1

Tracked Vehicle Running Gear and Suspension Systems

The running gear systems used on high speed, mainly military, tracked vehicles provide four essential functions:

- the transmission of drive to a relatively large number of road wheels;
- the distribution of the weight of the vehicle over a relatively large area;
- a large suspension displacement to allow high speeds over rough terrains; and
- a particular requirement of military armoured vehicles, the running gear system should occupy the minimum space in the overall vehicle envelope in order to maximise internal hull volume (as will be shown in Section 8.4, this is a particular attribute of tracked vehicles compared to wheeled vehicles of similar soft-soil performance).

In addition, the running gear must be of minimum weight, reliable, easy to maintain, and compared to some other vehicle components, relatively cheap to produce.

1.1 General Arrangement

Figure 1.1 shows the running gear of the Warrior Infantry Fighting Vehicle (IFV) and is typical of modern practice. Trailing suspension arms carry rubber-tyre road wheels and operate transverse torsion bars which run across the floor of the vehicle. Rotary vane hydraulic dampers are incorporated into the pivots of the front, second and rear road wheel stations. Link tracks run under the road wheels and around hull-mounted drive sprockets and return idlers. Track pretension is adjusted by means of oil-filled rams reacting against the idlers, which are carried on short pivoting arms. The drive sprockets are front-mounted but could be at the rear of the vehicle, depending on the position of the power pack. Small diameter rollers support the top run of the track. The track link pivots are rubber-bushed and the links are fitted with replaceable rubber road pads to minimise road damage and reduce noise and vibration.

Figure 1.2 shows the arrangement on the Leopard 2 Main Battle Tank (MBT). Rotary friction dampers are built into the front three and rear two axle arm pivots. The vehicle is fitted with rubber-bushed double-pin tracks (see Chapter 2).

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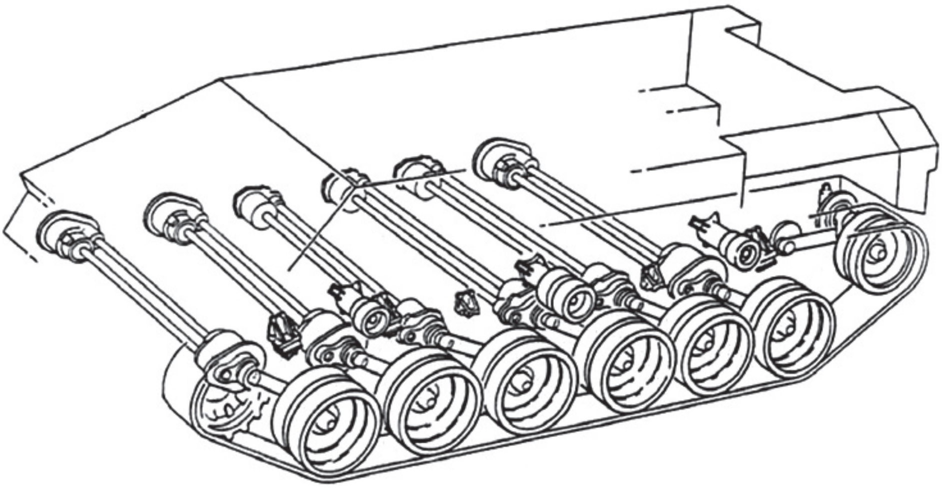

Figure 1.1 Warrior running gear layout. *Source:* Courtesy of Ministry of Defence.

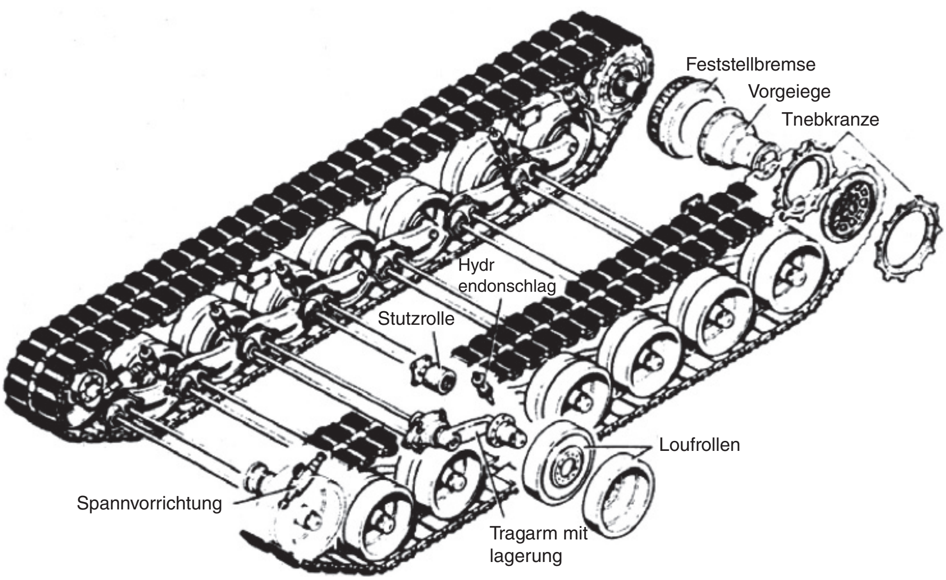

Figure 1.2 Leopard 2 running gear layout. *Source:* Courtesy of ATZ.

1.2 Transverse Torsion Bars

Modern high-strength spring steels, used with suitable presetting, shot peening and corrosion prevention techniques, allow nominal shear stresses of up to 1250 mPa to be used with a reasonable fatigue life [1.1, p. 226]. Suspension torsion bars are only loaded in one direction and so can be 'preset'. To preset a torsion bar, it is wound up to induce

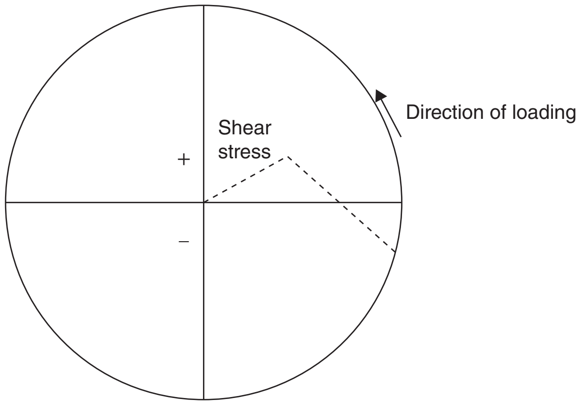

Figure 1.3 The principle of presetting a torsion bar.

partial yielding in the outer layers of the bar. On release, the outer layers take on negative shear stresses and torques opposed by positive stresses and torques in the inner layers of the bar (Figure 1.3).

The relationship between the various variables that affect the maximum shear stress in the bar can be explored by setting up a suitable spreadsheet. The vehicle will be considered as a notional MBT with a sprung mass of 600 kN and an effective torsion bar length of 2.13 m. The variables that can be considered are the axle arm length (initially taken as 450 mm), the number of road wheels (initially taken as 12) and the stiffness of the bar. The latter can deduced from the ratio of wheel loads at full bump and at static F_B/F_S, initially taken as 3:1, and the required static to bump suspension displacement Δ_{SB}, taken as 350 mm. This gives a heave natural frequency of about 1.2 Hz, which is typical for an MBT. The shear modulus C is set at 76 mPa [1.1, p. 226]. The diameter of the bar is left open.

This gives a maximum shear stress q_{max} of 1326 mPa, which can be considered too high for a good fatigue life. Increasing the arm length to 500 mm increases maximum torque on the bar, but also reduces maximum wind-up angle; q_{max} reduces to 1258 mPa. This may be acceptable depending on the duty cycle. Measurements show that the front wheels nearly always have the most severe duty, largely because of the pitching motion of the vehicle; this can be controlled by an adequate measure of damping.

