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Bending fatigue in high-strength fibre ropes

R.E. Hobbs and C.J. Burgoyne

Bending fatigue tests on parallel lay (Parafil) ropes of the aramid fibre Kevlar 49, and helically laid ropes of Kevlar 29 and high-modulus polyethylene (HMPE) are reported. The Parafil ropes were subjected to three different regimes: free bending–tension tests intended to produce failures at the mouth of a termination, sheave bending through 45° under varying axial load, and sheave bending through 180° at constant axial load. The Kevlar 29 and HMPE ropes were tested over a 180° sheave.

Full descriptions of the various procedures are given, and more than 20 separate results presented. Although the free bending–tension tests were intended to produce failures at the mouth of the termination, most failures occurred elsewhere: one test was stopped after one million cycles.The 45° sheave bending tests produced lifetimes of about 4000 cycles when tested at up to 50% of the static breaking strength. The 180° sheave bending test gave lives of 157 cycles at 25% of the static breaking load (SBL) for the Parafil ropes, and 100 to 400 cycles at about 40% SBL for the helically laid Kevlar 29 ropes. The HMPE rope gave nearly 6000 cycles at a load of 40% SBL.

Key words: bending fatigue; fibres; ropes

The applications of high-strength fibres in ropes have expanded rapidly in recent years as manufacturers try to meet the need for very strong tension members in an increasingly demanding market. While the primary purpose of a rope is to carry tension, it is inevitable that some bending will occur in any rope at some stage, perhaps during deployment, or as a secondary effect during use, or as a result of termination-, fairlead-, sheave-, or winch-related factors.

This paper explores three different types of bending phenomena, and some of their fatigue effects on a number of different types of high-specification rope.

The phenomena are:

(a) free bending–tension near a termination where a small cyclic lateral movement of the rope or termination superimposes local bending on the primary tension in the rope,

(b) sheave bending typical of the conditions at a fairlead, where cyclic variations in the axial load cause relatively small movements of the rope over a fairlead sheave with a contact (lap) angle of (say) 45° around the pulley circumference, and

(c) sheave bending typical of the conditions at the first lap of a rope on to a winch drum, or a pulley in a multipart fall, where the rope experiences large movements at virtually constant load, with a lap angle of 180° around the sheave.

The ropes employed were:

(1) type G Parafil in Kevlar 49 with a breaking load of 600 kN,

(2) lubricated Kevlar 29, laid up helically in a 'wire rope' construction (WRC),

(3) unlubricated Kevlar 29, laid up helically in a WRC

(4) a Kevlar 29 multirope construction, and

(5) a high-modulus polyethylene (Techmilon) laid up in a WRC.

Some 20 individual tests have been conducted, and while these by no means form a comprehensive study the results may be of broader interest, as they complement the very limited pre-existing work in this field.

Perhaps the most relevant work prior to that described here is by Karnoski and Liu.[1] They describe some 33 tests mainly on polyester and Kevlar ropes, initially in 25 mm and then in much larger sizes, up to 100 mm in diameter, covering a range of constructions and ratios of tension-to-breaking strength. They used a 180° sheave bending rig (mode (c) above) and sheave-to-rope diameter ratios ranging from 10 to 27.4.

Less pertinently, Markussen *et al*[2] conducted nine cyclic load tests (mode (b) above) on some fairleads, rollers and sheaves at various lap angles. They used No 12 (100 mm diameter) braid-on-braid nylon rope, but its much larger extensibility and different construction again complements rather than duplicates the present study.

Equipment

Fuller details of the apparatus for modes (a) and (b) may be found in Ref. 3.

Mode (a): free bending–tension

Figure 1 shows the general arrangement of the rig. One end of the specimen was restrained by a hold-down bolted to a strong floor, while the other end of the specimen was loaded

