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Application of finite element modelling to winch drum design

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SYNOPSIS

The value to designers of taking proper account of rope-load relaxation in heavily loaded winch barrels has been emphasised by the authors in another recent paper.[1] Hitherto most analyses leading to barrel design have depended upon classical strength of materials. Finite element (FE) methods have tended to be restricted to the drum and flange design once the nett rope load is known. Here the authors explain a modelling procedure which takes the rope into the FE analysis accounting for rope-load relaxation using axi-symmetric orthotropic elements and a temperature-loading technique.

INTRODUCTION

In a previous paper[1] the authors have drawn the attention of winch designers to the advantages of taking account of rope-load relaxation in multi-layer applications. The calculation procedures suggested by Egawa & Taneda[2] and by Torrance[3] were reviewed and recommendations were made in favour of the former. The present authors' technique of reiterating the calculation to allow for the non-linearity of lateral modulus of elasticity of wire rope was discussed and a sample calculation, based upon a typical heavy winch, served to highlight the differences in nett drum load resulting from simple and more complex treatment. It was shown that the lateral compliance of the wire rope itself dominates the design calculation which is relatively insensitive to radial compliance of the drum. Rope-load relaxation, it was found, is largely a function of the elastic properties of the wire rope. Emphasis was placed upon determining these properties as a precursor to the design procedure. The reader might usefully regard the present paper as an extension of the author's earlier publication where mechanisms of rope-load relaxation were dealt with in detail. Also some of the descriptive material which would have been included here, had this paper been totally free-standing, will be found in ref 1, together with a clarification of relevant definitions.

Dr Maw is Principal Lecturer in Applied Mechanics in the Mechanical Engineering Department of Sunderland Polytechnic. His present research interests include design and performance of heavy duty winches, cable lift systems, machinery health monitoring and robotics in manufacture. Miss Karbalai graduated from Sunderland Polytechnic with a BSc in Mechanical Engineering in 1983. For the next 3 years she followed an approved research investigation into the design of winches with particular reference to multi-layering. This work, now completed, is being written up for a PhD and caught the attention of NEI–Clarke Chapman Ltd who appointed her as a Design Engineer in September 1986. Miss Karbalai is now employed by Butler–Newall Machine Tool Co as Industrial Engineering Manager.

DRUM DESIGN

Once the radial load on the drum has been predicted its thickness is calculated typically by adopting the 'thin' or 'thick' relationships for a long unsupported shell, using the hoop stress as the design criterion. Flanges and internal supports, it is argued, add strength to the shell so the designer can, with confidence, use low factors of safety in determining the drum thickness.

This classical mechanics treatment of the design of the drum shell has been effective. The authors can find no report of shell failure by elastic instability (buckling) though they are aware of isolated cases of other modes of failure. For example, flanges bursting away from the shell due to rope wedging action, bearing failure resulting from shell elongation and others attributable to very high loading. However, as the rope loads and the number of layers increase, the design of the barrel assembly becomes more critical. It is not sufficient to deal with the shell design in isolation because the stresses induced by welded features, like flanges and supports, redistribute those produced by the rope compressive load. High stress concentrations can occur in parts of the unit. It is found that the classical mechanics approach becomes mathematically intractable as features are added to the shell so, whilst still providing a useful first estimate of sizes, for very heavy duty applications, there is little alternative to a numerical (eg finite element) treatment. This has been recognised in design offices and FE techniques are currently used. The purpose of this paper though is to outline a technique which takes rope-load relaxation into the FE analysis thus optimising on the barrel cost and weight.

The authors used the 'PAFEC' FE package and they acknowledge the help given to them in modifying the software slightly to allow access to the backing store.

FE MODELLING

In the usual method of modelling, by finite elements, the rope pressure load on a winch drum applies a series of radial concentrated forces round the barrel circumference. This ring of discrete forces acts on the drum surface at the centre of pressure of the coil being modelled. In order to minimise local distortion due to stress concentration a large number of forces is used. The magnitude of these forces is calculated from knowledge of the winding tension (often without taking account of load relaxation) so that the sum of the discrete loads has the same effect as that which would be computed from the radial pressure of the rope. Moreover, every successive ring of forces, corresponding to every rope coil, is regarded as having the same magnitude. No account is taken of the axial redistribution of forces as the barrel distorts non-uniformly. Naturally, the centre sections of the barrel deflect radially more than the ends so a small re-distribution might be expected.

The flange load is modelled in a similar way using knowledge of the nett rope load in each layer and remembering that the ropes lie in grooves formed by sub-layers. When each layer load is evaluated its wedging effect on the flange can be modelled by resolving it into a ring of axial discrete loads acting at the contact circle of the layer and the flange. Odd numbers of layers thrust against the flange so the accumulated effect of the whole rope mass on the flanges is modelled as a series of concentric rings, the first acting at the contact circle of layer 1, the second at layer 3, etc.

This modelling strategy for barrel and flange loading, though widely used, is considered to be insufficiently exact and an improved model is suggested which will:

1. provide a better representation of the uniform radial pressure due to each coil;
2. allow for rope-load relaxation in multi-layer applications;
3. be self-adjustable so that it can accommodate axial redistribution of radial pressure as deflections occur;
4. give a good estimate of flange loading and stresses (especially at the root);
5. use axi-symmetric elements.

The model devised to achieve all of the above conditions has, of course, now to include a model of the rope as well as the barrel assembly.