Softening the suspension to give a F_B/F_S value of 2.5 and with axle arm length R at 450 mm increases q_{max} to 1371 kPa. With the stiffer suspension, increasing the number of wheels to 14 reduces the value of q_{max} to 1276 kPa. With the 0.5 m wheel arms, q_{max} reduces to 1211 mPa. If the length of wheel arm can be further increased to 0.55 m without causing interference between the arms, then q_{max} further reduces to 1155 mPa.

Another possibility is of course to simply reduce the static to bump displacement to, say, 325 mm with 500 mm wheel arms, 14 wheels and the stiffer suspension; q_{max} is then 1158 mPa. Some of the different possibilities are summarised in the table overleaf.

Number of wheels n	Arm length R (m)	Static to bump travel (m)	F_S (kN)	F_B/F_S	Diameter of torsion bar (mm)	q_{max} (mPa)	Mass (kg)
12	0.45	0.350	50.00	3.000	62.0	1326	603
12	0.50	0.350	50.00	3.000	65.8	1258	677
12	0.45	0.350	50.00	2.500	57.7	1371	522
14	0.45	0.350	42.86	3.000	59.7	1276	651
14	0.50	0.350	42.86	2.500	58.9	1252	633
14	0.50	0.350	42.86	3.000	63.3	1211	731
14	0.55	0.350	42.86	3.000	66.6	1155	811
14	0.50	0.325	42.86	2.786	62.9	1158	722
12	0.50	0.325	50.00	2.786	65.3	1204	668

The factors that reduce maximum shear stress are longer wheel arms, stiffer suspension and increased number of wheels. As maximum shear stress is reduced, the weight of the bars increases in a virtually linear relationship. This is for the 'spring' part of the torsion bar, that is, neglecting the end fittings which are usually splines.

Suspension bump displacement, and hence maximum torsion bar stresses, is normally limited by some form of bump stop acting on the suspension arm as shown in Figure 1.1. However, bump stops are not fitted on all or some of the wheels of the Alvis Stormer and Scorpion family of vehicles; the wheels are allowed to bottom through the top run of the track onto the hull sponson and trackpads. This apparently crude strategy works well in practice; it saves weight and reduces torsional loading on the axle arms.

If it is not possible to obtain satisfactory values of shear stress with hull width torsion bars, then two strategies can be used to effectively lengthen the bars. One is to approximately double the length of the bar by 'folding' it back. This arrangement was used on the Second World War (WW2) German Panther tank as shown in Figure 1.4. The vehicle used eight interleaved wheels per side, both to improve soft-soil performance and to reduce loading on the rubber tyres of the wheels. Apart from the extra complication, another disadvantage of this arrangement is the possibility of mud and stones becoming stuck between the wheels; at low temperatures this could freeze and immobilise the vehicle. Maximum shear stresses in the torsion bars were limited to a mere 200 mPa because of the qualities of the available steel and the somewhat unrealistic – for a wartime tank – design life of 10 000 km. Factors tending to increase stress levels were the very soft suspension (a pitch frequency of only 0.5 Hz) and the very short axle arms; the latter was a requirement of the interleaved wheels. The static to bump displacement was only 200 mm, tending to reduce stress levels.

A second strategy is to enclose the torsion bars in torsion tubes. However, torsion tubes are intrinsically much stiffer than the torsion bars, and the diameter of the tubes is increased as a result of the need to pass them over the torsion bar end fittings. Some experimental work has been conducted on the bar and tube arrangement shown in Figure 1.5. The stiffness of the bar was measured at 0.204 kNm/degree and that of the

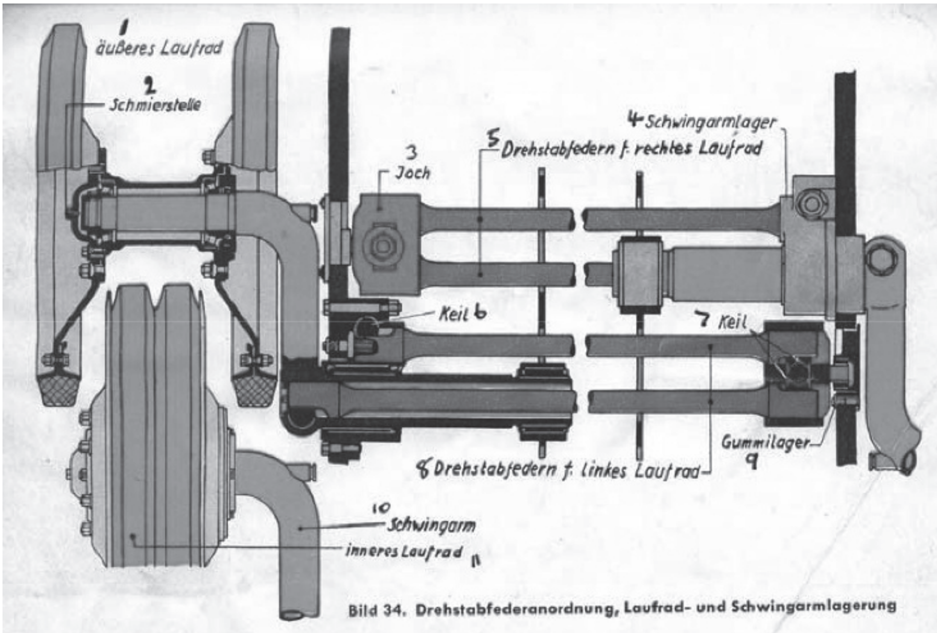

Figure 1.4 Panther torsion bar arrangement.

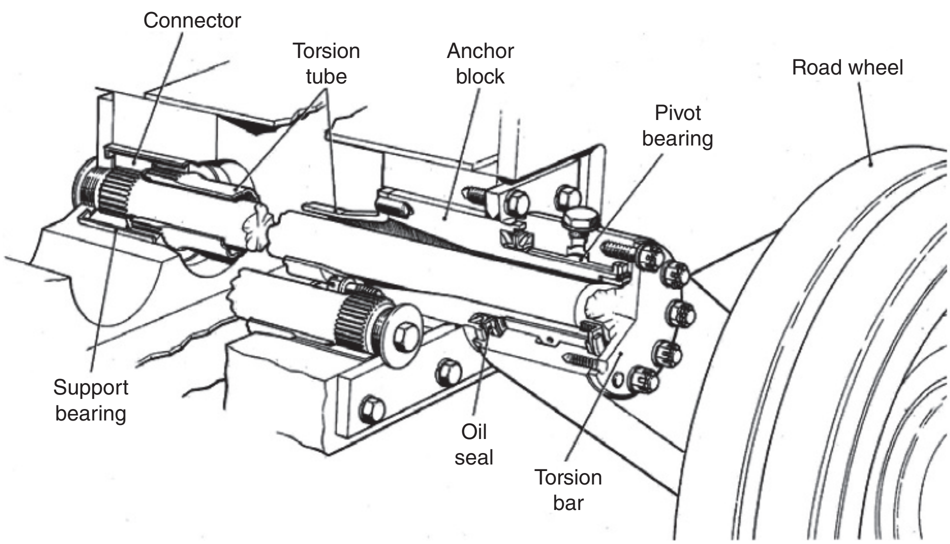

Figure 1.5 Torsion tube over bar arrangement. *Source:* Courtesy of Ministry of Defence.

tube at 1.89 kNm/degree; that is, the tube is over 9 times stiffer than the bar. The combined stiffness was 0.184 kN/degree.

The failure torque of the tube was measured at about 33 kNm and that of the bar at 14 kNm. It is therefore tempting to reduce the wall thickness of the tube and hence its stiffness, but there is then the possibility of the tube buckling because of the compression component of stress in the tube.

If the requirement was to provide a bar with the combined stiffness of bar and tube, then its effective length would need to be approximately 11% (180 mm) longer. In practice, it would be preferable to use either longer wheel arms or stiffer suspension.