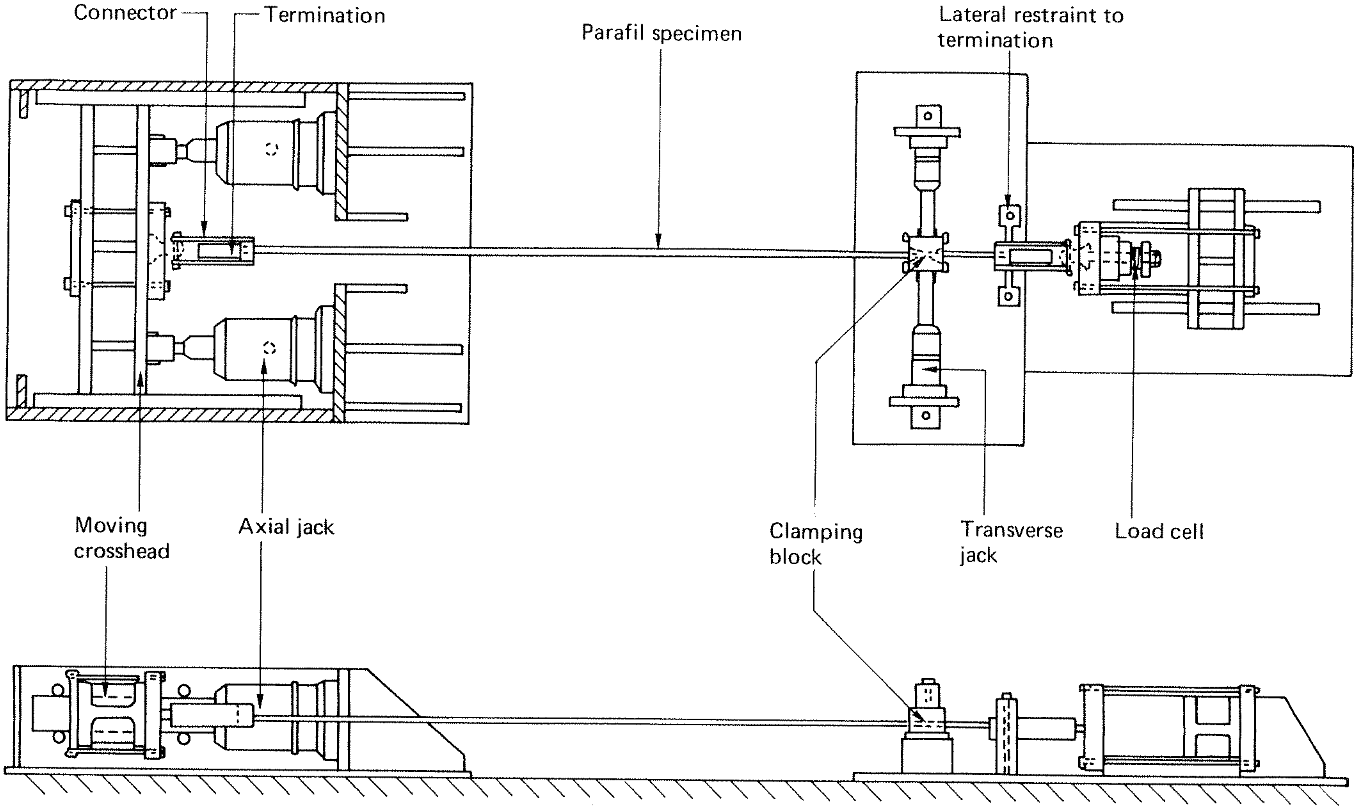

Fig. 1 General arrangement of the free bending–tension test rig

by two 0.5 MN dynamic capacity jacks through a crosshead moving on roller bearings. These jacks reacted against a robust frame also bolted to the floor, and were supplied with oil by a servo-controlled system with a low rate of oil delivery so that only quasistatic loading was possible. The axial load in the cable was monitored at the dead end by a very sensitive 1 MN capacity quartz load washer.

Cyclic lateral movements were imposed on the cable by a pair of 0.1 MN dynamic capacity jacks (Fig. 2) acting on a block clamped on to the cable. This lateral exciter consisted of a light-alloy block, with internal radii to reduce the stress concentrations at the clamping point, which ran horizontally on four ball races on a machined steel table that was clamped to the laboratory floor. A top plate over the ball races prevented the clamp from lifting.

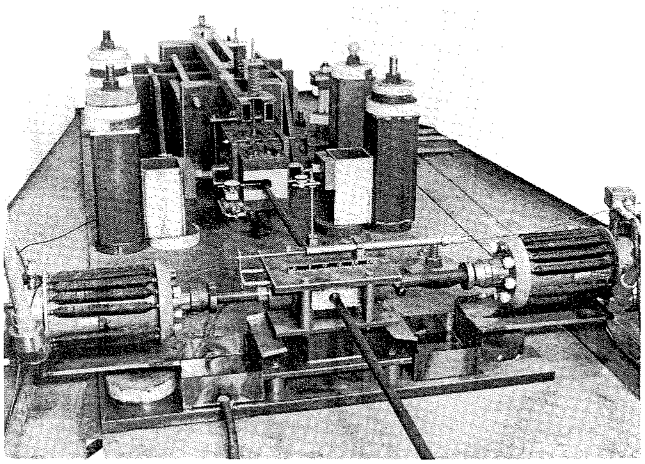

Fig. 2 Transverse jacks and details of the dead end

One jack was supplied with oil from an Amsler Pulsator (flywheel-based) fatigue machine, while the second acted as a constant-force return spring, being connected to an Amsler accumulator, which contained pressurized nitrogen over hydraulic oil and provided a nominally constant pressure.

