Rolling of materials with a high yield strength necessitates the use of a winder (coiler) which is suitable for this service. This paper reviews the development of high tension coilers and the salient features of a controlled collapse winder.

Sendzimir controlled collapse winder

John W. Turley,* T. Sendzimir, Inc., Waterbury, Conn.

THE winding (or coiling) of metal strip under very high tensions has always been a problem to the designers and operators of coilers. The classical solution to this problem was to wind the strip onto a solid cylindrical block (Fig. 1), and the coil could only be removed from the block by unwinding. Thus early sendzimir mill lines for rolling of stainless steel under high tension consisted of a coiling line to wind the strip from a collapsible pay-off reel onto a solid block, a mill with two solid block winders and a rewinding line to rewind the strip onto a collapsible mandrel at a comparatively low tension. This arrangement is still favored by some mill users, but most of the advanced mills, operating with tensions as high as 120,000 lb (55 tonnes) utilize the controlled collapse winders, giving the operators the benefit of lower capital costs, lower running costs and increased production.

The fundamental problem is that winding strip onto a drum produces very high compressive radial and hoop stresses in the drum. The process is analogous to the wire winding of gun barrels in which compressive prestresses are deliberately introduced to reduce the peak tensile stress which occurs during firing. Sims and Place[1] calculated the relations between radial and hoop stresses and coil build-up treating the mandrel and coil as a homogeneous tube (Fig. 2). It can be seen that a radial compressive stress at least equal in magnitude to the tensile stress in the strip can be developed at the surface of a solid block at a typical build-up ratio of 3. The average strip tensile stress when rolling stainless steel at light gages can be as high as 50,000 psi (35.5 kg/mm^2) and if the strip flatness is not perfect the peak tensile stress can easily be twice as high. Thus radial compressive stresses of the order of 50,000 to 100,000 psi will commonly be experienced on solid block mandrels winding stainless steel strip and hoop stresses will be even higher. It is, therefore, most important that the solid blocks should be made from sound homogeneous heat-treated forgings of alloy steel. The use of lower quality steel almost inevitably leads to "hour glassing" (plastic compressive deformation resulting in a reduced block dia underneath the coil and usually accompanied by an increase in axial length of the block).

To overcome the disadvantages of the solid block design the collapsible block (Fig. 3) was introduced in 1955. With very careful maintenance, this design performed well for winding tensions up to about 50,000 lb. However, its life was limited and there were operational problems. The pyramid angle of about 6° was less than the friction angle so normally no relative axial motion between segments and pyramid shaft took place during winding and unwinding. This meant that the radial pressure on the segment was almost as high as would be developed on the surface of a solid block. If the mandrel had not been lubricated for a few days it was often found impossible to collapse the mandrel to remove the coil as the segments and pyramid shaft would seize. The coil could then only be removed by rewinding at light tension onto the other reel. In an effort to alleviate this problem many users would burnish the mating faces of segments and pyramid shaft with molybdenum disulfide compounds. This frequently reduced the friction coefficient so that the friction angle dropped below the pyramid angle. The result was that the mandrel would collapse during winding of a coil with disastrous results since collapsing caused an axial displacement of the segments which led to a telescoped coil. (In Fig. 3 the collapse of the mandrel is accompanied by axial movement of segments 6, piston rod 28 and piston 8.) At best, with normal operation of this mandrel, the wear rate on the mating surfaces of pyramid shaft and segments was very high due to the relative motion between them at very high pressures as the mandrel was collapsed to remove the coil on completion of rolling. As tensions and coil sizes increased, it became apparent that this design would have to be improved and the result of the ensuing development was the controlled collapse mandrel (Fig. 4) which was introduced in 1960.

Fig. 1 — Solid block mandrel.