The first rope layer is represented by a series of contiguous rings placed along the barrel, each ring modelling one coil of rope, and dimensioned such that the centre of its cross-section [and therefore its PCD (pitch circle diameter)] lays on that of the rope. It is important to maintain this coincidence of PCD under load, so the material properties of the rings must be calculated from the rope properties to ensure that the rings behave elastically under load in exactly the same manner (in all directions) as the rope itself. Due to the nature of the rope properties the ring has to be orthotropic in character. Details of the modelling are given in the Appendix.

Each ring provides external pressure to the drum by lowering its temperature in the hoop direction only. The procedure is to guess a value for the temperature drop for all rings and use this on a pre-run.

On completion the solution, in terms of the final tensile load in the rings, is examined. This is compared with the winding tension in the rope. Where there is a discrepancy a simple proportional change is made to the temperature drop for that ring and the computation is then reiterated. A fixed tolerance can be specified on the tensile load in each ring and, once this is satisfied, the loading and deflections in the whole drum/rope system for one layer is deemed to have been solved. All deflections are stored for later retrieval. This is necessary because the deflections in the rope are large enough to re-arrange the geometry of the system.

The geometric relation between the rope and ring of the first layer is different to that of the rope and ring in the second and upper layers. This is because the first layer rope lies on a smooth cylinder whereas the second and other layers lie in grooves formed by adjacent coils in sub-layers. Consequently the rings for layer 2 and outwards have to be re-modelled. This completed, the second set of rings (cylinders) is placed over the distorted shape of the first set, each ring riding loosely on its inner neighbour. Because of axial variation in radial distortion in the first layer, the rings of the second layer are not necessarily all the same diameter so new axi-symmetric separate elements, one coil deep, have to be set up on the deflected shape of the first layer. This difficulty was overcome, on the specific package used, by coupling to imaginary mass elements at the outer nodes of the first layer and then 'repeating freedoms'. All mass elements were set at zero mass magnitude.

A preliminary solution for the change of stress due to the addition of the new set of rings is then obtained by lowering their temperature in the hoop direction. Applying once again the criterion for correct temperature drop, each outer ring is checked to see whether or not its final load is within the set tolerance on the winding tension. If not the solution is reiterated. Once the tensions in the rings of the second layer are satisfied the second layer loading is regarded as having been solved.

It is important to emphasise that the stresses induced in the sub-structure by second layer loading are not the total stresses. They are the changes of stress consequent upon adding the second layer. The total stresses are calculated by adding algebraically the new stress changes to the residual stresses induced by the first layer.

This process of incremental loading and stress up-dating is continued until all layers are accounted for.

Details of rope modelling strategy

Each coil of rope is to be modelled as a ring and the criteria for modelling are:

1. the centroids of the cylindrical ring and rope cross-sections should always be the same distance from the winch drum axis whether under load or not;
2. the tensile loads in the ring and rope should at all times be the same;
3. the radial (crushing) load on the ring should always be the same as the radial component of the crushing load on the rope;
4. the radial and hoop deflections of the ring and rope should be the same. (This partly re-states item 1 .)

Crushing stress is assumed to be constant across the rope cross-section. This is the simplest of a set of assumptions that can be made about lateral stress distribution and it is only justifiable if the FE results can be calibrated against an independent calculation. The assumption has however been employed to useful effect elsewhere.[2]

Modelling of all layers except the first

Fig 1 shows a typical rope cross-section with a ring, its model, superimposed. To maintain the contiguity of the ring with its neighbour, and to set its PCD at that of the rope, the cylinder has a width of d and a depth of $d \sin \gamma$.

The modelling procedure is simplified if the ring and rope are assumed to be on a drum of infinite radius, ie they are straight and each is of length δL. Tensile and pressure loads can then be treated separately and superimposed. It is appreciated that this does not correspond properly with the actual loading (if the radius is infinity the pressure forces are zero) but that correspondence is not necessary for the modelling.

The modelling equations are derived in detail for all layers except the first in the Appendix where the nomenclature can also be found. Detailed derivation for the first layer modelling is omitted.