The movement of the exciter (clamp) in the horizontal plane was continuously monitored by a displacement transducer. The frequency of the oscillation was 4 Hz.

In order that the exciter movement could be converted into a local curvature in the specimen close to a termination, it was necessary to immobilize that termination against lateral movements. At the start of test 1, the exciter was close to the moving crosshead and the termination at the crosshead was restrained laterally to form the test site. This location proved unsatisfactory for a number of reasons discussed below, and the exciter was relocated during test 1 to a position close to the dead end, and the specimen turned round and reclamped against lateral movements at that end. This arrangement was used in all subsequent mode (a) tests.

Mode (b): 45° sheave bending

Figure 3 shows the general arrangement of the sheave bending rig, which provided a 45° wrap angle around a 700 mm diameter steel pulley with a 35 mm diameter semicircular groove. This groove provided a fairly loose fit to the 32 mm diameter polyethylene sheath of the specimen more typical of fibre rope practice than of wire rope practice.

One end of the specimen was connected to a 0.5 MN jack, which was supplied from a servocontrolled hydraulic system. The other end was fixed to a vertical plate supported by six horizontal springs reacting against a fixed vertical plate connected to a strong floor.

This spring set provided a deliberate addition to the axial flexibility of the specimen. This flexibility (100 mm stroke for a load change of 300 kN with six springs, equivalent to

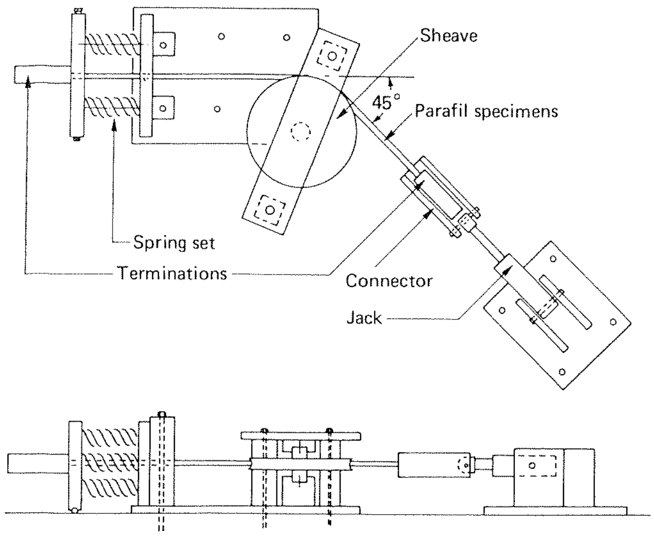

Fig. 3 General arrangement of the 45° sheave bending rig

a 12.2 m length of rope, and pro rata with a smaller number of springs) made it possible to provide the desired movement on the sheave while retaining a short specimen and a compact rig.

Using this arrangement, the load on the specimen was varied cyclically from near zero to full load. Because of the specimen flexibility, a movement of rather more than 100 mm was achieved at the pulley, with an angle of rotation of the sheave of about 16°. The cycling frequency was low (a period of about 17 s/cycle), and was restricted by the limited hydraulic power available rather than by considerations of specimen heating.

Mode (c): 180° sheave bending

Figure 4 shows the general arrangement of the 180° sheave bending rig, which differs from the more familiar 'endless-loop' type of rig because of its alternative use for sheave/rope friction measurements.[4]

Two 0.5 MN jacks with 1200 mm stroke capacities were connected through load cells to the ends of the rope. The end fittings (eye splices and thimbles except for the Parafil ropes, which again used a cone and spike, see below) were prevented from rotating in response to any torque imbalance by adjacent rollers and guide rails. The rope was taken from the first jack, around the sheave and back to the second jack. Different pulley diameters (between 400 and 800 mm) were accommodated by sliding the jacks apart on a cross beam on the jack frame, which was bolted to the strong floor of the laboratory.