1.3 Coil Springs

Transverse torsion bars take up space inside the vehicle and tend to raise the vehicle profile. Externally mounted suspensions are therefore often preferred, including the use of coil springs or hydrogas suspension units. Coil springs are less efficient than torsion bars in terms of energy storage per unit mass as shear stress is not symmetrical across the section of the spring. The spring curvature causes higher shear strains and hence stresses on the side of the spring. The spring also has to carry the direct load, causing direct shear stress across the spring. The maximum shear stress is usually calculated by using the Wahl stress correction factor [1.1]. Springs are normally preset ('scragged') and shot-peened. An advantage of coil springs is that failure of a coil usually allows the spring to still carry load. Springs can be nested coaxially to increase space efficiency. The centre lines of the springs should be kept straight as the spring deflects to avoid any extra stresses caused by bending of the springs.

Various arrangements have been devised to use coil springs in tracked vehicle suspensions. Figure 1.6 shows the system used on the WW2 Cromwell MBT. A bellcrank extension of the axle arm is pivoted to a cylindrical canister that contains the spring and operates one end of the spring. The other end is reacted by a rod that passes through the spring and is pivoted to the hull side. A compression spring is thus made to effectively act in tension. The springs are of small diameter to minimise intrusion into the hull space. However, this does give the springs a high Wahl stress correction factor. Telescopic dampers are fitted to axles 1, 2, 4 and 5. Available suspension displacement

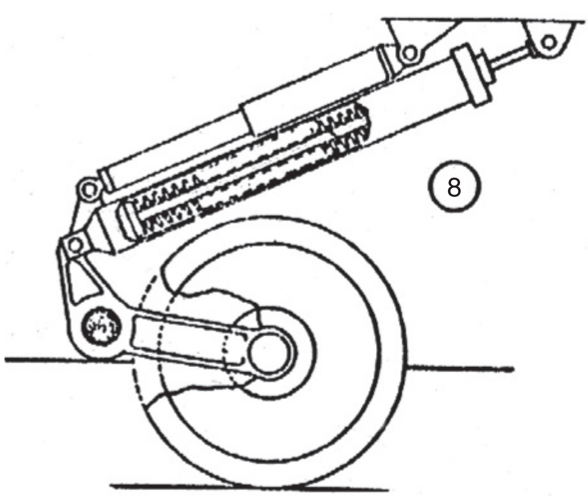

Figure 1.6 Cromwell MBT suspension unit. *Source:* Courtesy of Ministry of Defence.

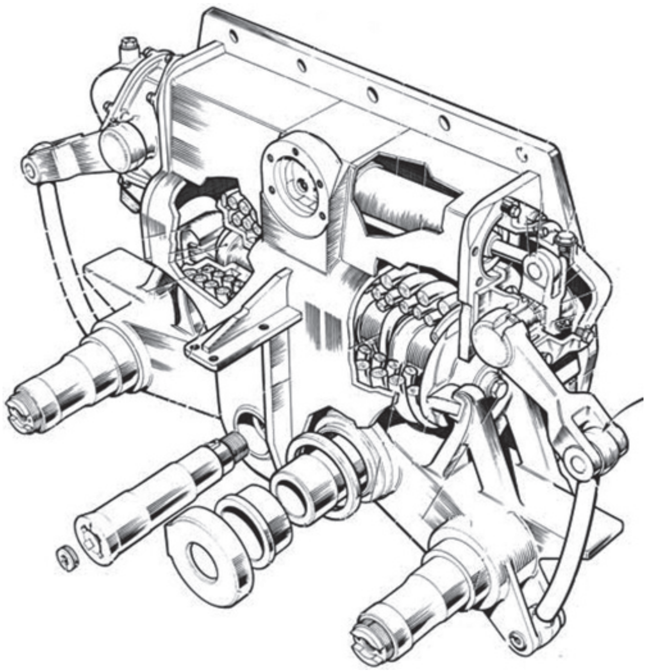

Figure 1.7 Chieftain bogie suspension unit. *Source:* Courtesy of Ministry of Defence.

was 226 mm bump and 190 mm rebound. The suspension was soft with a heave natural frequency of about 1 Hz.

Figure 1.7 shows the arrangement used on the Centurion and Chieftain MBTs, usually called a Horstman bogie. A coil spring pack reacts between leading and trailing wheel arms via bellcranks and 'knife-edge' bearings so that virtually equal loads are applied to both wheels. The wheels can also articulate without deflecting the springs. The spring pack comprises three nested coil springs. The innermost spring acts as a bump stop when both wheels move upwards, which limits the maximum average deflection of the two wheels to only 86 mm. This can severely limit performance when the vehicle is pitching at or near resonance on longer wavelengths, especially likely because damping levels are low. When the bogie articulates, the maximum bump displacement on one wheel can increase to 158 mm with the other wheel in the static position and the spring pack fully compressed. This can be useful when traversing large short-wavelength obstacles (e.g. rocks and tree trunks). Maximum spring shear stresses are quoted at about 1000 mPa. Telescopic dampers of fairly small force capacity are fitted. The weight of a complete assembly is 777 kg, of which the coil spring pack is 137.4 kg. Six units represent about 9% of vehicle mass, which is a high figure especially for a suspension of fairly limited performance. In comparison the suspension of Challenger 2 represents about 5.5% of vehicle mass for a suspension of far higher performance.

A larger improved bogie suspension was produced for the Khalid MBT. Here the maximum average double bump deflection was increased to 180 mm and the

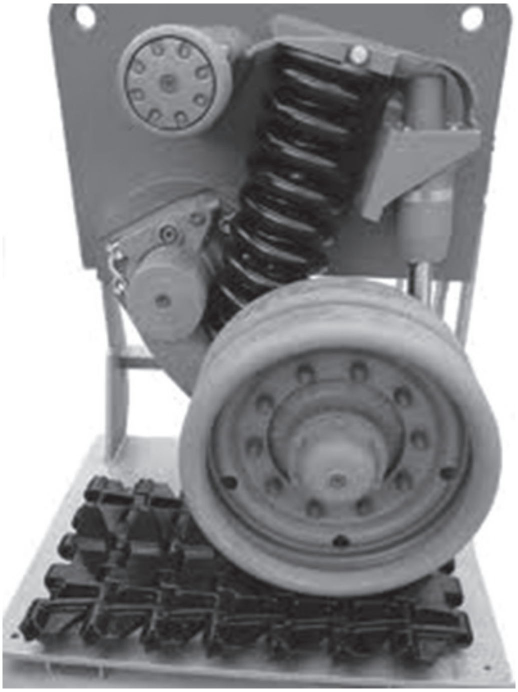

Figure 1.8 Merkava 4 suspension unit. *Source:* Courtesy of MANTAK.

single-wheel bump displacement increased to 241 mm. The mass was increased to 898 kg with the spring pack at 162 kg.

The Israeli Merkava Mk 4 MBT uses trailing arms operating individual coil spring units (see Figure 1.8). The first two and last two wheel stations have hydraulic rotary dampers. Road wheel travel is quoted as 300 mm bump and 304 mm rebound. The high static deflection implies a comparatively soft suspension with a low bump force. The suspension is also fitted with long-travel hydraulic bump stops similar to those on the Leopard 2.

1.4 Hydrogas Suspensions

Hydrogas (otherwise oleopneumatic, hydropneumatic, gas-over-oil) suspensions use, as their name implies, a gas volume as the spring medium actuated by a piston and an oil column. The gas, usually nitrogen, is normally separated from the oil by a floating piston or rubber diaphragm. Units have been built without a separator piston or diaphragm between the gas and oil, a similar arrangement to that used on most aircraft undercarriage ‘oleos’ [1.2].

1.4.1 Challenger MBT Hydrogas Unit

The hydrogas suspension units fitted to the Challenger 1 and Challenger 2 tanks were designed and developed at the Military Vehicles and Engineering Establishment

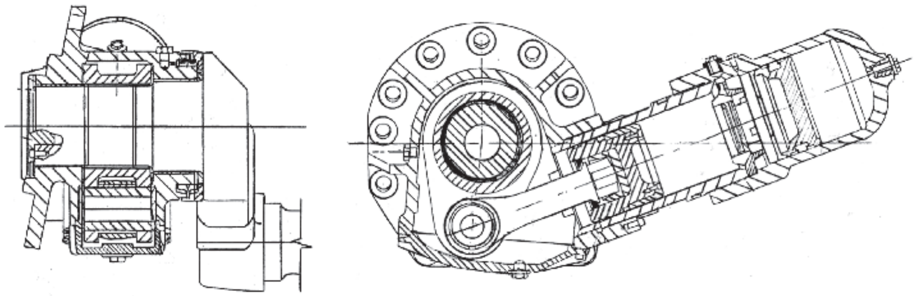

Figure 1.9 Challenger hydrogas suspension unit. *Source:* Courtesy of Ministry of Defence.

