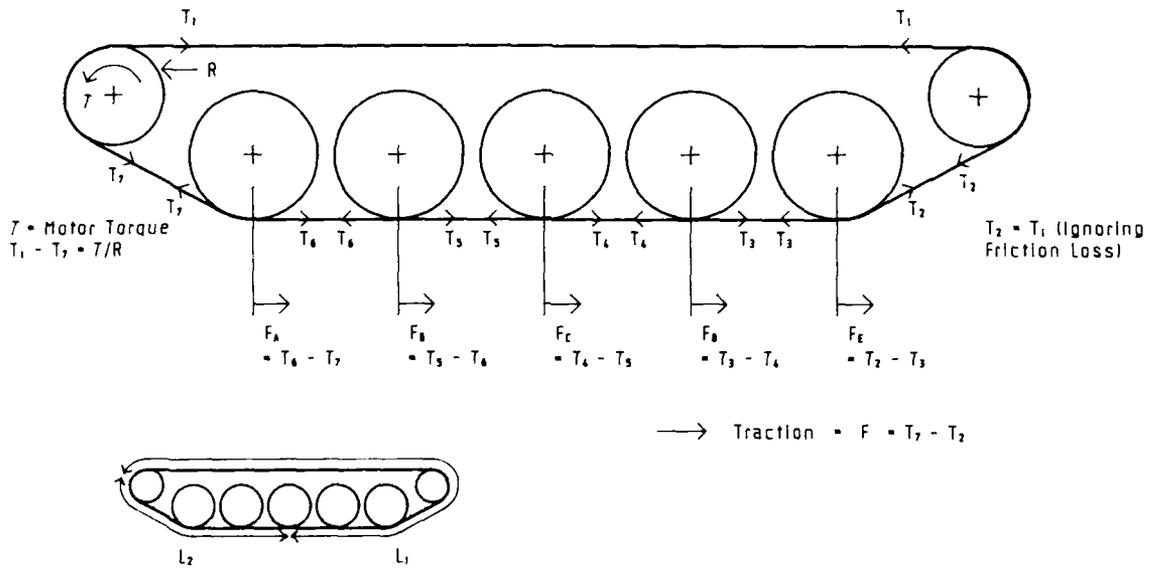
Figure 5   Distribution of tractive load.
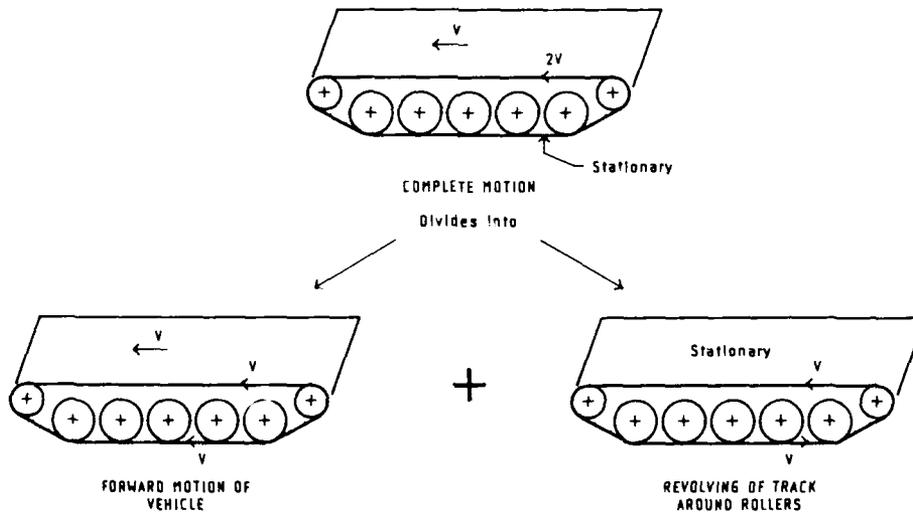
Figure 6   Division of motion for analysis.
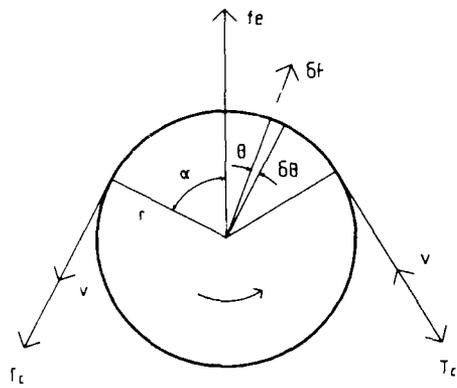
Belt Velocity = v
Centripetal Acceleration = v^2/r
Belt Mass = m per unit length
For ARC $\delta\theta$, Belt Mass = m . r . $d\theta$