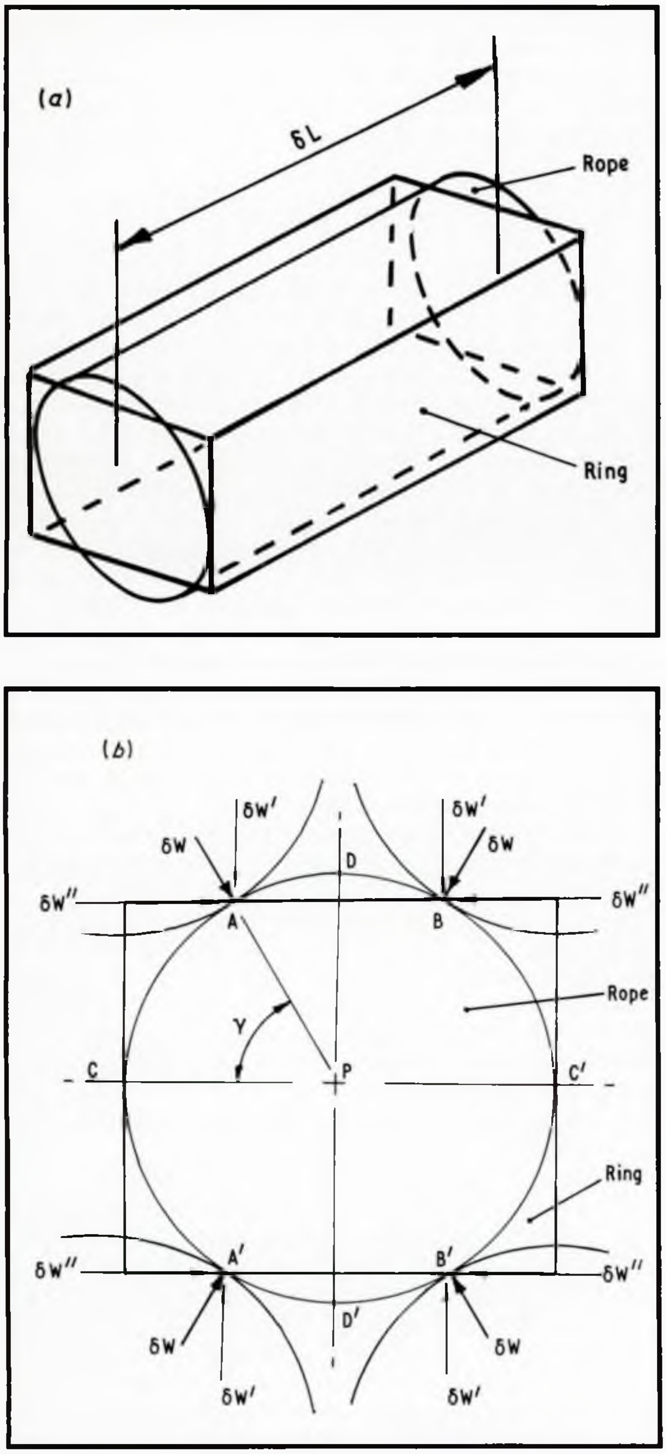

Fig 1: (a) The rope and (b) its ring model for all layers except the first

Modelling equations, all layers except the first

Accounting for radial changes:

$$-\upsilon_r\left(\frac{4P}{\pi d E_r}\right)\sin\gamma + \upsilon'_r\left(\frac{2\delta W''}{\delta L\, E'_r}\right)\sin\gamma - \left(\frac{2\delta W'}{\delta L\, E'_r}\right)\sin\gamma = -\left(\frac{2\delta W'}{\delta L\, E'_c}\right)\sin\gamma - \upsilon_c\left(\frac{P}{d\, E_c}\right) \qquad \text{(i)}$$

Accounting for longitudinal changes:

$$\left(\frac{4P}{\pi d^2 E_r}\right) + \upsilon''_r\left(\frac{2\delta W''}{d\,\delta L\, E'_r}\right) + \upsilon''_r\left(\frac{2\delta W'}{d\,\delta L\, E'_r}\right) = \left(\frac{P}{d^2 \sin\gamma\, E_c}\right) + \upsilon''_c\left(\frac{2\delta W''}{d\,\delta L\, E'_c}\right) \qquad \text{(ii)}$$

$$-\upsilon_r\left(\frac{4P}{\pi d E_r}\right) - \left(\frac{2\delta W''}{\delta L\, E'_r}\right) + \upsilon'_r\left(\frac{2\delta W'}{\delta L\, E'_r}\right) = -\upsilon_c\left(\frac{P}{d \sin\gamma\, E_c}\right) + \upsilon'_c\left(\frac{2\delta W'}{\delta L\, E'_c}\right) \qquad \text{(iii)}$$

Modelling equations, the first layer

Accounting for radial changes:

$$(1+\sin\gamma)\left(\frac{-\upsilon_r\, 2P}{\pi d E_r} + \frac{\upsilon'_r\, \delta W''}{2\delta L\, E'_r} + \frac{\delta W'}{\delta L\, E'_r}\right) = (1+\sin\gamma)\left(\frac{-\delta W'}{\delta L\, E'_c}\right) - \upsilon_c\left(\frac{P}{d\, E_c}\right) \qquad \text{(iv)}$$

Accounting for longitudinal changes:

$$\left(\frac{4P}{\pi d^2 E_r}\right) + \upsilon''_r\left(\frac{2\delta W''}{d\,\delta L\, E'_r}\right) + \upsilon''_r\left(\frac{2\delta W'}{d\,\delta L\, E'_r}\right) = \frac{P}{\frac{d^2}{2}(1+\sin\gamma)\, E_c} + \upsilon''_c\left(\frac{2\delta W'}{d\,\delta L\, E'_c}\right) \qquad \text{(v)}$$

$$-\upsilon_r\left(\frac{4P}{\pi d E_r}\right) - \left(\frac{\delta W''}{\delta L\, E'_r}\right) + \upsilon'_r\left(\frac{2\delta W'}{\delta L\, E'_r}\right) = -\upsilon_c\left(\frac{2P}{d\,(1+\sin\gamma)E_c}\right) + \upsilon'_c\left(\frac{2\delta W'}{\delta L\, E_c}\right) \qquad \text{(vi)}$$

The top layer

All layers except the outermost experience a reduction in their original winding tension due to elastic distortion and, although the top layer does crush as it is wound, its distortion does not affect the final drum/rope loading. Consequently its representation on the FE model is unimportant except that it must apply the correct (winding) tension. It is therefore convenient to model it as one of the outer layers using the quoted equations.

There being more modelling parameters than there are equations, some equalities are imposed. The implications of these impositions are reflected in other parameters as these are derived from the equations. Let

$$\upsilon_c = \upsilon_r$$

Also, since changes in rope length due to diametral loading are very small and since they will, in any case, be the same for loading on perpendicular diameters:

$$\upsilon''_r = \upsilon''_c = 0$$

E_r is the manufacturer's quoted value. E'_r, υ_r and υ'_r are determined experimentally.