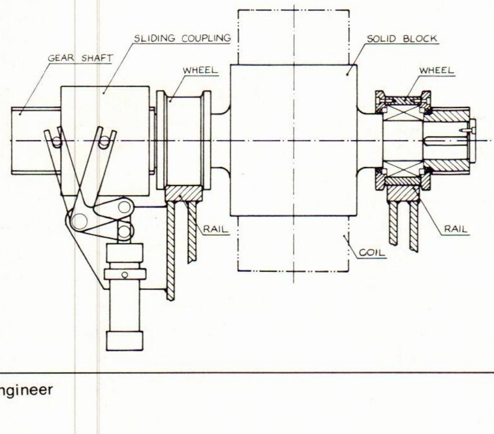
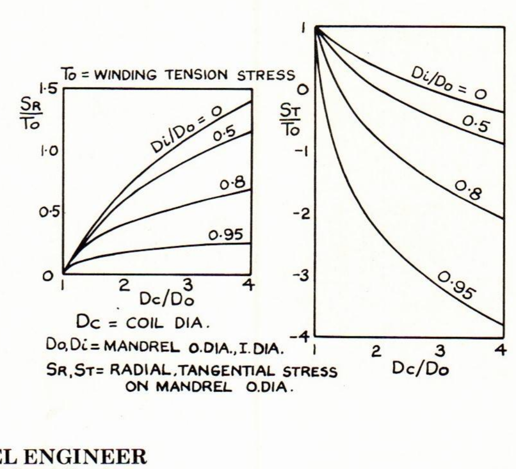


Fig. 2 — Sims and Place's results.



* Chief Engineer

Reprinted from **IRON AND STEEL ENGINEER**

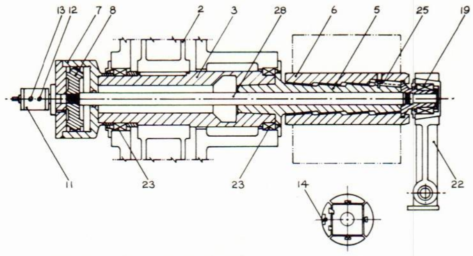
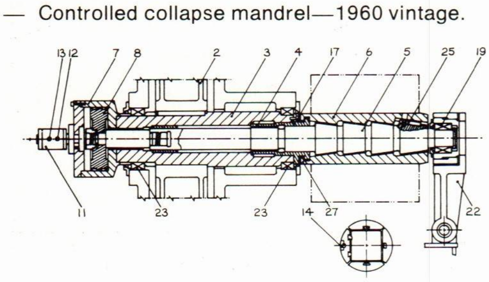


Fig. 3 — Early collapse block design—1955 vintage.

The most important fundamental design feature of the new mandrel was the controlled collapse property. The pyramid angle was increased to 7 to 8° to give an angle greater than the friction angle thus insuring that a well lubricated mandrel would collapse during the build-up of a coil. The rotating cylinder was sized so that the pressure in the cylinder during winding would provide resistance to the collapse of the mandrel. The amount of collapse during the winding of a coil could be controlled by the setting of the pressure regulator supplying oil to the rotating cylinder. The surfaces of the pyramids on the pyramid shaft were faced with a layer of aluminum bronze deposited by welding and finish machined to give a nongalling bearing surface. The segments were provided with 27 hooks at the inboard end which engaged with the gear shaft so the segments were prevented from moving axially. Expansion and collapse of the new mandrel was achieved by axial movement of the pyramid shaft, which was provided with a long extension which was mounted inside the gear shaft and extended back to the rotating cylinder at the back of the gear shaft. The hydraulic piston was mounted directly on to the pyramid shaft extension. In this way the mandrel could collapse during winding without the danger of the coil telescoping. For the wider mandrels an outboard support bearing was provided. This bearing was mounted on the end of the pyramid shaft and was supported by a gate which could swing open to allow coil removal. The gate was adjustable in vertical and horizontal planes to insure perfect alignment with the mandrel. The bearing was designed to slide axially relative to the gate during the axial movement of the pyramid shaft which took place as the mandrel collapsed during winding. This controlled collapse mandrel was very successful and many mandrels of this design are operating well on mills with tensions up to 100,000 lb (45 tonnes) and coil sizes up to 80 in. (2 metres). However, in certain instances problems arose due to failure of the hooks on the inboard end of the segments. This problem led to the development of the modern reverse pyramid design of controlled collapse winder (Fig. 5) which was first introduced in 1968, and which is now being installed on a number of mills in the U. S.

The reverse pyramid winder operates on exactly the same principles as the earlier controlled collapse design but the slope of the pyramids is reversed so that in operation the pyramid shaft is in tension instead of compression, and the segments are in axial compression instead of

Fig. 4 — Controlled collapse mandrel—1960 vintage.