$$\delta F = (m . r . \delta\theta)(v^2/r)$$

Total Force, F_c

$$= \int_{-\alpha}^{+\alpha} \delta F . \cos\theta$$

$$= \int_{-\alpha}^{+\alpha} m . r . d\theta . v^2/r . \cos\theta$$

$$= mv^2 . 2 \sin\alpha$$

Tension in Belt (resisting F_c) = T_c
By Equilibrium, $2T_c \sin\alpha = F_c$

$$= mv^2 . 2 \sin\alpha$$

$$\therefore T_c = mv^2$$

Figure 7   Derivation of belt tension due to centripetal forces.
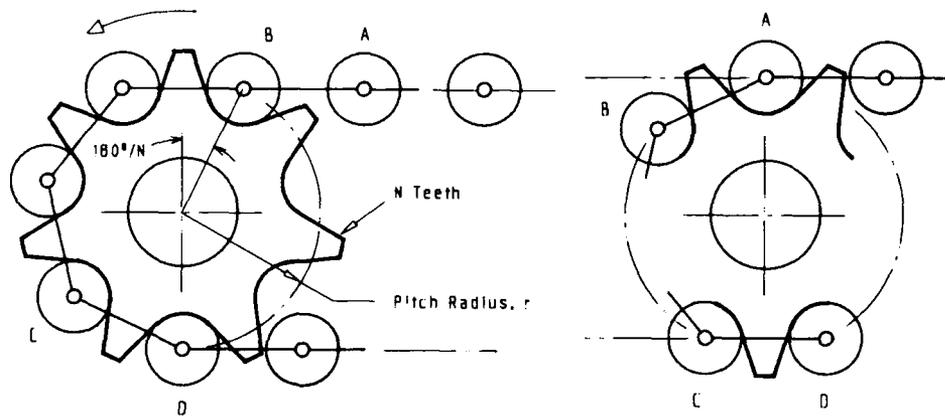
Figure 8   Chordal action of a roller chain.
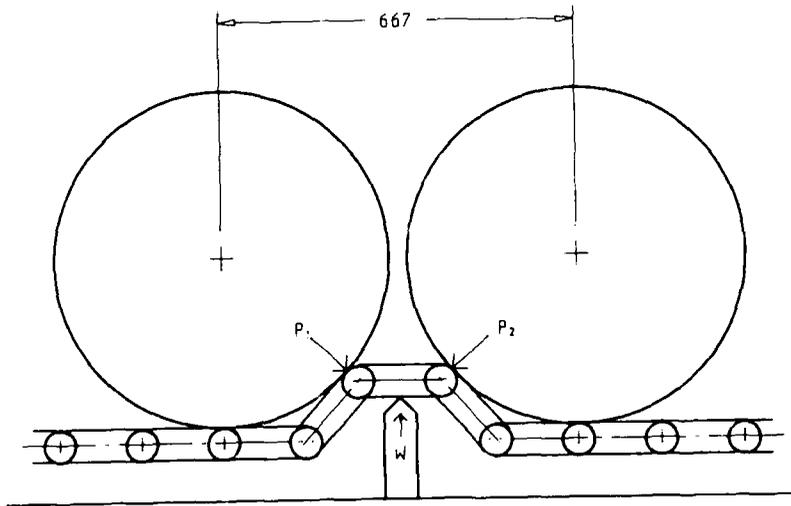
Slack Required