The sheave, in turn, was bolted to a substantial disc (provided for friction measurements) carried on a shaft supported in bearings on the base plate and the pulley frame, again connected to the strong floor. In operation the two single-acting jacks were supplied from a servohydraulic system: one jack was kept at the desired constant load while the other jack was operated under displacement control through a second servo channel. By ramping the displacement, an oscillatory motion of the rope over the sheave was provided. In general, rotational movements of some 150° (peak to peak) were used in these trials, sufficient to ensure that the movement at the sheave exceeded one lay length, at least of the helically laid ropes. The movement was made as large as possible, consistent with avoiding any risk of jack bottoming or movements close to 180° at the sheave. The point here is that for movements less than 180°, part of the rope undergoes a straight–bend–straight cycle during one

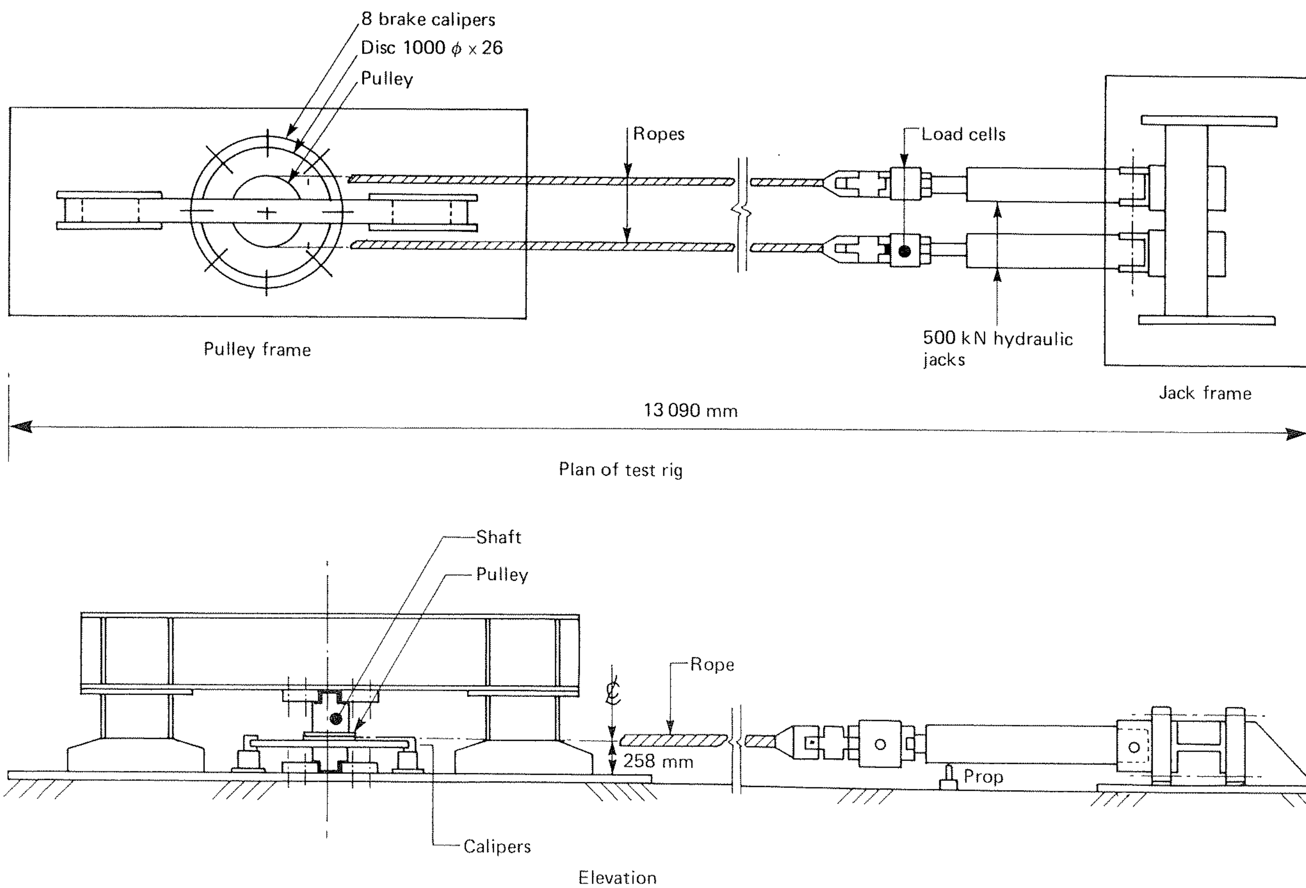

Fig. 4 Schematic of 180° sheave bending rig

complete cycle of pulley movements. For movements in excess of 180°, on the other hand, part of the rope undergoes a straight–bend–straight (off the far side of the sheave)–bend–straight sequence in one complete pulley cycle, which is roughly twice as damaging as movements of less than 180° at the sheave. Thus, to avoid the transition zone around 180° cyclic movement, it is necessary to use say 150° (as here), or 210° or more.

The sheaves used had a throat diameter of 630 mm (700 mm for the Parafil tests) and a 35 mm diameter semi-circular groove (for rope diameters of 32 mm).