(MVEE). Many hundreds of hours were spent testing units on 300 kN 0.5 m stroke hydraulic actuators in the MVEE test laboratories. Actuator displacement duty cycles were based on real-time inputs from test vehicles running on severe-roughness cross-country courses and from computer modelling studies. A particular requirement was to develop a sealing system and cylinder bore finish that would allow the units to run without attention between servicing at approximately 2000 km intervals. A 250 hour test was set as an objective to simulate this requirement. Hydrogas suspension units have been fitted to Challenger MBTs since 1983.

Figure 1.9 shows a cutaway of the unit. A one-piece axle arm pivots on plain bearings and operates a pressure piston via a crank and connecting rod. The connecting rod again uses a plain bearing at the crank end and a knife-edge bearing at the piston end. Oil and gas chambers are in line and separated by a light alloy floating piston. A compact disc-spring damping valve is fitted between the pressure and separator pistons. The main pivot bearing housing is a steel casting with a screwed-on forged-steel oil cylinder. The gas chamber is similarly a steel forging screwed onto the oil cylinder. The gas pressure at static is about 12.8 mPa.

Bump stops are not fitted to any of the wheel stations. However, as will be discussed later (see Section 1.4.2.6), the suspension on the Challenger is heavily damped and this reduces maximum suspension displacement. The suspension also has a strongly rising spring characteristic that results in a sort of built-in bump stop. The units are sufficiently robust to carry the resulting forces. However, if the unit is overloaded (overpressured) it fails benignly by leakage around the screw thread between the cylinder and the gas chamber. Each unit weighs 287 kg.

1.4.2 Measured Characteristics of a Challenger Unit

1.4.2.1 Spring Characteristics

As part of a research project, an extensive series of laboratory tests were conducted to measure the spring and damper characteristics of a Challenger hydrogas unit. The unit was subjected to sinusoidal inputs over a frequency range of 0.001–2.0 Hz and with amplitudes of ±175 mm and ±200 mm. Tests were conducted with and without the damper unit in place.

Figure 1.10 shows the measured force/displacement characteristic at 0.8 Hz from −50 mm to +350 mm. The effective polytropic index is calculated at about 1.66.

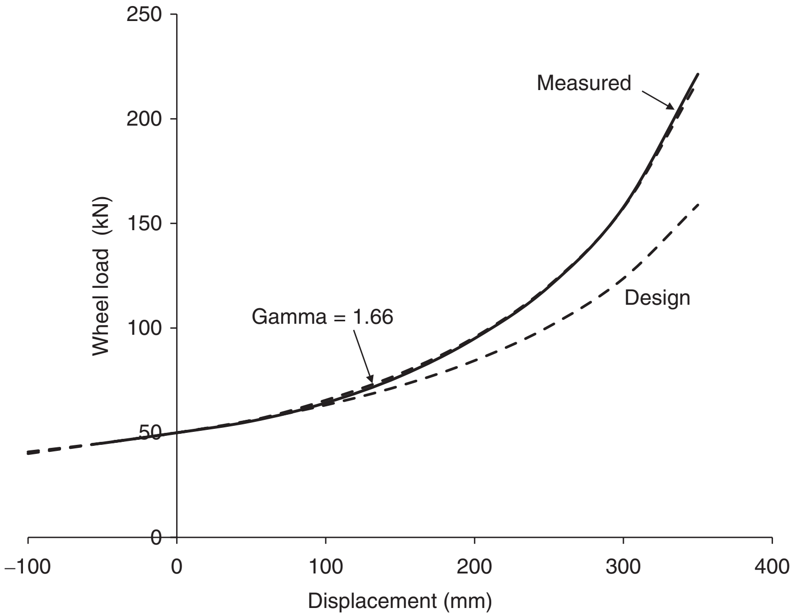

Figure 1.10 Challenger suspension characteristics at 0.8 Hz as measured, a fitted curve and a typical design curve.

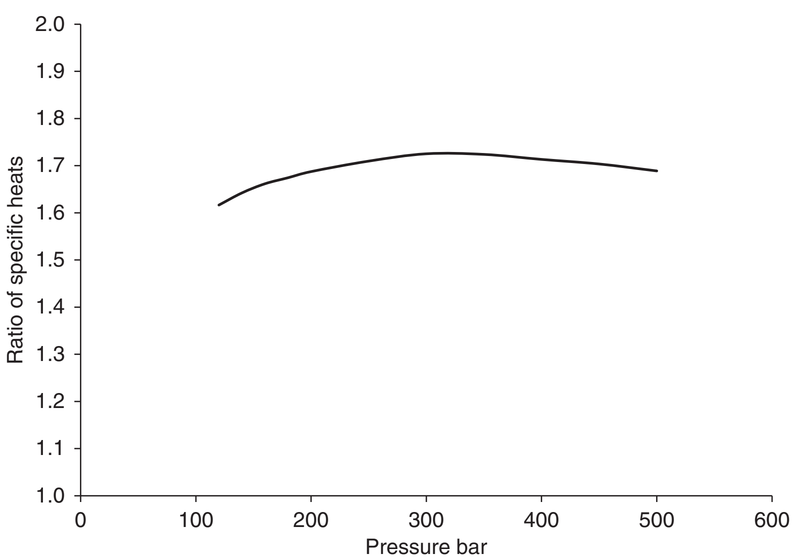

Figure 1.11 Ratio of specific heats for nitrogen at different pressures. *Source:* Din, 1961 [1.3]. Reproduced with permission of Butterworths.

This value is in good agreement with the values shown in Figure 1.11 [1.3]. Figure 1.12 shows the polytropic indexes derived from the measured force/displacement characteristics over a frequency range of 0.001–1.0 Hz; this shows that the index is virtually

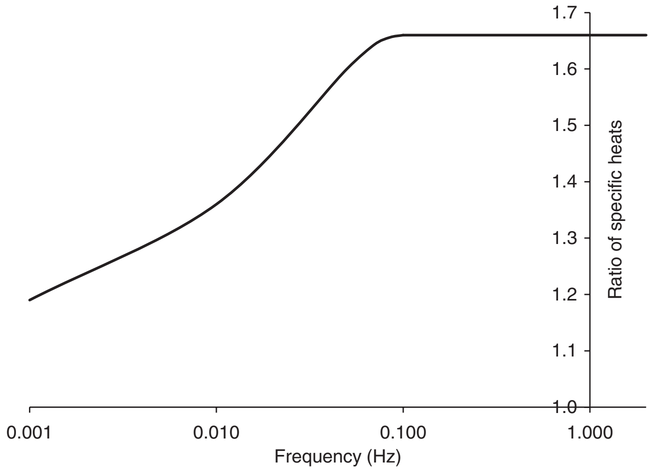

Figure 1.12 Measured ratio of specific heats at different frequencies.

constant at a value of 1.66 down to a frequency of about 0.04 Hz. The value then falls to 1.37 at 0.01 Hz. Even at 0.001 Hz, that is, with a complete cycle time of almost 17 min, the process is not isothermal with an index of 1.15. This means that over the normal working frequency range the suspension operates at near-adiabatic conditions with an index of 1.66. Pressures were also measured inside the gas chamber, yielding a slightly higher polytropic index of 1.69.

1.4.2.2 Damper Characteristic

The damper characteristics of a hydrogas unit can be derived from three measurements:

1) the differential pressure across the damper valve;
2) the wheel force/displacement loop of the unit; and
3) by placing a damper assembly in a suitable flow rig.