= 5 x 152 - 667

= 93 mm

Figure 9   Crossing a sharp object (1).
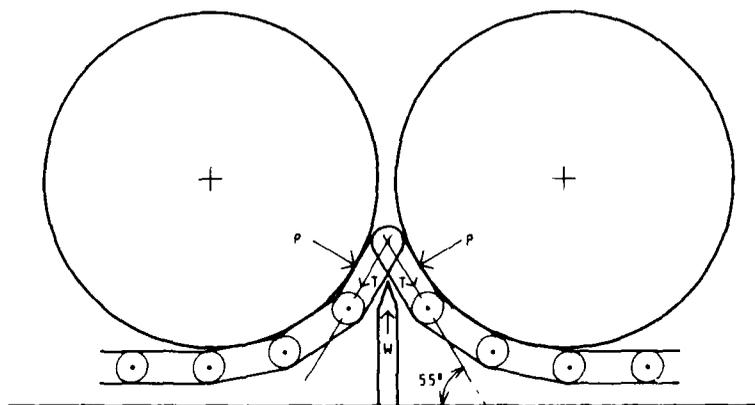
Slack Required

= 6 x 152 - (667 + 2 x 30)

= 185 mm

Figure 10   Crossing a sharp object (2).

Vehicle Path

mv²/2   C.G.

mg

965

H_0

R_0 R_1

1080

Plan View

$\frac{mv^2}{r}$

r
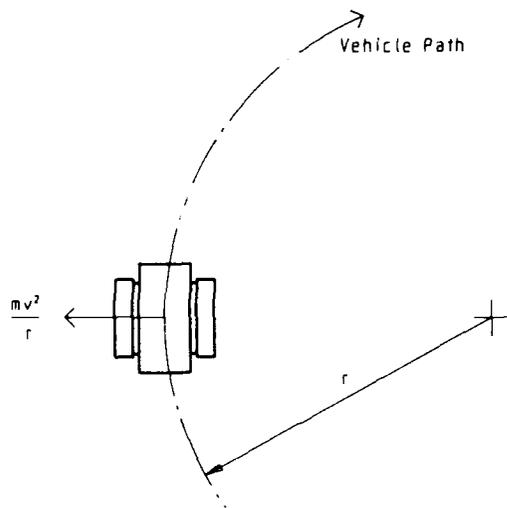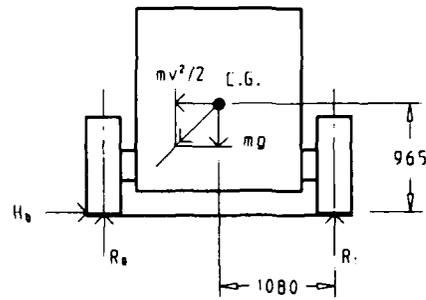
AT POINT OF OVERTURNING

$$R_1 = 0 = \frac{mg}{2} \cdot \frac{mv^2}{r} \bullet \frac{965}{2 \times 1080} \quad OR \quad v^2 = r.g \frac{1080}{965}$$

For M-113, Minimum Turn Radius = 22.7 ft = 6920.

giving $V_{max\ (turning)}$ = 8.7 m/s

At this speed add Radius, R_0 = mg = 120 kN

H_0 = mv²/r = 134 kN

Figure 11   *Vehicle turning – basic forces.*

AT POINT OF OVERTURNING

R_1 = 0, R_0 = mg.cos α, H_0 = mg.sin α
for M113,   α = tan⁻¹ (1080 / 965) = 48°

AT POINT OF SLIDING

$(H_0 + H_1) = \mu (R_0 + R_1)$
gives   sin α = μ.cos α
for   μ = 0.8,   α = 38°

α

mg

R_1

R_0

H_0
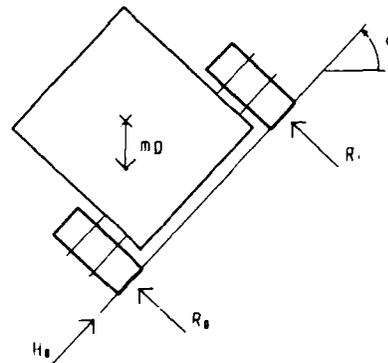
Figure 12   *Vehicle on side slope – basic forces.*