Specimens

Table 1 summarizes the five types of rope construction used in the tests described here.

Table 1. Construction details of the types of rope used

Type	Rope diameter (mm)	Rope fibre and makeup	Rated strength (kN)
(1)	27 core	Kevlar 49, Parafil, 310 mm² net area inside black polyethylene cover with outer diameter 32 mm.	600
(2)	32	Kevlar 29/960 (high lubricant). Wire rope construction: 6 outer strands, 8 tpm* S of 27 yarns (15/9/3) 18 000 denier 8 tpm Z. Centre strands: 8 tpm Z, 18 yarns 36 000 denier, 8 tpm S + 3×20 000 denier polyester core yarns.	450
(3)	32	Kevlar 29/964. Wire rope construction: 6 outer strands, 10 tpm S of 27 yarns 17 200 denier 18 tpm Z. Centre strands: 18 tpm Z, 18 yarns 34 200 denier, 10 tpm S + 3 polyester yarns as type (2).	500
(4)	32	Kevlar 29/960. Multirope construction: 5 S twist and 5 Z twist core ropes (each of 3 strands) 2 separate plaited covers.	480
(5)	32	Techmilon HMPE. Construction as type (2) (6/27 + fibre core).	500

*tpm, turns/metre

The Parafil ropes were terminated by means of an internal conical spike (Fig. 5), which grips the parallel fibres against an outer conical socket or barrel. This system provides a positive grip on all fibres without overstressing individual yarns and allows the full static strength of the rope to be mobilized.

Rope types (2)–(5) were terminated by forming eye splices, which were supported on small thimbles and connected to the load cells using pins and extension plates.

Results and commentary: free bending–tension tests (mode (a))

Table 2 summarizes the mode (a) 'tension–bending' (TB) tests on rope of type (1).

Test TB1: details

As noted above, the transverse jacks that applied a bending displacement near the rope termination were initially positioned close to the moving crosshead of the rig. This had been acceptable for testing steel spiral strands.[5] Parafil ropes, however, are more extensible and this flexibility caused considerable problems in maintaining the required axial tension and providing effective lateral restraint for the end connector at the same time.

As a result, the transverse jacks were moved close to the dead end of the rig. It was possible to get closer to the dead end with the exciter than had been achieved at the live end (1170 mm from the front face of the socket as compared with 2450 mm previously, in a specimen 6630 mm long between socket faces), because of the different arrangement of the holding down bolts at the two ends. The dead end termination then became the test site. In consequence, a much more rigid transverse support could be provided to the test termination so that the applied curvature could be measured and controlled much more accurately in static and dynamic tests. The maximum applied deflections of the rope were then equivalent to ±1.5° as compared with roughly half this value before. In addition, the axial force transducer, located in the dead end crosshead, was protected from lateral disturbances and functioned with much greater stability. A further advantage of taking the curvature measurements at the fixed end was that the rope was stationary in a direction normal to the axis of the transducer.

The rope had been subjected to 237 000 cycles at an amplitude of ±0.7° prior to the changeover, and then failed after 94 600 cycles of ±1.5° in the new position. Failure occurred at the restrained socket. As with most failures of Parafil ropes, not all fibres failed at exactly the same point; in this case, failure was spread over a length of about 100 mm.

Test TB2: details

In this test failure occurred at the deflector at 69 000 cycles rather than at the termination. It was suspected that while the clamp block held the outer polyethylene sheath of the

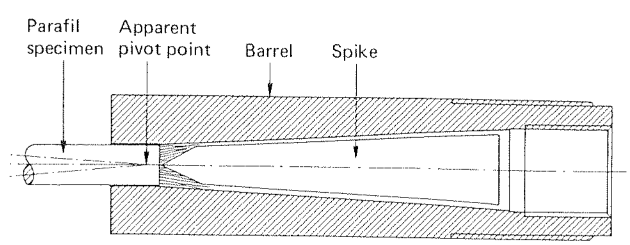

Fig. 5 Termination for Parafil rope

specimen very snugly, the Parafil bundle inside contracted radially under an axial load (as a result of Poisson's ratio or the very shallow helical lay present in the nominally parallel filament bundles) causing a radial clearance between bundle and sheath. The high localized pressures then induced as the deflector moved from side to side, coupled with a bending angle some 20% greater than at the termination, could have led to failure, in spite of the large internal radius of curvature (500 mm) of the clamp block.