Implementation of modelling equations

As an example of the use of the modelling equations take the outer penultimate layer. It can be shown that:

$$2P = \sigma d D$$

where σ is the crushing stress. Also

$$\sigma \, d \, \delta L = \delta W$$

$$\text{therefore} \quad \frac{\delta W}{\delta L} = \sigma \, d$$

$$\text{Since} \quad \frac{\delta W'}{\delta W''} = \tan \gamma$$

$$\frac{\delta W'}{\delta L} = \frac{\delta W}{\delta L} \sin 60 = \frac{2P}{D} \sin 60$$

and

$$\frac{\delta W''}{\delta L} = \frac{\delta W}{\delta L} \cos 60 = \frac{2P}{D} \cos 60$$

By eqn (ii), since $\quad \upsilon''_c = \upsilon''_r = 0$

$$E_c = \frac{\pi}{4 \sin \gamma} E_r$$

E'_c is found from eqn (i) and υ'_c from eqn (iii).

Calibration

The applicability of the modelling procedure was tested by treating a simple shrink-fit problem in which a wire rope of construction: 5 mm diameter (7 x 19), was wrapped in one layer round a drum of diameter 220 mm. The rope was modelled into a series of contiguous rings for the FE solution. Each ring was defined as one axi-symmetric element and the drum was modelled also as a series of single, solid axi-symmetric elements of length equal to the rope pitch. A reduction in temperature of, for example, 288°C was found by using the coefficient of expansion for steel to be equivalent to an interference fit between rope and drum of 0.697 mm. Solving the problem by normal methods gave a rope load of 2753 N whereas by FE analysis the rope load was 2554 N. The discrepancy in the FE solution is 7.2%

The error value in the calibration was considered to be sufficiently small to encourage advancement to a check of the modelling and FE technique for an unsupported shell, analysed also for comparison by the Egawa & Taneda[2] method. The rope and drum are the same as those used for calibration. Five coils were applied, wound at a tension of (nominally) 250 kgf to five layers. (kgf is an accepted unit in the wire rope/winch industry.) The error tolerance in the procedure was set to be within 1% of winding tension as each layer was added and it was never necessary to reiterate the solution for a layer more than once. The results are shown in Table I.

DISCUSSION

The developed form of the modelling equations for the first layer shows that the derived parameter, E'_c, is dependent upon the drum diameter. A quantity related to this also appears in the equations for the second and outer layers; here the diameter is not the drum diameter but the effective diameter upon which the current layer is being wound. For a small diameter drum, carrying large diameter rope in many layers, it might be necessary to apply the modelling equations to each layer, increasing by steps the quantity D.

Any FE analysis should aim to restrict the number of elements to a sensible optimum. The calibration procedure quoted here suggests that each rope coil can be reasonably represented by one axi-symmetric element. However the comparisons quoted in Table I show that, as the number of layers increase, there is an increasing discrepancy between the FE and the Egawa & Taneda[2] procedures. At five layers the discrepancy on nett drum load is 17% and the comparison suggests an approx 2% increase per layer once the system has settled by adding the second layer.

It must be borne in mind however that the Egawa & Taneda[2] method is not necessarily that by which other methods should be judged.

The authors have extended their shell FE procedure to a flanged drum to check whether or not the analysis is applicable to a winch barrel assembly in which the rope is confined between flanges. They discovered that their procedure absorbed considerable computing time and, for a large winch, would require more power than they had available to them. However, it was found that:

1. the principle of the procedure is applicable to the new problem.
2. there is significant variation of hoop deflection along the drum, so coil-by-coil adjustment has to be made to the temperature drop. Nevertheless convergence is still rapid; never more than two iterations were necessary.
3. some modification to the element structure is necessary to quantify, realistically, flange thrust by the confined rope. Only layers 1 and 3 (and other odd-numbered layers) impinge upon the flange, so arrangement must be made to attach the ultimate element of layers 2 and 4 (and other even-numbered layers) to the penultimate elements of layers 1, 3, etc. This ensures that, at the join, they are constrained to the same axial displacement.

CONCLUSIONS

The adoption of a single axi-symmetric element per ring needs further investigation. It may well be that the sensitivity of the solution is dependent upon the number of elements representing each rope coil. The authors used the Egawa & Taneda[2] calculation as validation for their procedure but this is, of course, not sufficient. All FE-derived designs must be checked by experimental stress analysis on the prototype and much work of this nature remains to be done.

Table I: Comparison of load measurements

Load on drum (kgf)	Load on layer 1 (kgf)	Load on layer 2 (kgf)	Load on layer 3 (kgf)	Load on layer 4 (kgf)	Load on layer 5 (kgf)	Method	Number of layers on the drum
249.4	249.4 249.4 0.0					FE method Ref 2 method % difference	1
416.7 448.6 7.1	169.9 198.6 14.4	250.8 250.8				FE method Ref 2 method % difference	2
524.8 598.5 12.3	115.4 159.7 27.7	169.2 188.7 10.4	250.2 250.2			FE method Ref 2 method % difference	3
603.6 710.8 15.1	77.4 130.6 40.6	112.2 142.9 21.5	176.0 187.3 6.1	249.7 249.7		FE method Ref 2 method % difference	4
660.2 795.7 17.0	50.2 108.9 53.9	71.3 108.6 34.3	122.2 140.3 12.9	178.8 187.4 4.6	250.5 250.5	FE method Ref 2 method % difference	5

Further development work on the operation of the procedure is also needed to make it more 'user friendly'. This, the authors feel, should be pursued in collaboration with a software house.

However, there appears to be advantage to be gained in design optimisation of heavy duty winches by adopting an FE approach along the lines of that described here. Rope load relaxation and distribution of stress throughout the barrel assembly can be effectively treated.

REFERENCES

1. N Maw & D Karbalai, 'The influence of rope lateral compliance on winch drum design', *TransIMarE*, Vol 100 (1988).
2. T Egawa & M Taneda, 'External pressure produced by multilayers of rope wound round hoisting drums', *Bulletin Japan Soc Mech Eng*, Vol 1, p 133 (1958).
3. B M Torrance, 'The design of winding drums', *The South African Mechanical Engineer*, Vol 15, p 123 (1965).