27
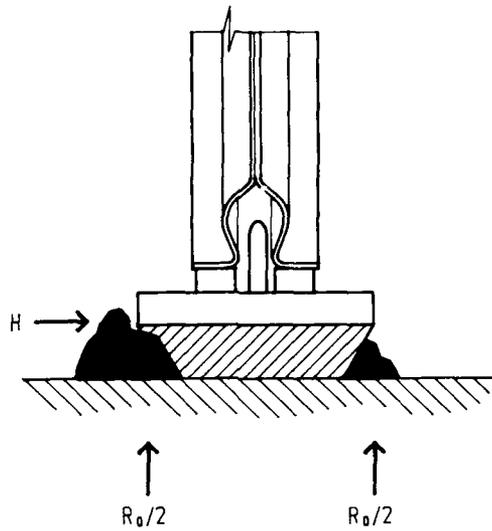
Figure 13   Turning vehicle striking object.
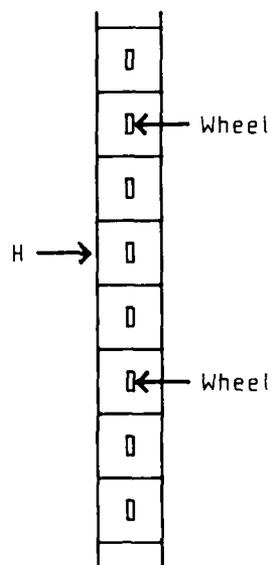

Figure 14   Impact between wheels.
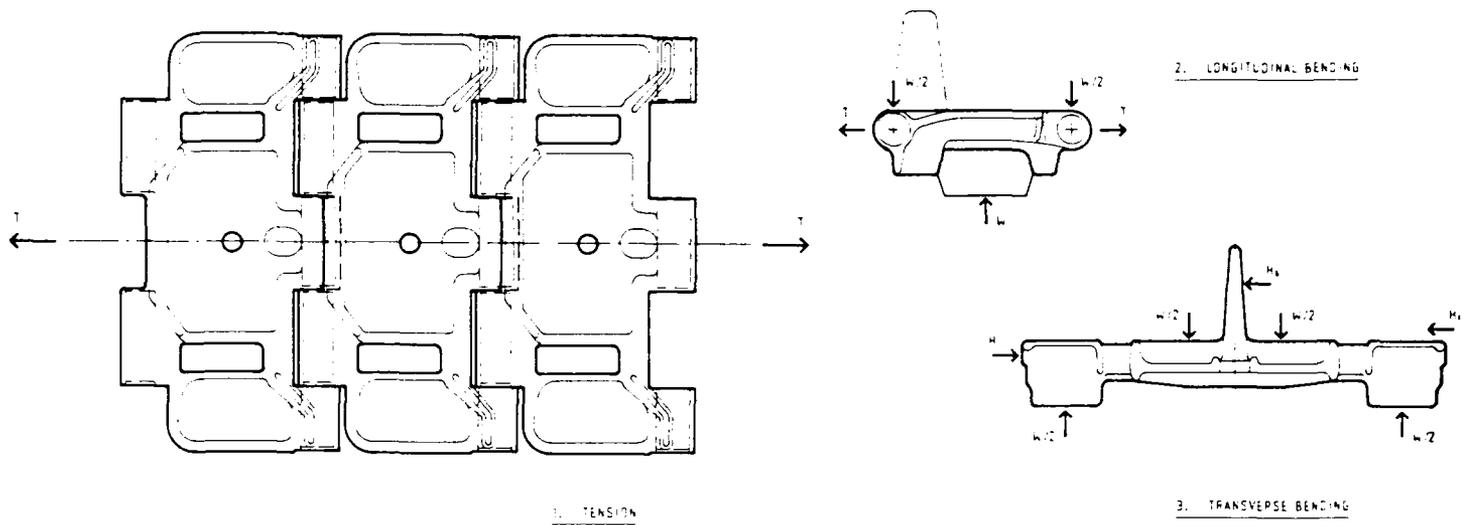
Figure 15   *Design load cases.*