Additionally, there may well be some interfibre fretting as fibres on the outside of the bundle try to take a shorter path as the rope is displaced. The individual fibres within the rope do not completely fill the cross section, and there is about 10% more space than would be required for perfect packing of cylinders. Thus, individual fibres may be able to move within the bundle, even if the outer sheath remains perfectly round. At the termination, every fibre is held by the spike in its original position relative to its neighbours, so the potential for such fretting is much reduced, although not eliminated entirely.

As a result of these considerations, the radius of the deflecting block was doubled (to 1000 mm), in an attempt to reduce the damage to the specimen at the deflector in subsequent tests.

Test TB3: details

As noted in Table 2, the increased radius of curvature led to a significant increase in the fatigue life in this test. However, failure again occurred at the deflector rather than at the clamp. Time did not permit further modifications to the exciter (although a number of ideas were debated).

Test TB4: details

In this test, at a reduced axial load, the specimen reached 1 million cycles, and rather than run the test until failure it was decided to check the residual axial strength of the cable. This was found to be 0.53 MN, with failure again at the deflector, as compared with a mean value of 0.597 MN observed in short-term static tests on similar ropes.[6] These static tests were carried out using an identical rope and the same terminations, with a length between terminals of about 7.5 m; the results are thus directly comparable with the present tests.

Table 2. Free bending–tension tests

Test	Preload (kN)	Amplitude (±deg)*	Cycles	Comments
TB1	300	0.7 then 1.5	237 000 94 600	Live end test site. Dead end test site. Failure at socket.
TB2	300	1.5	69 000	Failure at deflector. Radius of deflector then increased from 500 mm to 1000 mm.
TB3	300	1.5	236 000	Failure at deflector.
TB4	200	1.5	1000 000	No failure in fatigue test. Residual tensile strength 0.530 MN, failure at deflector.
TB5	400	0.75	473 000	Failure at deflector.

* That is, half-range

Test TB5: details

In this test at a higher axial load (400 kN) and a reduced angular movement (±0.75°) failure occurred yet again at the deflector.

Temperature measurements

During these tests the temperature on the surface of the rope was measured by thermocouples at:

(i) the mouth of the termination,
(ii) half way between the termination and the deflector,
(iii) the deflector,
(iv) a point far from the influence of the deflector.

As a typical example, for test TB3 at an ambient temperature of 22 °C the temperature rises found at these four locations were 12 °C, 10 °C, 2 °C and 0 °C, respectively.

In spite of the high test frequency and rigorous test regime, these increases were rather small: similar tests on wire ropes[7] rapidly developed temperature rises approaching 80 °C at the termination. The small increase at the deflector may be explained by the higher thermal conductivity of the aluminium deflecting block, as compared with steel. The deflecting block makes contact with a considerable length of the rope at the extremes of each cycle, whereas the body of the termination is in close contact with the sheath over a much shorter length of the mouth of the socket.

With the lateral exciter placed asymmetrically, the deflection angles in the two ends of the rope are different. There will thus be a higher shear strain between the exciter and the dead end than between the exciter and the live end. This will lead to a higher axial movement between the fibres in that part of the rope, and increased heating, as was observed.

Bending stiffness measurements

Several attempts were made to obtain a bending stiffness of the cable. In principle, this can be done by tensioning the cable, deflecting it at a known distance from the socket and deducing the curvature in the cable at the termination by measuring offsets from the undeformed position at several points along the rope.

In order to measure the curvature accurately a special device was designed that made it possible to eliminate rigid body movements.

In each of several sets of measurements it was found that the cable had no significant curvature as it left the termination, so that any curvature took place entirely in the neck of the socket. The straight line measured intersected the termination centreline, typically 13 mm inside the termination. For an angular offset of 2.0°, the maximum static value used, simple geometry suggests that even if the curvature started immediately inside the socket neck, the radius of curvature was only 372 mm, and it is likely to have been significantly less than this.

Tie beam theory[5] suggests that the axial preload, T, the radius of curvature, ρ, the angular offset in radians, θ, and the effective second moment of the area, I, are related by

$$I = (T/E)(\rho\theta)^2 \qquad (1)$$

Taking 372 mm as an upper bound on ρ, the effective second moment of area was only 400 mm^4, or about 1% of that of a solid circular rod of the same diameter. Thus, in spite of the axial preload, the Parafil bundle was functioning very much as a bundle of discrete fibres able to slide freely over one another rather than as a solid rod. This freedom to slide also implies very small bending stresses in the individual fibres rather than the large axial stresses that would appear in the extreme fibres if plane sections were remaining plane across the bundle.