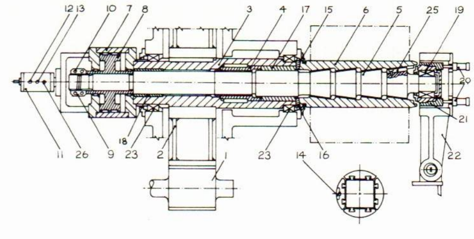


Fig. 5 — Reverse pyramid controlled collapse mandrel—1968 design.

tension. By this means the troublesome hooks on the inboard ends of the segments are eliminated. The design is much more robust than the earlier design, and is suitable for the highest tensions and coil sizes that are likely to be used in the foreseeable future. The reverse pyramid design has two minor drawbacks when compared with the earlier design. First, because the rod end of the rotating cylinder is pressurized during operation, the cylinder dia has to be increased. Second, a separate retractable sliding sleeve (with hydraulic cylinder actuation) has to be mounted in the outboard bearing gate. This is because the pyramid shaft is in its most forward position when the mandrel is collapsed, and without the sliding sleeve there would be interference between outboard bearing and gate during opening and closing of the latter. However, the sliding sleeve gives the advantage that the gate can be opened and closed with mandrel expanded or collapsed.

Apart from the earlier mentioned work by Sims and Place, the most important published work is that of Wilkening et al.[2] They extend the homogeneous tube theory presented by Sims and Place to cover the case of cylindrical mandrel and strip with different elastic moduli. The mandrel and coil are treated as a composite cylinder with successive rings of strip shrunk on. The theory is developed to include the effects of allowing the mandrel to deform elastically sufficiently to allow relative motion between wraps. Under these conditions it is possible to generate higher radial pressure on the mandrel than the shrink-ring theory would indicate. The actual pressure developed depends on the friction coefficient between wraps. As far as the mandrel designer is concerned, this condition is hypothetical as it is necessary to design the mandrel to avoid interwrap slippage occurring since this would cause scratch marks on the strip.

Reverse pyramid collapsible winder

A modern design of winder is illustrated in Fig. 5. The drive is transmitted from pinion 1 through gear 2, gear shaft 3, spline bushing 4, and pyramid shaft 5 to segments 6 and rotating cylinder 7. The mandrel is collapsed and expanded by means of pressurized hydraulic oil acting on piston 8 which is retained on the pyramid shaft by means of split nut 9, clamped on the pyramid shaft by means of safety bolts 10. Hydraulic oil is supplied to the rotating cylinder and strip gripper 14 by means of rotary coupling 11 mounted on the rear of the rotating cylinder. The segments are retained on the pyramid shaft by keys 25 which allow relative axial movement between segments and shaft. When the pyramid shaft is moved forward to collapse the mandrel, forward motion of the segments 6 is prevented by means of the split retainer ring 15, so the segments are drawn radially inward by means of keys 25. Expansion is achieved by backward motion of the pyramid shaft. Thrust pads 16 are used during winding to transmit thrust from the segments to the gear shaft. The pyramid shaft is mounted within the gear shaft on bushings 17 and 18. The complete mandrel, gear and rotating cylinder assembly is mounted in the gear reducer on two bearings 23. For wide strip a third bearing 19 is provided outboard of the mandrel, and this bearing is mounted on a pivoted gate 22, and is supported within the gate by sleeve 21 which can be retracted by hydraulic cylinders 20. To enable coils to be loaded and removed from the mandrel sleeve 21 is retracted and the gate is then swung open by means of a hydraulic cylinder. The strip end is gripped by hydraulic gripper 14 and this enables full tension to be applied to the winder after less than one lap has been wound.

Lubrication is extremely important to the proper operation of the mandrel and separate grease points are provided for each individual sliding surface, including all segment surfaces, bushings, splines, thrust faces and outboard bearing. Reducer bearing lubrication is by force feed supply of oil. It is recommended that greasing is performed at least every three shifts. The winder mandrel has been designed for ease of maintenance in every respect. Removal of retainer ring 15 and split nut 9 enables the four segments and the pyramid shaft to be entirely removed and replaced without disturbing any other piece of equipment. The use of jigs during manufacture insures that all parts are fully interchangeable, and segments and grippers are designed to be suitable for installation on either left or right-hand winders.