Plan of Rubber Bushes

Rubber Bushes
30.5 O.D. x 22.5 I.D.
(as compressed)
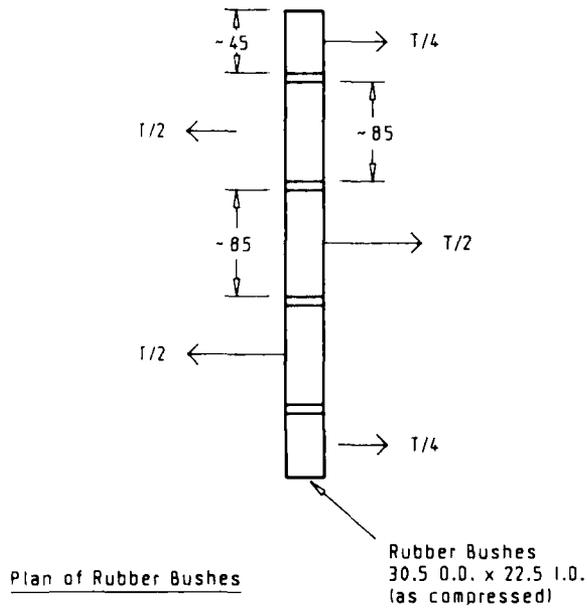
Average Bearing Pressure

T = 58 kN

$$T/A = \frac{58000}{2 \times 85 \times 22.5} = 16 \text{ MPa}$$

$\therefore$

Bush Stiffness

$K_r = T_1/x = \beta LG$ 6
$D/d = 30.5 / 22.5 = 1.36$

for Long Bush, $\beta = 280$ (from Ref. 3)
Shore 70 Rubber   G = 1.73 MPa
$\therefore K_r = 2800 \times 1.73 \times 85 \times 2$ / side
      $= 0.8 \times 10^6$ N/mm
with 2 sides. Double Deflection
for same load
$\therefore K_{r\ eff} = 0.4 \times 10^6$ N/mm / joint

for 63 Links @ 58 kN Tension
   $6 = 63 \times 58000 / 0.4 \times 10^6$ mm
   = 9 mm Total

Figure 16   *Rubber bush analysis.*

STRESS ANALYSIS   (T = 58 kN)

1. Pin & Sleeve Acting Together

    Moment = 0.7 kN.m
    Section Modulus = $\pi \times 22.5^3/32$
    M/Z = 630 MPa

2. Pin alone

    Moment = 0.4 kN.m
    Section Modulus = 438 mm³
    (Octagon = 15.875 across flats)
    M/Z = 913 MPa
    Area = 209 mm²
    Shear Stress = 58000/4/209 = 70 MPa