1.4.2.3 Differential Pressure Across the Damper Valve

Figure 1.13 shows the wheel load derived from the measured differential pressure across the damper for a total amplitude of 350 mm and at a frequency of 0.9 Hz. The lag in the curves can be ascribed to compressibility of the hydraulic fluid, and friction and inertia of the separator piston. The curves show a near-linear rate with a value of 120 kN (m s^{-1})$^{-1}$ at the wheel. The unit starts to limit or 'blow-off' at a force of approximately 40 kN at the wheel, reaching 50 kN at a wheel speed of 1 m s^{-1}. In the rebound direction the limiting force is shown to be approximately 30 kN. It is important that rebound damping limits at a force less than the static wheel load so that the wheel does not 'hang', particularly so on the front wheel.

1.4.2.4 Force/Displacement Loop

Damping forces derived from the force/displacement loop include the sliding friction component of the unit measured at about ±0.045 of wheel load. The damping rate is in

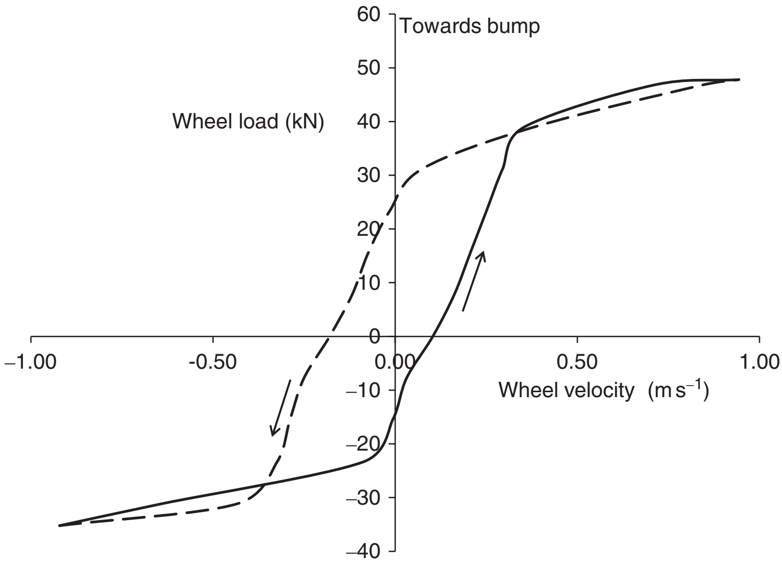

Figure 1.13 Challenger suspension unit damper characteristics from differential pressure across damper valve.

fact too high to be derived from the loop. Maximum force in the bump direction is 50 kN at a wheel speed of 1.0 m s^{-1}, and −40 kN in the rebound direction.

1.4.2.5 Flow Rig

A damper valve was mounted in a hydraulic flow rig with a controllable flow rate. The differential pressure across the damper and the flow rate were measured; Figure 1.14

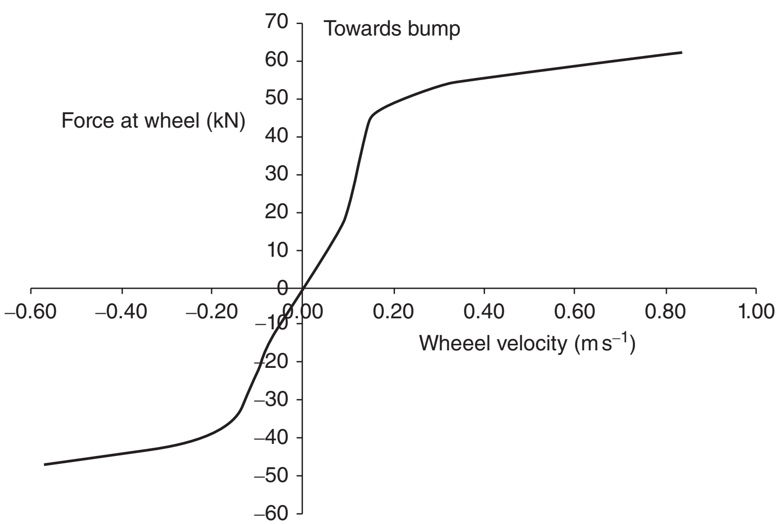

Figure 1.14 Challenger suspension unit damper characteristics from flow rig.

depicts the results. Because the damper rate is controlled by a simple orifice it shows an approximate square-law characteristic. The equivalent linear rate is approximately 220 kN (m s^{-1})$^{-1}$, that is, appreciably more than that given by measuring differential pressure on the suspension unit actuator rig. The unit starts to limit at approximately 50 kN in the bump direction and –40 kN in the rebound direction.

Because the solid rubber tyres on tracked vehicles are comparatively stiff, wheel speeds of up to 10 m s^{-1} in the bump direction have been measured when a wheel comes into contact with large obstacles at high speeds. It is therefore important that the damper valve can allow high flow rates without generating excessively high pressures. The flow rig shows the wheel force only rises to about 80 kN at 10 m s^{-1}.

The overall conclusion is that the damper rate is between 120 and 220 kN (m s^{-1})$^{-1}$ with bump limiting starting at around 50 kN and the rebound at around –40 kN.

1.4.2.6 Suspension Damping of a Multi-Wheeled Vehicle

Consider a six-wheel-per-side vehicle with equal wheel spacing and a damping coefficient of C_w (kN (m s^{-1})$^{-1}$) per wheel. For the Challenger with a 1.0 m wheel spacing, the pitch damping coefficient C_p (kN m (rad s^{-1})$^{-1}$) is defined:

$$C_p = 2 \times 2 \times C_w \times l^2 (0.5^2 + 0.3^2 + 0.1^2) = 32.26\ C_w \tag{1.1}$$

For critical pitch damping C_{pc},

$$C_{pc} = 2 I_p \omega_{pn} \tag{1.2}$$

where I_p is the pitch moment of inertia (kg m^2) and ω_{pn} is the pitch natural frequency (rad s^{-1}). The pitch moment of inertia can be measured by means of a compound pendulum or, if the complete vehicle has been designed by computer-aided design (CAD), then the pitch inertia can be calculated. However, if this information is not available (as in this case) then it can be estimated by considering the vehicle as a uniform rectangle (length × height) and multiplying by a factor, usually taken as 1.15, to allow for the high masses at the ends of the vehicle (frontal armour and power pack). For the Challenger, the pitch moment of inertia I_p is therefore:

$$I_p = \left(\frac{8.3^2 + 2.5^2}{12} \right) \times 61\,000 \times 1.15 = 439\,256 \tag{1.3}$$

The pitch natural frequency has been measured as approximately 1.0 Hz; we therefore have $C_{pc} = 2 \times 439\,256 \times 2\pi = 5520 \times 10^3$ kN m (rad s^{-1})$^{-1}$ and individual wheel critical damping coefficient $C_{wc} = (5520 \times 10^3)/35 = 158$ kN (m s^{-1})$^{-1}$.

This compares with the measured damping coefficient of between 120 and 220 kN (m s^{-1})$^{-1}$, implying that the Challenger suspension is very heavily damped at or near to critical damping. This is confirmed by suspension performance measurements with the vehicle as described in Chapter 3.

1.4.3 Temperature Effects

The ride height of hydrogas suspensions is sensitive to changes in ambient temperature and also temperature rises caused by damper heating; this affects the front suspension

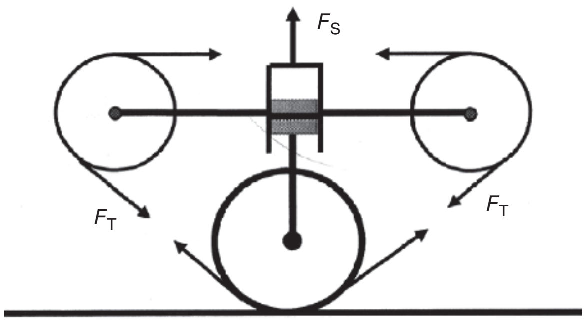

Figure 1.15 Diagram of interaction between hydrogas suspension and track tension.

units in particular, where displacements are greater. The rear units may also be affected by heat soak from the power pack. To calculate the effects of temperature on the Challenger, the units will be assumed to be serviced at 20°C to give a nominal ride height with 100 mm of rebound displacement. If the effects of track tension are ignored, ride height at 50°C increases by 59 mm and available rebound displacement reduces to 41 mm; at –30°C the ride height reduces by almost 100 mm. In practice, the track has a considerable effect on changes in ride height.