The pyramid angle of 7½ or 7¾° insures that the pyramids are not self locking and that the mandrel can collapse during winding, thus preventing excessive radial pressure on the mandrel, provided that the pyramids are well greased. The same angle is normally provided on the back faces of the segments and the mating face on thrust pads 16. This is to provide a radially inward force on the back face of each segment. This overcomes the friction force at the interface which would otherwise tend to prevent the back end of the segment from moving radially inward during winding possibly damaging the segment and retaining keys.

Theory of operation

The theory is presented in full in Appendix 1 for the case of winders of the reverse pyramid type (Fig. 5). The theory is also applicable to the 1960 design of mandrel (Fig. 4).

Winding — Based on the successive shrink ring theory of Wilkening, the mandrel and coil are treated initially as a composite cylinder. The internal dia of the equivalent tubular mandrel having the same radial compliance as the actual mandrel is covered in the appendix. Also the change in radial and tangential stresses at the coil bore due to winding one lap are derived. Successive laps are wound using the theoretical model until the condition for collapse of the mandrel is reached, i.e., when the radial pressure on the mandrel (equals radial stress at coil bore) reaches the limiting value, equation 15.

Collapse during winding — The mandrel collapses when the radial pressure on the mandrel overcomes the combined effects of the resistance of the oil pressure in the rotating cylinder, and the static friction at the interfaces between segments and pyramid shaft. The mandrel continues to collapse and the pyramid shaft moves forward until a new condition of equilibrium is reached when the radial pressure has dropped until it is insufficient to overcome the combined effects of oil pressure and dynamic friction on the pyramids. In fact it is because the dynamic friction coefficient is lower than the static friction coefficient that collapsing occurs in a series of stick-slip steps. During each collapse step, which occurs very rapidly, the oil pressure in the rotating cylinder rises until the pressure relief valve has had time to respond. The safe assumption is made in the calculation that the collapse step is complete before the relief valve has responded, and the

---

Nomenclature

Parameter		Units Metric	English
E_1	Elastic modulus of mandrel	kg/mm²	psi
E_2	Elastic modulus of strip	kg/mm²	psi
E_3	3.141593 (π)	—	—
W	Strip width	mm	in.
S	Strip thickness	mm	in.
T	Pyramid angle	degrees	degrees
T_1	Tangent of T	—	—
M_5	Poisson's ratio for mandrel	—	—
M_6	Poisson's ratio for strip	—	—
R_o	Initial radius of mandrel = coil internal radius	mm	in.
A_1	Coil external radius	mm	in.
A	Average ½ side of pyramid	mm	in.
A_o	Internal radius of equivalent tubular mandrel	mm	in.
S_1	Winding tension in strip	kg/mm²	psi
M_1	Static friction coefficient of mandrel	—	—
M_2	Dynamic friction coefficient of mandrel	—	—
M_3	Dynamic friction coefficient of strip	—	—
Q_5	Pressure in rotating cylinder at start of winding	kg/cm²	psi
Q_6	Pressure in rotating cylinder when unwinding	kg/cm²	psi
K	Bulk modulus of oil in rotating cylinder	kg/mm²	psi
$L_1, L_2, D_1, D_2, D_3, D_4$	mandrel dimensional constants	mm	in.
K_6	Axial stiffness of mandrel	kg/mm	lb/in
I	Number of laps on coil at any instant	—	—
R_1	Coil external radius after (I-1) laps have been wound	mm	in.
P_o	Radial stress at coil bore	kg/mm²	psi
Q_o	Hoop stress at coil bore	kg/mm²	psi
P_1	Change in P_o due to winding or unwinding of one lap on outside of coil	—	—
Q_1	Change in Q_o due to winding or unwinding of one lap on outside of coil	kg/mm²	psi
P	Change in P_o during collapse or expand movement	kg/mm²	psi
Q	Change in Q_o during collapse or expand movement	kg/mm²	psi
P_2	Radial stress at outer surface of (I-1)th lap due to addition of Ith lap	kg/mm²	psi
H_1	Axial displacement of rotating cylinder piston relative to end of stroke (mandrel expanded)	mm	in.
H_2	Change in H_1 during collapse or expand cycle	mm	in.
U_1	Elastic inward displacement of mandrel external surface	mm	in.
U_2	Elastic inward displacement of coil bore	mm	in.
S_R	Radial stress on mandrel OD	kg/mm²	psi
S_T	Tangential stress at mandrel OD	kg/mm²	psi

collapse movement is absorbed by elastic axial deformation of mandrel components.