Section of Pin & Sleeve
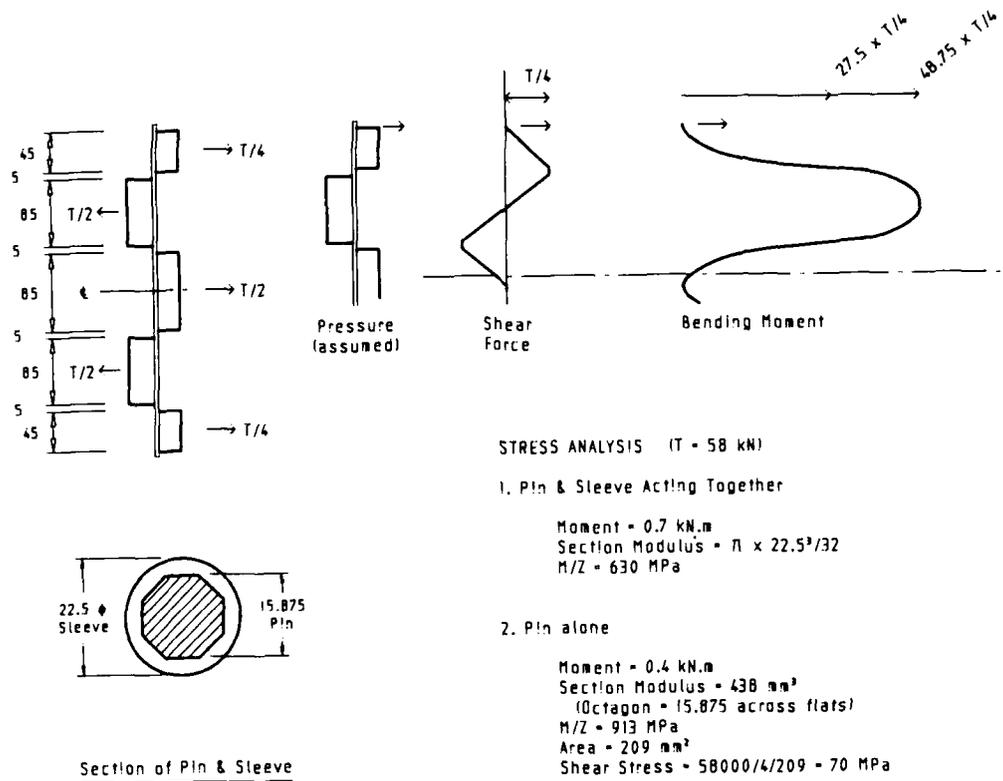
Figure 17   *Pin bending analysis.*
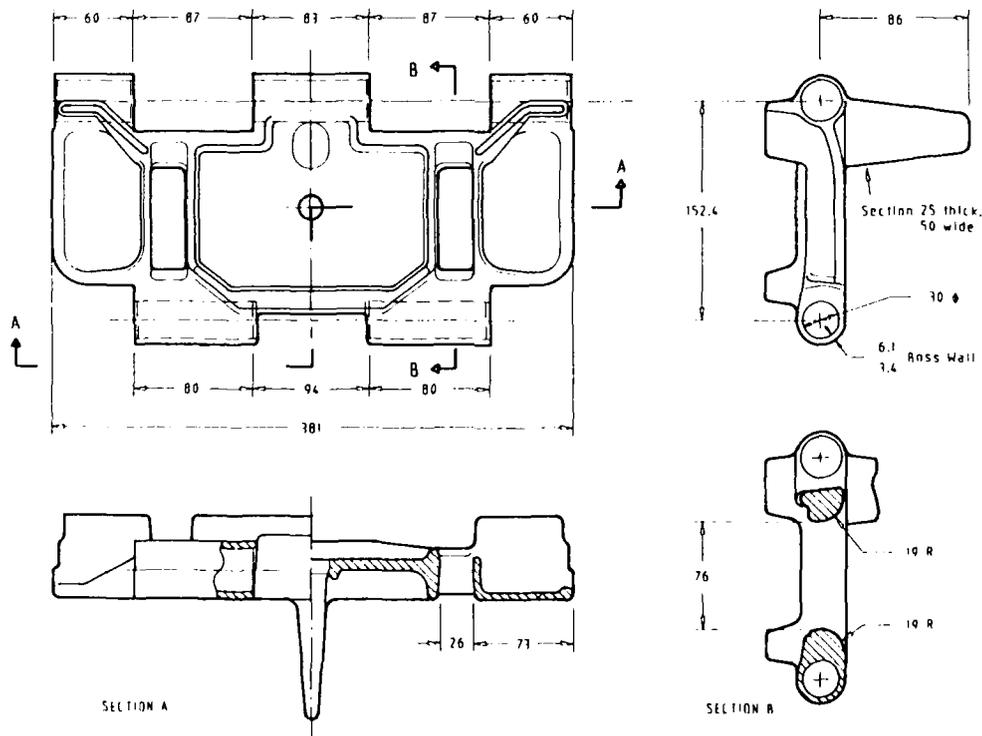
Figure 18   *Shoe layout.*

30
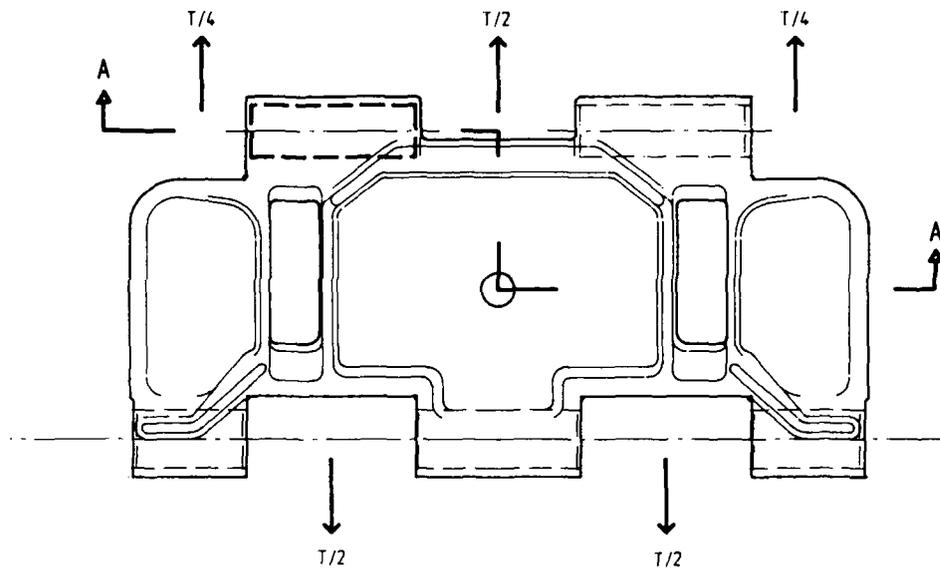
For T = 50 kW
@ (1) T/2A = 69 MPa
@ (2) T/2A = 19 MPa