These low stresses are one factor in the unexpectedly high resistance to free bending–tension fatigue at the termination. Another pertinent factor is the relatively small fretting movements that occur in bending as compared with those found in a purely axial (tension–tension) fatigue situation. In tension–tension fatigue, failures have been found by other authors[8] to initiate from fretting on the nose of the spike (Fig. 5). In bending, such movements on the spike are less than those that occur at the extreme fibres, and (of course) are concentrated at the two points on the spike at the greatest distance from the neutral axis rather than taking place uniformly all round the spike.

Results and commentary: 45° fairlead bending (mode (b))

Table 3 summarizes the 45° sheave bending tests, while Fig. 6 shows the results in graphical form. Because of the high lives achieved in tests FB4–6, a test with a load range of 150 kN, which had been included in the original programme, was not run. All ropes failed at one of the points where the rope ran on to the sheave during each cycle. It is thus presumed that failure was caused by interfibre fretting, which must be highest in these regions.

The nominal radial pressure between sheave and specimen is given by

$$p = 2T/dD \qquad (2)$$

where p is the pressure, T the axial load, and d and D are the groove and sheave diameters, respectively. For a 300 kN axial load, this pressure was close to 30 N mm^{-2}. In spite of this high value, the lives to failure were certainly greater than some of the more pessimistic guesses made before testing started.

Table 3. Sheave bending tests at 45°

Test	Load range* (kN)	Number of cycles to failure	Comment
FB1	300	4 000	In all tests failure occurred at the sheave.
FB2	300	4 550	
FB3	300	4 045	
FB4	200	28 000	
FB5	200	13 100	
FB6	200	23 250	

*With a minimum of about 1.5 kN in all cases

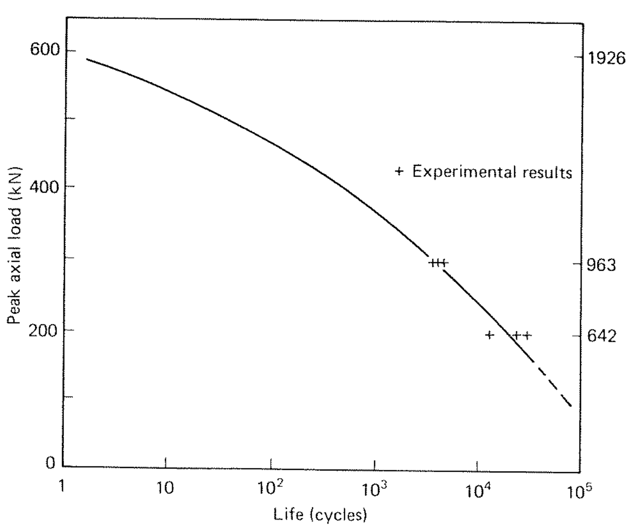

Fig. 6 Results for 45° sheave bending tests

Results and commentary: 180° sheave bending (mode (c))

Table 4 summarizes the 180° sheave bending tests while Fig. 7 presents the results.

Rope stretch was monitored during the fatigue tests. After the first few cycles, the extension remained sensibly constant until a sudden increase just before final failure. Incipient failure was detectable in the last few cycles by ear and in the form of lumps in the individual strands of the helically laid ropes, lumps thought to be associated with the kinking of failed or partially failed strands going into compression.

As would have been expected, the Parafil ropes had the shortest lives, since the parallel lay does not permit the

Table 4. Sheave bending tests at 180°

Rope type	Details	Mean axial load (kN)	Mean axial load % of rated	Life to failure (cycles)
(1)	Kevlar 49 Parafil	150	25	157
		200	33	127
(2)	Kevlar 29/960 WRC	150	33	666
		200	44	300
		225	50	196
(3)	Kevlar 29/964 WRC	150	30	672
		200	40	392
		300	60	17
(4)	Kevlar 29/960 Multirope	150	31	414
		200	42	78
(5)	Techmilon HMPE WRC	200	40	5862
		300	60	1063

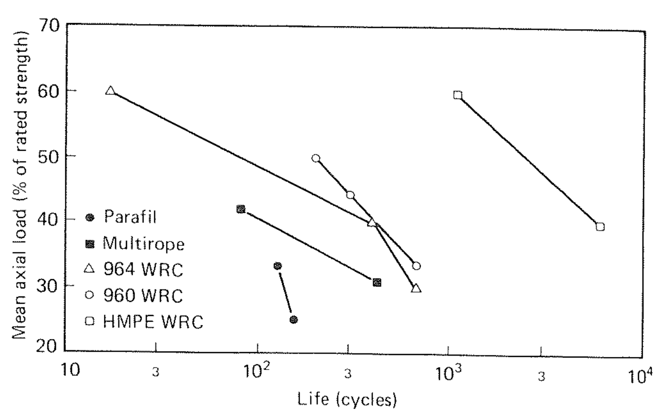

Fig. 7 Results for 180° sheave bending tests

elements on the outside of the curve around the pulley to shed load in the way that is available to the elements of a helical rope structure. By the same token, the multirope specimens had the next shortest lives: this rope consists of a number of parallel subropes within an outer sheath and although these individual ropes are helically laid, because they are parallel the load will also be shared unequally between them as the rope passes over the sheave.