Figure 1.15 shows a simplified model of the track system. All the suspension units are merged into one 'super' unit. The suspension displacement is calculated by equating the vertical components of track tension with the suspension force. These are dependent on the linear elasticity of the track, the approach and departure angles, and the deflection of the suspension units which are dependent on the isothermal gas laws. The specific linear stiffness of the Challenger double pitch track has been measured as 17 280 (kN m^{-1})$^{-1}$. The stiffness of a length of track is inversely proportional to its basic stiffness. For a track length of 7.68 m and approach and departure angles of 30°, this gives an effective vertical stiffness of 562.5 kN m^{-1} at sprocket and idler.

Figure 1.16 shows the changes in ride height for the Challenger with and without tracks. With tracks the ride height reduces by 32 mm at –30°C compared to almost 100 mm without tracks. At 50°C the ride increases by 18 mm with tracks, compared to 59 mm without tracks. At –30°C the track tension reduces to 14 kN compared to the normal value of about 50 kN. Damper heating will tend to warm the suspension units when the vehicle starts running, but it would still be desirable to retension the tracks to prevent sprocket jumping. If it is required to restore the suspension to its normal ride height, one possibility is to alter the oil volume of the unit in a similar manner to that used on some Citroen road cars. This system would not be suitable for the Challenger hydrogas unit, however. If oil is bled from the unit to restore the normal ride height at high temperatures, then there is the possibility of the separator piston contacting the damper and depressurising the oil cylinder. At –20°C the gas volume at static is reduced from 2.241 L to 1.935 L. If oil is injected into the unit to restore the ride height, the pressure at 350 mm displacement will be increased to 106.5 MPa compared 56.8 MPa at 20°C.

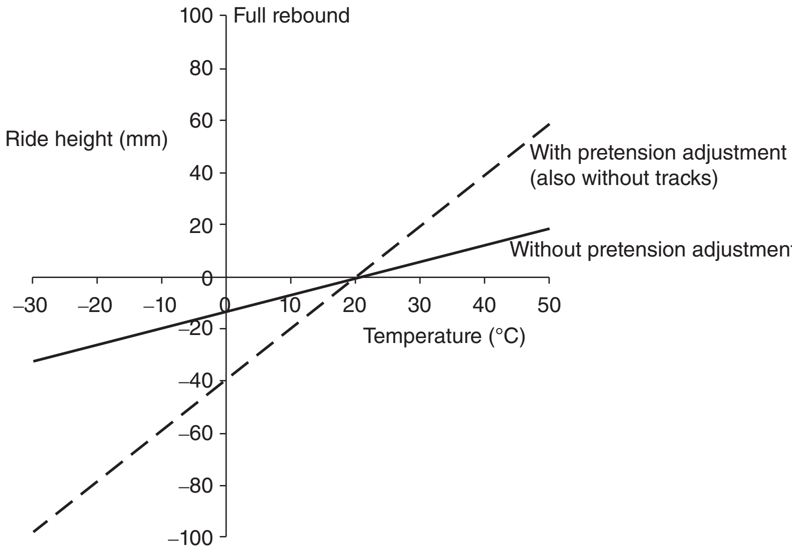

Figure 1.16 Effect of temperature on hydrogas suspension ride height.

If the vehicle was fitted with a compensating idler (see Figure 3.26) then the suspension would tend to respond to temperature changes as if the vehicle was not fitted with a track.

The conclusion is therefore that to restore the ride height to its normal condition, it would be necessary to restore the gas volume of the unit to the value at 20°C. This would normally require the procedure to be carried out as a workshop operation. Two methods of reducing temperature sensitivity and increasing stiffness at the static position are either to use two-stage units or counter-spring units.

1.4.3.1 Two-Stage Units

As well as the thermal effects described above, a further disadvantage of large-displacement single-stage hydrogas suspension units is the comparatively low spring rate around the static position; this can result in comparatively large pitch changes when accelerating and braking or when operating on steep slopes. These effects can be greatly reduced by using two-stage units as shown diagrammatically in Figure 1.17. A smaller gas volume is used around the static position to increase stiffness. As displacement increases, a second gas volume is engaged. The alternative spring curves that can be produced can be analysed with the aid of a spreadsheet. Figure 1.18 shows a possible load/deflection curve for a unit with a 50 kN static load. This demonstrates another benefit of the arrangement, in that the second stage can have a fairly soft rate with a peak load at 350 mm deflection of 154 kN at a pressure of 39.6 mPa. This compares to a peak load of 218 kN for the standard unit at a pressure of 56.3 mPa. The standard unit has a stiffness of 144 kN m^{-1} at the static position compared to 380 kN m^{-1} for the two-stage unit, that is, it is 2.64 times stiffer.

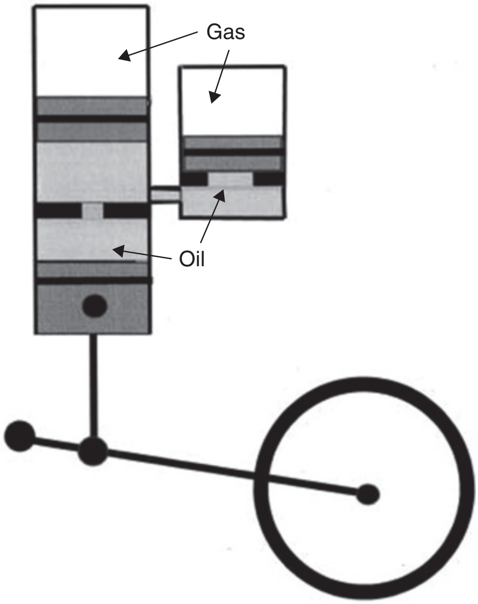

Figure 1.17 Diagram of two-stage hydrogas suspension unit.

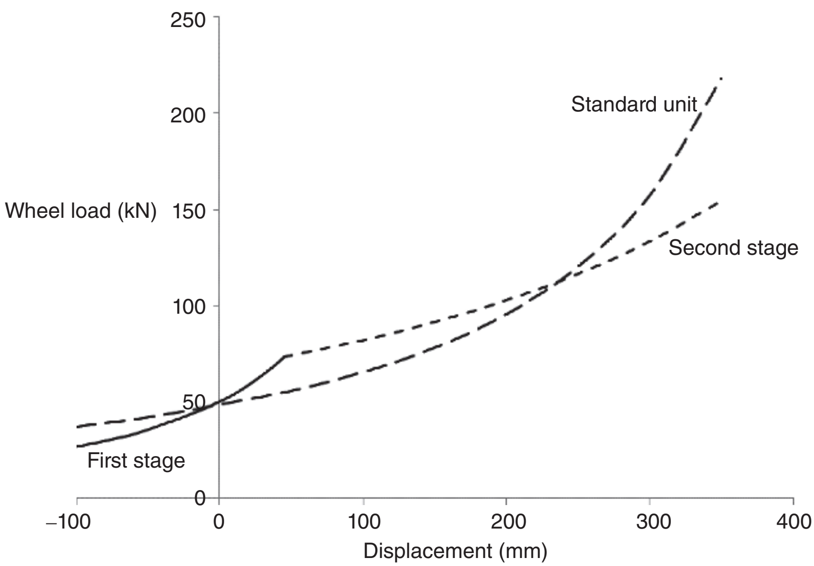

Figure 1.18 Two-stage hydrogas suspension load/deflection characteristic.

At −30° the ride height would reduce by 32 mm and the static track tension would be halved to 25 kN. At 50°C the ride height would increase by 18 mm and the track tension would increase to 64 kN.