ADDENDUM

Since writing this paper the authors' attention has been drawn to recent work at NEL on wire rope modelling, the reference of which is given below.[1] Here, a structural mechanics technique based on vector analysis is used to evaluate component stresses in the rope wires under bending and torsion. This work adds to an extensive research literature on mathematical modelling techniques and contains a reference list which may be useful to the researcher rather than the designer. The authors are not however aware of any other research work which uses experimentally obtained parameters in an FE analysis.

Reference

1. S K Lee, N F Casey & T G F Gray, 'Helix geometry in wire rope', *Wire Industry*, p 461 (August 1987).

APPENDIX

Nomenclature

P	rope tensile load
d	rope diameter
δL	representative length of rope and its ring model
E_r, E_c	longitudinal Young's modulus of rope and ring respectively
E'_r, E'_c	lateral Young's modulus of rope and ring respectively
υ	Poisson's ratio (defined in derivation)
γ	rope contact angle defined in Figures
δW	rope lateral loading clarified in Figures
D	rope pitch circle diameter

The tensile load is P and loads due to rope pressure are δW at A, B, A' and B' (Figs 1 and 2). These resolve to $\delta W'$ and $\delta W''$ radially and axially.

For the ring only the radial components are used, $\delta W'$, which represent the clamping load of the next outer ring as it is cooled.

Diametral changes

For a length, δL, of rope the hoop tension is P, so the hoop stress

$$= \frac{4P}{\pi d^2}$$

and the strain

$$= \frac{\Delta\ \delta L}{\delta L}$$

$$= \frac{4P}{\pi d^2\ E_r} \tag{1}$$

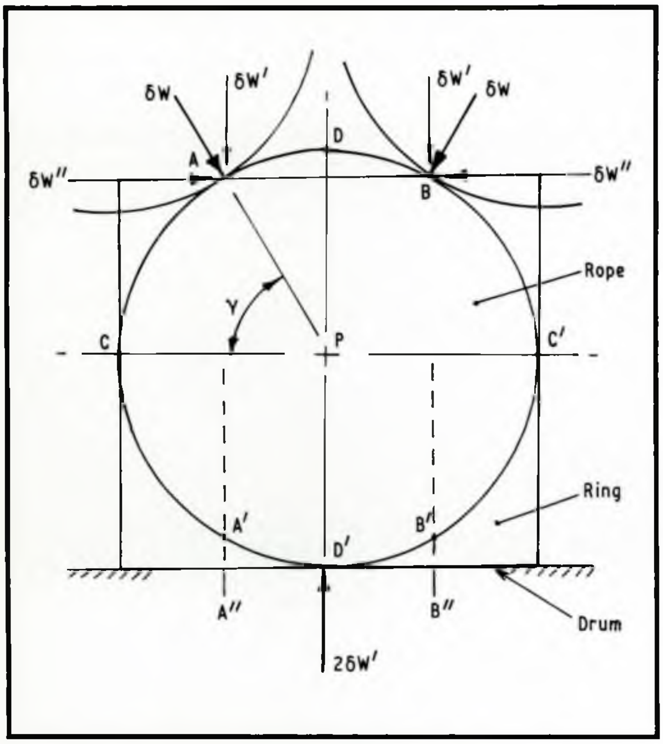

Fig 2: The rope and its ring model for the first layer

Poisson's ratio

$$\upsilon_r = \frac{\left(\dfrac{\Delta d}{d}\right)}{\left(\dfrac{\Delta \, \delta L}{\delta L}\right)}$$

therefore

$$\Delta d = \upsilon_r \left(\frac{\Delta \, \delta L}{\delta L}\right) d$$

$$= \upsilon_r \left(\frac{4P}{\pi d^2 E_r}\right) d \qquad (2)$$

The deflection in AA' (radially) due to the tension P is, by proportion, using eqn 2:

$$(\Delta \mathrm{AA}')_P = \upsilon_r \left(\frac{4P}{\pi d \, E_r}\right) \sin \gamma \qquad (3)$$

The stress over AB is assumed to be the same as that at the rope centre, ie

$$\frac{2\,\delta W'}{d \;\, \delta L}$$

So the diameter change:

$$\Delta \mathrm{DD}' = \frac{2\,\delta W'}{\delta L \;\, E'_r} \qquad (4)$$

By proportion, using eqn 4, the deflection (radially) along AA' due to 2 $\delta W''$ is:

$$(\Delta \mathrm{AA}')_{2\delta W} = \left(\frac{2\,\delta W'}{\delta L \;\, E'_r}\right) \sin \gamma \qquad (5)$$

Now, the stress on AA' due the force 2 $\delta W''$ is also assumed to be the same as that at the rope centre

$$= \frac{2\,\delta W''}{d \;\, \delta L}$$

so the diametral change

$$\Delta \mathrm{CC}' = \frac{2\,\delta W''}{\delta L \;\, E'_r} \qquad (6)$$

And, again by proportion, the deflection along AB due to 2 $\delta W''$ is:

$$(\Delta \mathrm{AB})_{2\delta W''} = \left(\frac{2\,\delta W''}{\delta L \;\, E'_r}\right) \cos\gamma \qquad (7)$$