The wire rope constructions, ropes (2) and (3), had significantly better performances, reflecting their fully helical lay.

Finally, rope (5) in a high-modulus polyethylene but otherwise identical with rope (2), outperformed it by a factor of nearly 10 on life, emphasizing the much greater resistance of HMPE to abrasion, fibre bending and transverse pressure. In applications where its poorer resistance to creep was unimportant, this fibre would be of considerable interest. Between the other four constructions, choice would (as always) be a question of compromise and balance, weighing axial stiffness and fatigue resistance against the available bending performance. For example, no one would use Parafil primarily for its bending resistance, but its high axial stiffness is combined with a bending capability that would be acceptable for certain applications.

Conclusions

In free bending–tension fatigue, the area adjacent to the Parafil termination proved to be less vulnerable than had been expected, and indeed was less vulnerable than the area within a carefully radiused deflector clamp which caused an angular offset only a little larger than that taking place at the termination. With a preload of 300 kN on a rope with an ultimate load of 600 kN, a life in excess of 240 000 cycles can be expected at an offset of ±1.5° (a range of 3°) at the termination.

In 45° sheave bending fatigue, lives of around 4000 cycles were obtained with an axial load range of 300 kN (that is, 50% of the ultimate load of 600 kN), and about 20 000 cycles with a load range of 200 kN, using a pulley with a D/d ratio of 22.

Under a 180° sheave bending fatigue, lives at about one third of the static break load ranged from 127 cycles (Parafil) to an estimate of 10 000 cycles for the HMPE rope. Even the Parafil result is, in fact, quite encouraging, given the onerous nature of the test: a pulley with a D/d ratio of 22 and the absence of any helical lay in the rope to accommodate the length difference around the sheave.

Acknowledgements

The work described in this paper is a synthesis of results from a number of separate programmes. The free bending–tension and 45° sheave bending studies were undertaken as part of an examination of the potential use of Parafil in mooring systems for offshore oil rigs which was coordinated by Advanced Production Technology Ltd on behalf of a consortium of industrial sponsors. Their support, and the support of the MoD Procurement Executive for the 180° sheave bending work on the non-Parafil ropes, is gratefully acknowledged.

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Finally, it should be noted that Kevlar is a trade name of E I DuPont de Nemours Inc., that Parafil is a trade name of Linear Composites Ltd, and that Techmilon is a trade name of Mitsui Petrochemicals.

References

1. **Karnoski, S.R. and Liu, F.C.** 'Tension and bending fatigue test results of synthetic ropes' *Proc of 20th OTC, Houston, Texas, 1988* paper OTC 5720 pp 343–350
2. **Markussen, B.H., Hunt, E.S. and Hobbs, R.E.** 'Wear of nylon hawsers over rollers, pulleys and fairleads' *Proc of 16th OTC, Houston, Texas, 1984* paper OTC 4765 pp 463–470
3. **Burgoyne, C.J., Hobbs, R.E. and Strzemiecki, J.P.** 'Tension–bending and sheave bending fatigue of parallel lay aramid ropes' *Proc of 8th Int Conf on Offshore Mechanics and Arctic Engineering* (ASME, New York, 1989) Vol 1 pp 691–698
4. **Nabijou, S. and Hobbs, R.E.** 'Frictional behaviour and fatigue performance of wire ropes bent over small diameter sheaves' *CESLIC Report SC5* (Department of Civil Engineering, Imperial College, London, 1988)
5. **Hobbs, R.E. and Smith, B.W.** 'The fatigue performance of socketed terminations to structural strands' *Proc Inst Civ Eng Part 2* **75** (1983) pp 35–48
6. **Chambers, J.J.** 'Parallel lay aramid ropes for use as tendons in prestressed concrete' *PhD thesis* (University of London, 1986)
7. **Strzemiecki, J.P. and Hobbs, R.E.** 'Properties of wire ropes under various fatigue loadings' *CESLIC Report SC6* (Department of Civil Engineering, Imperial College, London, 1988)
8. **Crawford, H. and McTernan, L.M.** ''Fatigue' properties of Parafil' *Proc Symposium on Engineering Applications of Parafil Ropes, Imperial College, London, 1988* Ed C.J. Burgoyne pp 29–38

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