(1) AREA ~ 420 mm²
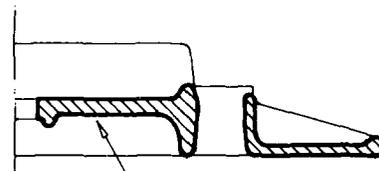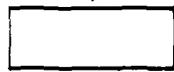
(2) AREA ~ 1500 mm²
SECTION MODULUS ~ 15000 mm³

SECTION A

Figure 19   Tension on shoe.

Rubber Shoe

W

26   100   26
(approximate dimensions)
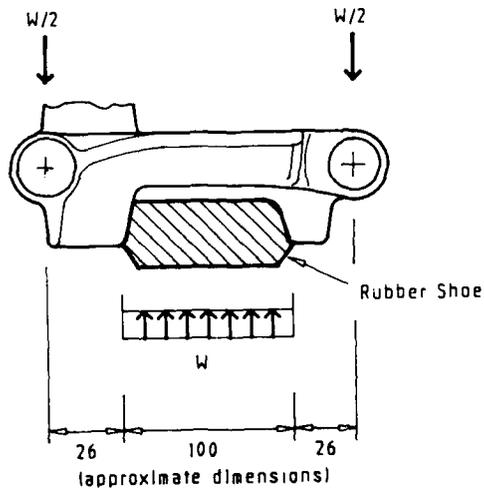
Moment in Shoe $= \dfrac{W}{100} \cdot \dfrac{(76^2 - 26^2)}{2}$

$= 25.5W$ kN.mm

Section Modulus $= 15000$ mm³
see (2), figure 19

for W = 120 kN,
M/Z = 204 MPa

Figure 20   Bending between pins.
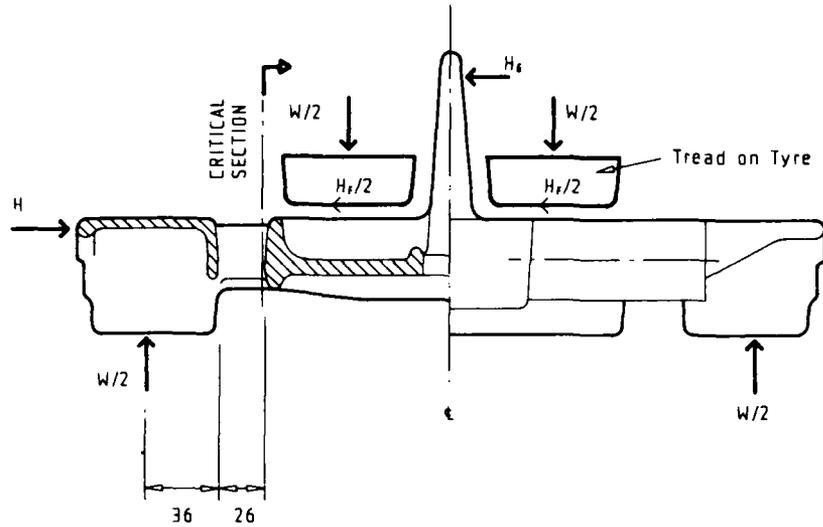
Moment = W/2 (36 + 26)