Out-of-roundness errors — Because the segments are of fixed geometry, the outside of the mandrel is a true circle only at one axial position of the pyramid shaft relative to the segments, although, the deviation from the true circle is very small at all times during operation. The relationship between mandrel dia and axial position of pyramid shaft (equal to rotating cylinder piston displacement) is affected by this deviation. During the collapse movement, changes in the oil pressure, (hence cylinder force) radial stress at coil bore and displacements, and the change in hoop stress at coil bore are calculated, the coil and the mandrel being treated as separate cylinders.

After the first collapse movement is completed, further laps are added, the mandrel and coil being treated again as a composite cylinder until the radial pressure on the mandrel reaches the limiting value, (equation 15) when the second collapse occurs. The whole process is continued, collapsing occurring from time to time until the coil is completely wound.

Unwinding — The process of unwinding the coil is treated theoretically in a similar manner to the winding process. During unwinding, however, the friction at the pyramids acts in a direction to resist the cylinder force whereas during winding it acts in a direction to aid the cylinder force in resisting the collapse of the coil. The equations which establish the conditions under which the mandrel expands are given in the appendix along with those which give the changes in forces, stresses and displacement.

Drive spline friction — There are a number of second order effects which modify the behavior of the mandrel during collapse and expansion when operating and the most important of these is the effect of frictional resistance to axial motion of the pyramid shaft at the drive splines and pyramids.

Slippage between coil wraps — It is clear that the winding operation can only be treated with the coil and mandrel as a composition cylinder with successive rings shrunk on if there is no slippage occurring between laps of the coil. Section 9 of Appendix 1 derives the condition which must be satisfied if this slippage is not to take place. Basically, slippage tends to occur because the coil is transmitting torque between mandrel and strip. The term, slip ratio, is defined as the ratio of maximum torque transmissible without slip to actual torque. A slip ratio of less than unity indicates not only that the theory breaks down, but also means that strip marking is likely to occur because of the slip. No attempt is made here to establish the theoretical behavior of a coil whose laps are slipping. Rather it has been the intention to use the concept of slip ratio to help in designing and operating mandrels so that slip does not occur.

Values of friction coefficient — Friction coefficient values for the mandrel were based largely on experience. It is well known the dynamic friction coefficients in the range 0.08 to 0.1 apply for grease lubricated steel/bronze bearings at low rubbing speeds after running in surfaces initially machined to a fine surface finish. Such values have been obtained by power measurements on 4-h mill screwdowns for example. Considering the care taken in the manufacture of the mandrels, a value of 0.08 was considered to be realistic. Static friction of course varies considerably with ambient conditions, but based on the knowledge that a 6° pyramid angle (tangent = 0.105) would give in general a noncollapsing mandrel, and a 7.5° angle (tangent = 0.132) would generally give a collapsing mandrel, a static friction coefficient of 0.12 was taken for the mandrel for the purposes of the calculations.

Friction coefficient values for the interwrap conditions within the coil were taken from Wilkening et al., who reported values of friction coefficient of 0.15 to 0.18 (static) and 0.10 to 0.13 (dynamic). For the purposes of the calculations a conservative value of 0.10 was used.

Scratch marks — Scratch marks are generally produced when slippage between wraps takes place. The marks are probably caused by small airborne dust particles or metal fines which cut the strip surface during slippage. Some materials are more sensitive than others—stainless steels being generally the most critical due to the high surface quality which is normally demanded. The most common cause of scratch marks is unwinding a coil at a different tension from the tension at which it was wound. This results in relative motion between the outermost wraps of a coil due to change in elastic strain of the strip as it is removed. The most common cure for scratch marks is the use of paper interleaving. This is used almost invariably for the final pass when rolling stainless steel because the coil is normally wound at high tension, then subsequently it is unwound at low tension on a processing line.

Unless paper interleaving is used, some scratching can almost always be expected on the innermost and outermost few wraps of any coil due to the loss of tension which takes place at the coil ends. This can be minimized by the use of leader strips. On the other hand the main body of a properly formed coil should be entirely free of such defects.