1.4.3.2 Counter-Spring Units

Another way of increasing the spring rate around the static position is to use a counter-spring acting in opposition to the hydrogas spring. The spring could be a small metal spring or a small hydrogas unit as shown diagrammatically in Figure 1.19. Again, a simple spreadsheet analysis can be used to assess the different possibilities. Figure 1.20 shows the spring characteristics for a possible hydrogas counter-spring

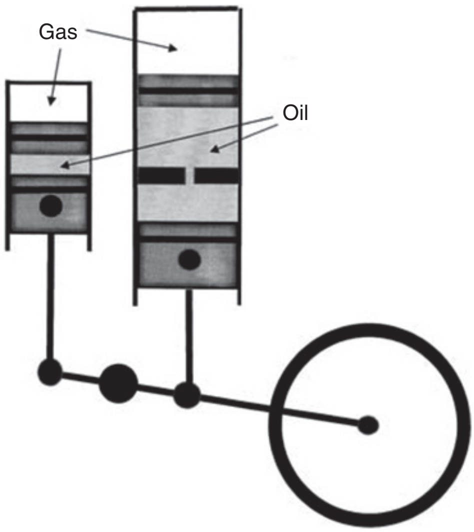

Figure 1.19 Diagram of hydrogas suspension with counter-spring.

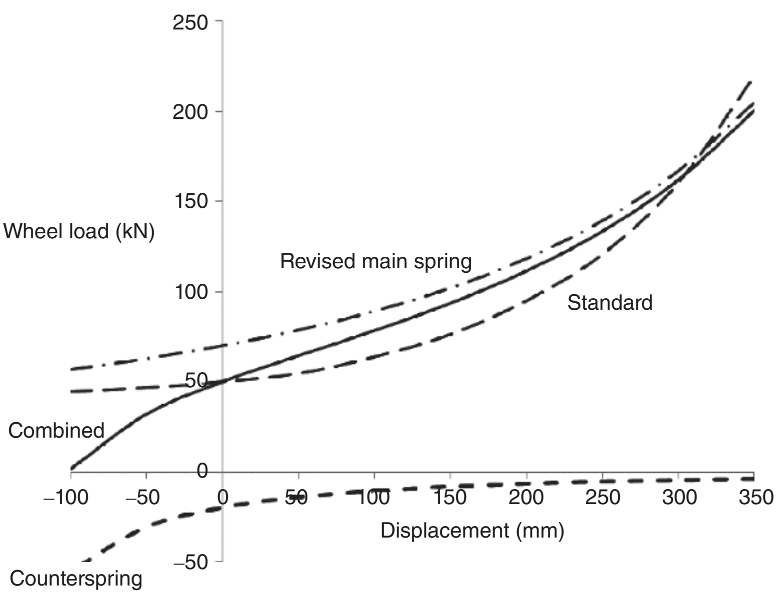

Figure 1.20 Load–deflection characteristic of hydrogas suspension with counter-spring.

arrangement. The stiffness in the static position is 320 kN m^{-1}, which is 2.2 times stiffer than the standard unit. This system is more conveniently arranged as a double-acting telescopic unit with the spring and counter-spring either side of the piston.

1.4.4 Other Types of Hydrogas Suspension

1.4.4.1 Twin-Cylinder Units

The Challenger is fitted with a tapered hull side to reduce the effects of mine damage. However, many armoured vehicles have vertical hull sides to maximise internal hull volume. This limits the space available for externally mounted suspension units. Figure 1.21 shows the unit fitted to the Leclerc MBT. This has two opposed cylinders that enable the unit to be narrower for a given piston area. Other features of the unit are the use of rubber diaphragms to separate the oil and nitrogen, and heat pipes to help carry heat from the damping valves. Each unit weighs about 250 kg.

1.4.4.2 In-Arm Units

Units can be made even narrower by adopting an in-arm arrangement. This is a kinematic inversion of the Challenger arrangement with a modified wheel arm attached to the hull and the axle and wheel attached to a modified oil/gas pressure cylinder as shown diagrammatically in Figure 1.22.

A US-produced in-arm unit is interesting on two counts: (1) it does not use a separator piston between the oil and the gas (an advantage of using a separator piston is that

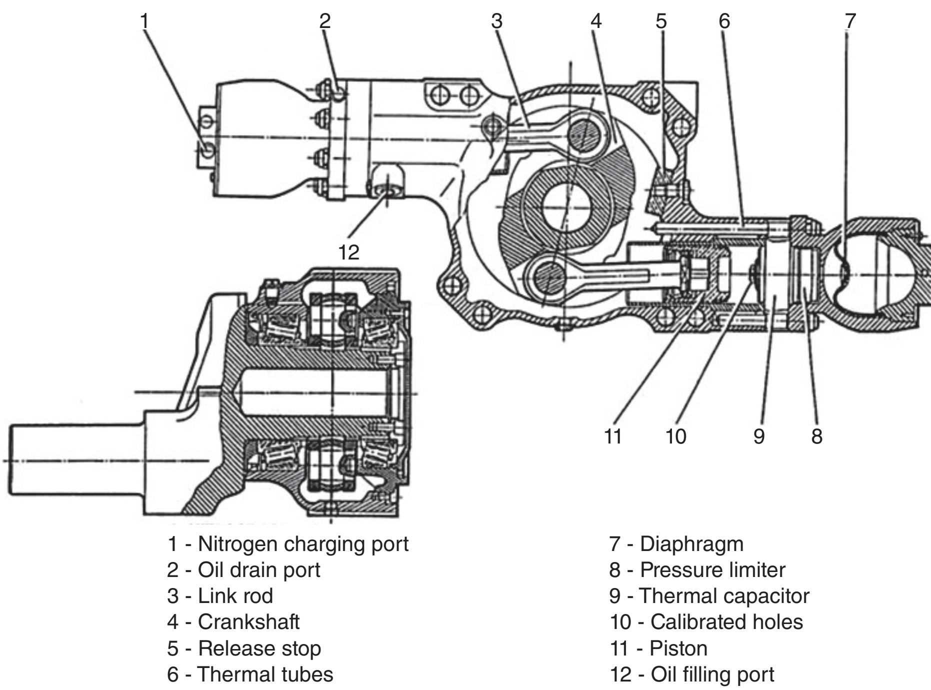

Figure 1.21 Leclerc MBT opposed piston suspension unit. *Source:* Courtesy of Nexter Systems.

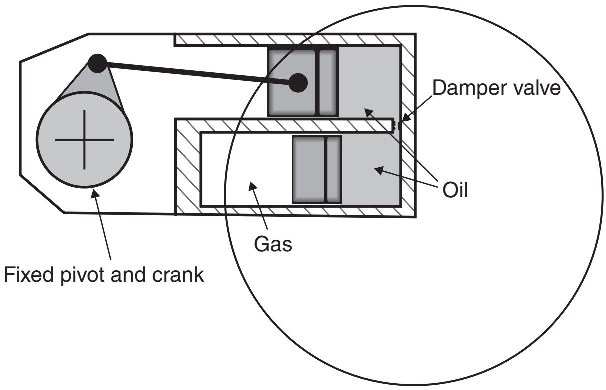

Figure 1.22 Diagram of In-arm suspension unit.

servicing is generally easier because the correct quantities of oil and gas are more readily inserted into the unit); and (2) it uses a variable-force friction damper. The damper is a hydraulically loaded multi-plate friction brake built around the main unit pivot. Hydraulic pressure is derived from suspension movement and generated by a small piston actuated by a cam. The oil feeds through an orifice to provide a velocity-dependent damping characteristic, and a pressure relief valve is used for force limiting. The position of the damper helps dissipate heat into the hull side. It is not thought that this unit proceeded to production, however. The South Korean K2 MBT is fitted with in-arm hydrogas suspension units.

An alternative to in-arm units that are similarly narrow is to use telescopic hydrogas or liquid spring units mounted above the axle arm as shown diagrammatically in Figure 1.23, similar to the arrangement shown in Figure 4.4. Liquid springs are considered in Chapter 4.