By definition,

$$\upsilon'_r = \frac{\left(\dfrac{\Delta \mathrm{DD}'}{\mathrm{DD}'}\right)}{\left(\dfrac{\Delta \mathrm{CC}'}{\mathrm{CC}'}\right)}$$

$$= \frac{\left(\dfrac{\Delta \mathrm{AA}'}{\mathrm{AA}'}\right)}{\left(\dfrac{\Delta \mathrm{AB}}{\mathrm{AB}}\right)}$$

therefore

$$(\Delta \mathrm{AA}')_{2\delta W''} = \upsilon'_r \; \Delta \mathrm{AB} \left(\frac{\mathrm{AA}'}{\mathrm{AB}}\right)$$

which, by eqn 7,

$$= \upsilon'_r \left(\frac{2\,\delta W''}{\delta L \;\, E'_r}\right) \sin \gamma \qquad (8)$$

Thus the total deflection in AA' due to forces P, $\delta W'$ and $\delta W''$ is by eqns 3, 8 and 5:

$$(\Delta \mathrm{AA}')_{\mathrm{TOTAL}} = -\upsilon_r \left(\frac{4P}{\pi d \, E_r}\right) \sin \gamma$$

$$+ \, \upsilon'_r \left(\frac{2\,\delta W''}{\delta L \;\, E'_r}\right) \sin \gamma$$

$$- \left(\frac{2\,\delta W'}{\delta L \;\, E'_r}\right) \sin \gamma \qquad (9)$$

taking a negative sign as indicating compression. The procedure is now repeated for the ring model. For the ring, δL long, under a tensile load P, the stress

$$= \frac{P}{d^2 \sin\gamma}$$

so the strain

$$\frac{\Delta\,\delta L}{\delta L} = \frac{P}{d^2 \sin\gamma\; E_c} \qquad (10)$$

$$\upsilon_c = \frac{\left(\dfrac{\Delta d\ \sin\gamma}{d\ \sin\gamma}\right)}{\left(\dfrac{\Delta\,\delta L}{\delta L}\right)}$$

therefore

$$(\Delta d\ \sin\gamma\,)_P = \upsilon_c\, d\ \sin\gamma \left(\frac{\Delta\,\delta L}{\delta L}\right)$$

$$= \upsilon_c \left(\frac{P}{d\, E_c}\right) \qquad (11)$$

Due to the load 2 $\delta W'$, the stress on the ring

$$= \frac{2\,\delta W}{d\ \delta L}$$

so the change

$$(\Delta d\ \sin\gamma\,)_{2\delta W'} = \left(\frac{2\ \delta W'}{\delta L\ E'_c}\right) \sin\gamma \qquad (12)$$

There is no axial load on the ring so the total deflection in $d \sin\gamma$, by eqns 12 and 11, is

$$(\Delta d\ \sin\gamma\,)_{\mathrm{TOTAL}} = -\left(\frac{2\ \delta W'}{\delta L\ E'_c}\right) \sin\gamma \qquad (13)$$

$$-\upsilon_c \left(\frac{P}{d\ E_c}\right)$$

Eqns 9 and 13 are the modelling equations for radial changes in dimension. We must now derive corresponding equations for longitudinal dimension changes. Here we need the changes in δL due to P, $\delta W'$ and $\delta W'$ for the rope and, for its ring model, due to P and $\delta W'$.

Since the rope and ring are the same length we need now only equate strain in the longitudinal direction.

For the rope, due to load 2 $\delta W'$, strain

$$= \frac{\Delta \mathrm{DD'}}{\mathrm{DD'}} = \frac{2\,\delta W'}{d\ \delta L\ E'_r}$$

$$\upsilon''_r = \frac{\left(\dfrac{\Delta\,\delta L}{\delta L}\right)}{\left(\dfrac{\Delta \mathrm{DD'}}{\mathrm{DD'}}\right)}$$

therefore

$$\left(\frac{\Delta\,\delta L}{\delta L}\right)_{2\delta W} = \upsilon''_r \left(\frac{2\,\delta W'}{d\ \delta L\ E'_r}\right) \qquad (14)$$

and, due to 2 $\delta W''$ the stress (assumed constant over cross-section)

$$= \frac{2\,\delta W''}{d\ \delta L}$$

The strain in the direction of the load

$$\frac{\Delta \mathrm{CC'}}{\mathrm{CC'}} = \frac{2\,\delta W''}{d\ \delta L\ E'_r}$$

therefore

$$\left(\frac{\Delta\,\delta L}{\delta L}\right)_{2\delta W''} = \upsilon''_r \left(\frac{2\,\delta W''}{d\ \delta L\ E'_r}\right) \qquad (15)$$

And, due to P by eqn 1,

$$\left(\frac{\Delta\,\delta L}{\delta L}\right)_P = \frac{4P}{\pi d^2 E_r} \qquad (16)$$

Collecting all strains by eqns 16, 15 and 14:

$$\left(\frac{\Delta\,\delta L}{\delta L}\right)_{\mathrm{TOTAL}} = \frac{4P}{\pi d^2 E_r} + \upsilon''_r \left(\frac{2\,\delta W''}{d\ \delta L\ E'_r}\right)$$

$$+ \upsilon''_r \left(\frac{2\,\delta W'}{d\ \delta L\ E'_r}\right) \qquad (17)$$

For the cylindrical ring, stress due to 2 $\delta W'$

$$= \frac{2\,\delta W'}{d\ \delta L}$$

and strain

$$= \frac{\Delta d\ \sin\gamma}{d\ \sin\gamma}$$

$$= \frac{2\,\delta W'}{d\ \delta L\ E'_c}$$

The relevant Poisson's ratio:

$$\upsilon''_c = \frac{\left(\dfrac{\Delta\,\delta L}{\delta L}\right)}{\left(\dfrac{\Delta d\ \sin\gamma}{d\ \sin\gamma}\right)}$$

therefore

$$\left(\frac{\Delta\,\delta L}{\delta L}\right)_{2\delta W} = \upsilon''_c \left(\frac{2\ \delta W'}{d\ \ \delta L\ \ E'_c}\right) \qquad (18)$$

The strain due to P is:

$$\left(\frac{\Delta\,\delta L}{\delta L}\right)_P = \frac{P}{d^2 \sin\gamma\; E_c} \tag{19}$$

The total strain by eqns 19 and 18 is:

$$\left(\frac{\Delta\,\delta L}{\delta L}\right)_{\text{TOTAL}} = \frac{P}{d^2 \sin\gamma\; E_c} + \upsilon''_c \left(\frac{2\,\delta W'}{d\;\delta L\;E'_c}\right) \tag{20}$$

Eqns 20 and 17 are the modelling equations for longitudinal changes in dimensions.