Critical Section is Section B, Figure 18
Area ~ 1500 mm², Z ~ 8500 mm²

for W = 120 kN, M = 3.72 kN.m, M/Z = 437 MPa
for H = 134 kN,   H/A = 89 MPa

                        Maximum Stress = 526 MPa

Figure 21   Transverse bending.
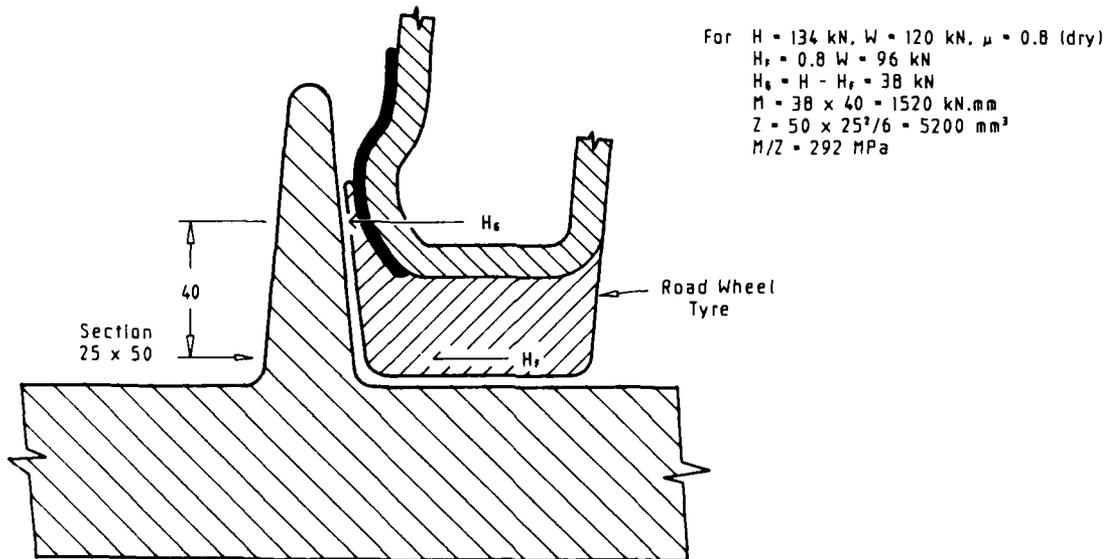
For   H = 134 kN, W = 120 kN, μ = 0.8 (dry)
      H$_f$ = 0.8 W = 96 kN
      H$_s$ = H - H$_f$ = 38 kN
      M = 38 x 40 = 1520 kN.mm
      Z = 50 x 25²/6 = 5200 mm³
      M/Z = 292 MPa

Figure 22   Guide horn bending.
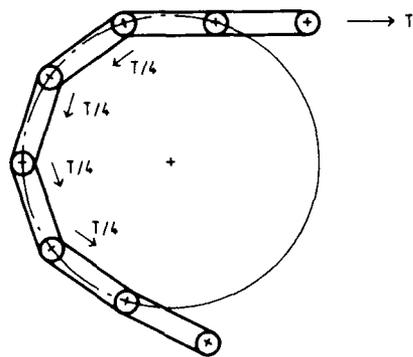
Each sprocket has 2 teeth
with constant radius of 19 mm
and bearing width of 17 mm
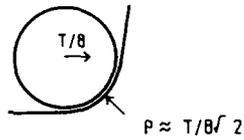
For   T = 58 kN
         P = 10.25 kN
         A = 19x17 = 323 mm²
         P/A = 32 MPa

$P \approx T/8 \sqrt{2}$

DETAIL OF TOOTH

Figure 23   Sprocket loading.

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Analysis of armoured-vehicle track loads and stresses,
with considerations on alternative track materials

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KEYWORDS

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Stress		

SUBJECT GROUPS   0071L

ABSTRACT

This Report sets out the basis for calculating the loads applied to the Track of an Armoured-Vehicle in service conditions. The analysis includes the effects of normal traction, inertia forces from rotation of the individual shoes, friction, and vibration. It also includes a basic analysis of forces induced when the vehicle turns and negotiates obstacles.

The Report looks in some detail at the tracks of the M113 Vehicle, and provides an estimate of the loads and load cycles applicable to this configuration. A stress analysis of the three components of the track (Shoes, Rubber Bushes, and Pins) is included.

Consideration is given to the replacement of the existing Track Shoes with ones made from an Aluminium Alloy/SiC metal matrix composite. The Report includes a summary of the effects to be tested and the property requirements to be met if such a replacement were to be tried.