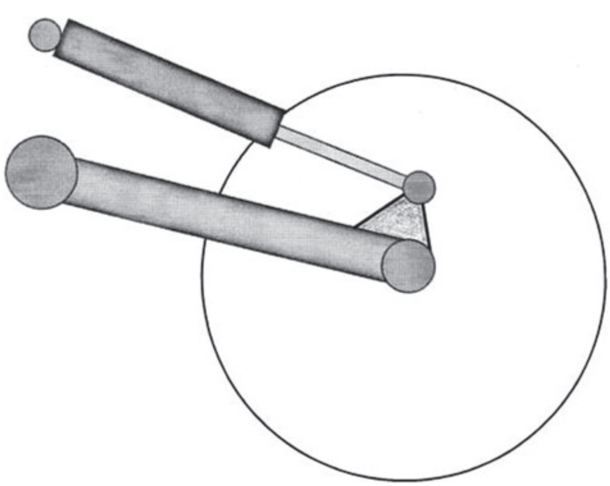

Figure 1.23 Suspension with an external hydrogas or liquid spring strut.

1.5 Dampers

1.5.1 Hydraulic Dampers

Most tracked vehicles use some form of hydraulic damper, either: (1) telescopic; (2) lever-operated opposed piston; (3) rotary vane; or (4) built in to a hydrogas suspension unit.

Telescopic dampers are fitted to the M 113 and Bradley IFV vehicles. Although comparatively simple to manufacture and fit, dissipating heat on severe cross-country terrain is difficult and usually requires the use of special high-temperature hydraulic fluids.

Lever-operated opposed piston dampers are fitted to the Alvis Stormer vehicles. Being hull-mounted, they have good heat-dissipation properties.

Rotary vane dampers can be lever-operated or built in to axle arm pivots, as on the Abrams MBT and Warrior IFV vehicles. Figure 1.24 shows a cross-section of the Horstman unit used on the Warrior vehicle. Arm pivot mounting gives a neat installation with good heat dissipation properties.

1.5.2 Friction Dampers

The Leopard 2 MBT uses friction dampers supplemented by hydraulic bump stops as shown in Figure 1.25. A problem with friction dampers is that if comparatively high values of friction are used, for example a value equivalent to static wheel load, then a lower value would need to be used in the rebound direction to prevent wheel 'hang up'. Further, vibration from the track would be increased, especially noticeable when running on smoother surfaces. As shown in Figure 1.26 the friction dampers on the Leopard 2 have a progressive action with force increasing with suspension displacement. Friction force at the wheel in the static position is 6 kN, about 0.13 times that of

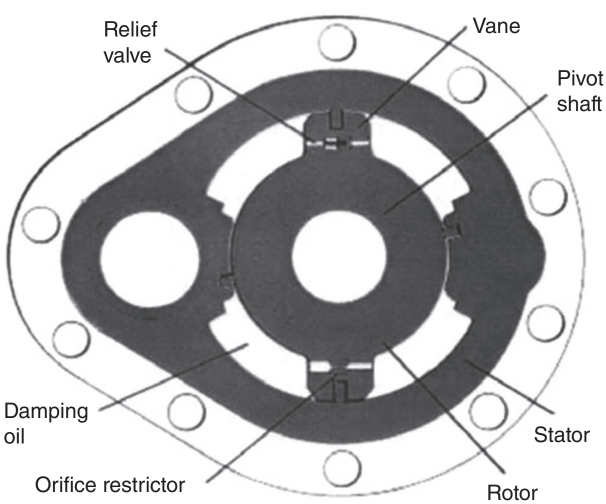

Figure 1.24 Warrior rotary damper cross–section. *Source:* Courtesy of Horstman Defence Systems.

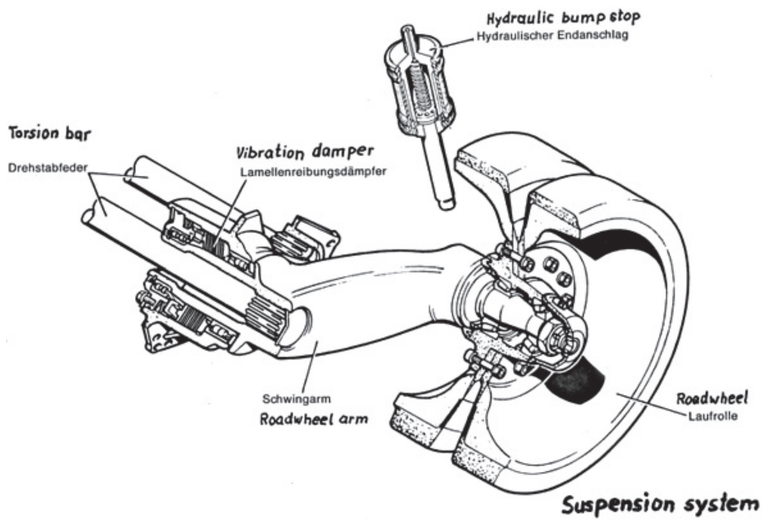

Figure 1.25 Leopard 2 suspension. *Source:* Courtesy of ATZ.

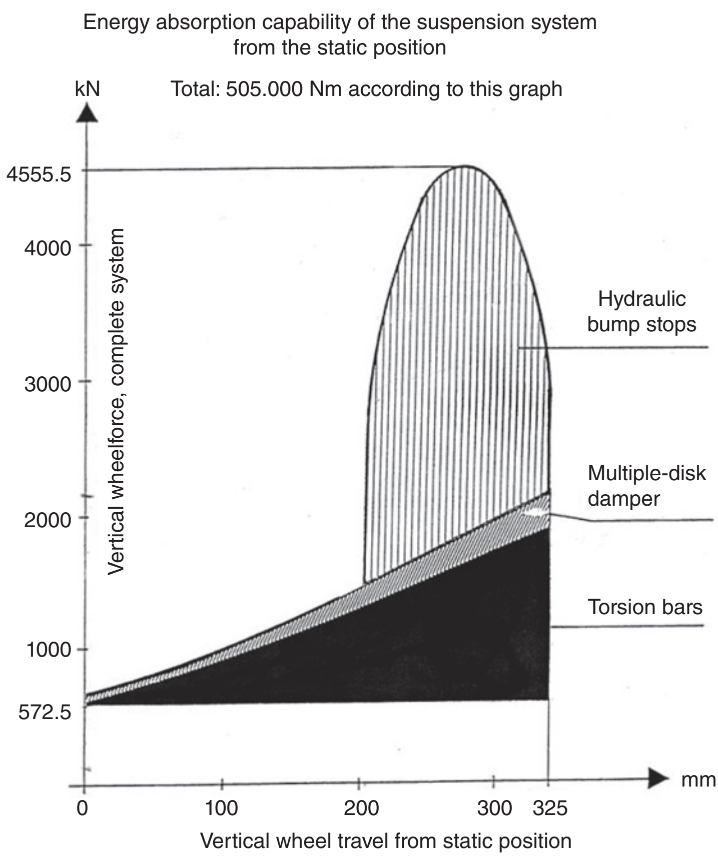

Figure 1.26 Leopard 2 suspension characteristics. *Source:* Courtesy of ATZ.

static wheel load, rising linearly to 26 kN at full bump, about 0.6 times that of static wheel load. The hydraulic bump-stops-come-dampers operate over the last 130 mm of bump displacement and are only effective in the bump direction. They can produce a maximum force of about 180 kN, 4 times static wheel load, at 3 m s^{-1}. In comparison, the limiting (hydraulic) damping force on the Challenger 2 MBT is about 50 kN, that is, static wheel load, but this can operate over the full suspension displacement of 450 mm from rebound to full bump.

Figure 1.26 depicts the Leopard 2 suspension characteristics; somewhat confusingly, these are for the whole vehicle, that is, 14 torsion bars and 10 dampers and bump stops. Only a detailed computer model or experimental measurements on defined profiles could show how this unusual combination of components performs.

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