We must now consider axial (ie parallel to winch centre line) changes in dimension for ring and rope. The procedure is much the same as that leading to eqn 13 and gives eqn (iii) where:

$$\upsilon'_c = \frac{\left(\dfrac{\Delta CC'}{CC'}\right)}{\left(\dfrac{\Delta d\;\sin\gamma}{d\;\sin\gamma}\right)}$$

For a full treatment it would now be necessary to repeat the whole procedure for modelling the first layer. Fig 2 shows the difference in configuration but the modelling procedure is similar to that described above and the detailed derivations are omitted for brevity.

Discussion

C R Chaplin (University of Reading) Dr Maw and Dr Manson (formerly Ms Karbalai) have been assiduously pursuing this topic, which is still of great practical interest, concerning the interactions between rope tensions and drum deformation in multi-layer winding. The relaxation of rope tension as both drum and rope are compressed by successive layers of rope is important not only as regards the influence on drum design but also in terms of the potentially damaging effect of the deformations on the rope itself.

After a rope has been installed on a drum, operation will never completely unwind and tension the rope so that there are always a few turns which are not tensioned during winch operation to normal working loads. These 'dead turns' also provide a modest reserve of rope should inspection and maintenance procedures require some shortening of the rope at either the front or back end. The dead turns should ideally be tensioned during installation but this is frequently neglected due to practical difficulties or ignorance of the possible consequences. The loading and winding of the tensioned rope results in the deformations which Dr Maw and Dr Manson have been concerned with. In particular the deformation of a flexible drum and the inner layers of rope in response to radial rope loads results in a relaxation of rope tension.

This effect is well known but of uncertain magnitude being a complex function of rope deformation characteristics which are hard to measure, especially since they are non-linear functions of rope loading and change with rope loading history and condition. The uncertainty of these effects is further demonstrated by the very considerable, and rather worrying, differences reported in this paper between the results obtained through finite element modelling and the Egawa & Taneda method (ref 1 below). Nor is the reader's confidence helped by the author's apparent reluctance to reveal the values assumed for various stiffnesses and so-called Poisson's ratios. (For fully anisotropic materials Poisson's ratios are defined in terms of appropriate stiffnesses, so that for example, as given by Tsai (ref 2 below),

$$\varepsilon_1 = \frac{\sigma_1}{E_1} - \frac{\nu_{12}\sigma_2}{E_2} - \frac{\nu_{13}\sigma_3}{E_3} + \frac{\nu_{16}\sigma_6}{E_6} \qquad (1)$$

(The derivation of the relevant equations in the winch drum analysis is not trivial and the use of unorthodox nomenclature and definitions makes it even harder to follow.)

As the authors point out there is a need to perform some careful experiments to check the validity of the calculation methods, and until such an exercise has been performed, any comparison of the accuracy of different calculation methods would be of 'academic' interest only. As has also been pointed out previously (ref 3 below) these calculations are totally dependent on the values assumed for the rope properties for which, as far as this investigator is aware, there still exist no valid data.

It is clear therefore that caution must continue to be exercised in drum design especially as regards the influence of drum compliance on the dead turns mentioned above. The implication of the results presented here is that, because distortion under load causes rope relaxation, it effectively reduces the drum loading. Discounting this effect results in drum over-design, while due consideration enables the design of even more flexible drums with further savings. However the consequences of this deformation on dead turns which have not been (or could not be) adequately pre-tensioned is potentially disastrous. The dead turns with zero tension will experience a similar distortion to the rest of the first layer. But instead of leading to a reduction in tension, these deformations effectively put this part of the rope into compression. This compressive deformation can accumulate in one location resulting in a phenomenon involving buckling of wires and strands which is appropriately described as a 'birdcage', although a somewhat flattened birdcage in this situation. This form of rope deformation inevitably results in a significant reduction in tensile strength which could have catastrophic consequences when at some later stage the dead turns become live.

A method which accurately predicts the rope and drum deformations involved in multi-layer winding would help to quantify the pre-tension of dead turns necessary to avoid this form of rope damage. The immediate needs therefore are for careful measurements not only of the complete rope and drum system, but also of the full three-dimensional load displacement characteristics of an appropriate range of rope constructions.

References

1. T Egawa & M Taneda, 'External pressure produced by multi-layers of rope wound round hoisting drums', *Bulletin Japan Soc Mech Eng*, Vol 1, p 133 (1958).
2. S W Tsai, 'Composites design 1986', Think Composites, Dayto, Ohio (1986).
3. C R Chaplin & A E Potts, written discussion of 'The influence of rope lateral compliance on winch drum design', by N Maw & D Karbalai, *Trans IMarE*, Vol 100, pp 9–10 (1988).