Slippage and scratch marks can also be caused by poorly designed mandrels which depart considerably from the desirable truly circular form. They can also be produced on controlled collapse winders if incorrect pressure settings are used in the expand/collapse cylinder.

Operating Analyses

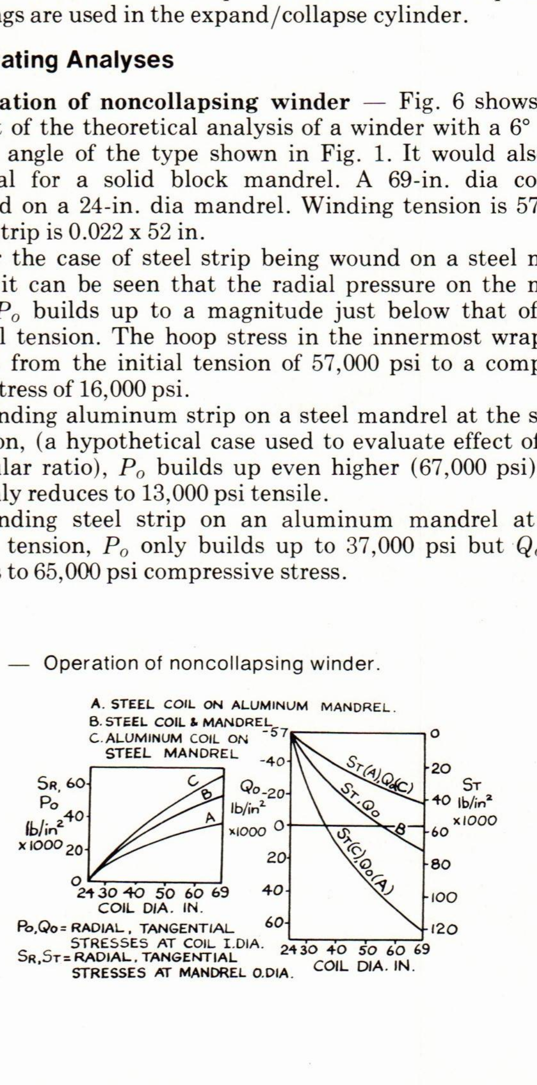
Operation of noncollapsing winder — Fig. 6 shows the result of the theoretical analysis of a winder with a 6° pyramid angle of the type shown in Fig. 1. It would also be typical for a solid block mandrel. A 69-in. dia coil is wound on a 24-in. dia mandrel. Winding tension is 57,000 psi. Strip is 0.022 x 52 in.

For the case of steel strip being wound on a steel mandrel, it can be seen that the radial pressure on the mandrel P_o builds up to a magnitude just below that of the initial tension. The hoop stress in the innermost wrap Q_o drops from the initial tension of 57,000 psi to a compressive stress of 16,000 psi.

Winding aluminum strip on a steel mandrel at the same tension, (a hypothetical case used to evaluate effect of the modular ratio), P_o builds up even higher (67,000 psi) but Q_o only reduces to 13,000 psi tensile.

Winding steel strip on an aluminum mandrel at the same tension, P_o only builds up to 37,000 psi but Q_o reduces to 65,000 psi compressive stress.

Fig. 6 — Operation of noncollapsing winder.



These results indicate that a mandrel which yields under the squeezing force of the coil gives reduced radial pressure on the mandrel surface, but increased compressive hoop stresses in the inner wraps of the coil.

Operation of controlled collapse winder with equal wind and unwind pressure — Fig. 7 shows the results of the theoretical analysis of a winder of the type shown in Fig. 5 under similar conditions to those in Fig. 6. The pyramid angle is increased to 7¾° and a 34-in. dia cylinder is pressurized to 1200 psi to provide the controlled resistance to collapse of the coil during winding and unwinding.

As winding proceeds the radial pressure P_o increases to about 15,000 psi and at this point collapse occurs causing P_o to drop to about 10,000 psi, Q_o to become more compressive and also causing a sharp peak of about 2700 psi (or just under 2½ times the pressure setting) in the rotating cylinder. Winding continues with P_o increasing to about 15,000 psi when collapse occurs again—the process continuing until winding is complete, seven collapse steps having occurred. The slip ratio is high throughout winding.