N Maw (Sunderland Polytechnic) Dr Chaplin's point about 'dead turns' is, of course, familiar to us. In fact dead turns can be accommodated in the Egawa/Taneda procedure which, under their circumstances, predicts a final compression load in first turns. We think this is unlikely and, when we use the Egawa/Taneda method, we disregard the negative sign and use it only as an indicator that the load in the bottom layers remains at zero. This point is returned to later.

Our finite element procedure will also allow for dead turns.

We are not particularly worried by discrepancies between our results using finite elements and the Egawa/Taneda calculation. In the first place the discrepancies are not as large as Dr Chaplin suggests – they are well within normally acceptable tolerances for design. Also there are insufficient experimental results to favour one method over the other. It seems likely to us that both lead to over-design of the barrel so the product is still on the safe side.

We take exception to Dr Chaplin's remark that we are reluctant to reveal results. There is no intention, actual or implied, that our work is unavailable for inspection. We have invited Dr Chaplin (and Mr Potts) to talk to us. We have discussed our work with eg British Coal, British Ropes Ltd, NEL, universities and polytechnics and, if anyone wishes to know what values we used, they are contained in Dr Manson's (nee Karbalai) thesis (ref 1 below). We have not quoted them here simply because they are too specific to our own rig. In any case we have clearly defined our parameters and we are sure that Dr Chaplin is capable of performing the necessary experiments to yield the required values.

The relevant equations for modified 'elastic' properties are tedious but not difficult to solve and the nomenclature is not unorthodox – where standard parameters are used we have used standard symbols. If Dr Chaplin perseveres he will find that the derivations contained in the Appendix of the paper are easy enough followed.

Dr Chaplin again raises the point of 'dead turns' and talks about compressive loads and the 'bird cage' effect. We feel that this is all too speculative and unsupported experimentally to discuss with any confidence. Much more work is needed on the material properties, taken as a structure, of wire rope *in situ*. Dead turns, type of rope bed, flanges, lubrication, are all matters which should be addressed but only after careful basic experiments on loading of long winch barrels with rope under constant tension, much as we did, but on a larger scale.

Reference

1. D Karbalai, 'The design of winch barrels with particular reference to multi-layering', PhD thesis, Sunderland Polytechnic (1988).

A E Potts (Global Maritime) The authors are to be congratulated on an interesting paper which presents an innovative procedure for analysing multi-layered winch drums using FE techniques. It is a worthwhile extension of the previously reported mathematical modelling techniques which deal with the problem of the influence of rope-load relaxation on winch drum and flange stresses.

Although the stated purpose of the paper was to outline the FE modelling technique for optimising the design of winch barrels, with regard to cost and weight, the critical factor in any such analysis, as recognised in this and the authors' previous paper (ref 1 of the paper), is the accurate modelling of the rope. The modelling of each coil of rope as a ring with orthotropic properties (and the use of equations taking account of radial and longitudinal changes in the rope), appear to deal quite adequately with the likely loading–response conditions within the multi-layered, wound rope-winch drum system.

The major issue of concern for the direct application of this procedure, and for previously reported analytical methods, is the actual rope properties to be used, how they were measured, and whether they describe the response properties embodied in the rope property parameters used in the model equations. This is not introduced to detract in any way from the analytical procedure presented, but more as a cautionary note, that before winch manufacturers rush to optimise their designs, a detailed investigation of rope properties under winch conditions is required, and the model checked and calibrated from experimental stress analysis of actual winch drums. The authors rightly acknowledge the latter point. However it is considered that the latter cannot be achieved without the information provided by the former.

This point was also a 'hot topic' of discussion with the authors' previous paper (ref 1 of the paper) and is considered central to the problem, and not a 'side issue' on this subject. Unfortunately the author's expectation raised in the reply to discussion of their previous paper (ref 1 of the paper), that the large amount of rope research to date should have produced the 'useful applicable data in easily assimilated form' that winch designers need, is not the case and should be addressed in the near future. Such information can only enhance the validity of the analytical approach proposed by the authors.

The authors' addendum draws attention to recent work at NEL on theoretical stress analysis techniques for wire ropes. An unfortunate point about all such techniques reported to date for analysing wire ropes, is that they are unable to adequately model with any accuracy friction within the rope construction, which in turn governs rope response and therefore wire stresses. Consequently it is considered that such procedures would be unsuitable for determining rope properties for use in winch analysis and that a careful programme of tests is required.

Further developments of the reported procedure will be followed with keen interest, in particular results from checking of the procedure by experimental stress analysis. In their previous paper (ref 1 of the paper) the authors mentioned that, at present, most winch designers use the conservative Torrance approach, which does not take full account of all the rope-load relaxation mechanisms involved with multi-layering. Accordingly, winch drums designed using the Egawa & Taneda method (ref 2 of the paper) could be made lighter and cheaper. If, as the results in Table I of the paper suggest, the Egawa & Taneda procedure actually significantly over-estimates drum loadings with multi-layered winding, and, the FE model proves to be a more accurate method for determining design stresses, then winch drum design could be radically improved in the future using the FE procedure proposed by the authors in this paper.

N Maw (Sunderland Polytechnic) Some of the points raised by Mr Potts are dealt with in reply to Dr Chaplin. We can only agree with him however that much careful work still needs to be done on rope properties. We have been advocating this for years and we are pleased to see that, at last, someone is supporting us in our request for useful tabulated data for the winch designer.

Mr Potts emphasises the immense complexity of modelling wire rope from first principles using as his sample the effects of friction. But there are other equally acute difficulties, for instance the contact phenomena between individual wires. Some work is being done on this at Imperial College but, it seems it us, until this is completed all other effects are secondary. Our paper has, to some extent, dodged these difficulties by combining simple experiment with FE analysis.