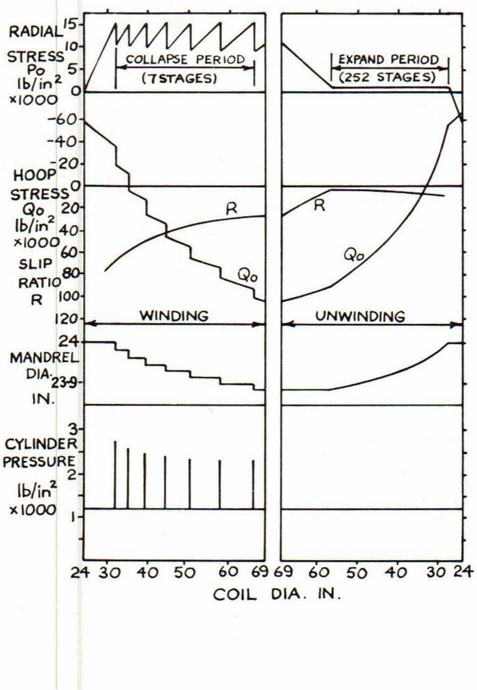
When the coil is completely wound, the hoop stress Q_o is about 105,000 psi compressive, the mandrel having collapsed by 0.12 in. in dia to 23.88 in.

As unwinding proceeds, P_o gradually reduces until it reaches about 700 psi. An expand step then occurs which increases P_o to around 1000 psi. This increase is so small that after two or three more wraps are unwound the pressure P_o drops to 700 psi again and another expand step occurs. In fact during unwinding so many small expand steps (252) occur that on the scale of the graph the steps are not visible and the process appears to be continuous. As unwinding proceeds the hoop stress Q_o becomes more tensile. No sharp peaks in rotating cylinder pressure occur as the pressure tends to drop during the expand steps. The slip ratio during unwinding drops to a minimum of about 2 as the expand cycles start.

The most interesting feature of Fig. 6 is that during unwinding the mandrel expands to its full dia before unwinding is complete, i.e., when the coil dia is about 28 in. The coil bore hoop stress Q_o reaches its original value of 57,000 psi tensile (i.e., the winding tension) at this time. As unwinding proceeds beyond this point the radial pressure P_o and the slip ratio drop to zero, and the rest of the coil could, therefore, be expected to suffer from scratch marks unless paper interleaving is used.

Most sendzimir controlled collapse winders still operate with equal cylinder pressures during winding and unwinding, and paper interleaving is sometimes used for the inner part of the coil at the lighter gages in order to eliminate the scratch marks.

Fig. 7 — Operation with same cylinder pressure during winding and unwinding.



Thermal effects — Strip produced by cold rolling is generally hot when it is wound (100 to 300 F) and this heat is transmitted by conduction to the mandrel components, and by convection and radiation to the surrounding atmosphere. When a coil takes a long time to wind, thermal contraction of the coil (due to loss of heat) and expansion of the mandrel (due to gain of heat) lead to even higher radial pressures for noncollapsing winders than Fig. 6 indicates. On the other hand the controlled collapse winder inherently prevents the build-up of excessive radial pressure whether caused by mechanical or thermal effects.

Operation with reduced pressure during winding — Fig. 8 shows the result of reducing cylinder pressure to about 150 psi during winding, and increasing to 1500 psi during unwinding, in an attempt to obtain substantially equal radial pressures during winding and unwinding.

In this case P_o values of 1500 to 2000 psi prevail throughout winding and unwinding, all collapse and expand steps are small and occur frequently, and the hoop stress Q_o reaches a maximum (compressive) of about 210,000 psi.

The objective of preventing the loss of radial pressure below 28 in. dia when unwinding has been achieved. Radial pressure now drops to zero, when the mandrel has expanded fully and when the coil dia reduces to about 24½ in.

There are two disadvantages to this mode of operation. The value of Q_o of 210,000 psi in compression is higher than one would like, and the slip ratio R drops to below unity just before completion of winding of the full coil.

In an initial attempt to correct this situation the cylinder pressure during winding was increased to 300 psi giving a radial pressure of 3000 to 4000 psi during winding, but in this case the radial pressure P_o dropped to zero during unwinding when the coil dia was 25.4 in. so about 30 laps of coil were subject to slippage. Clearly the increased radial pressure during winding was responsible for this effect.

Operation with two pressure stages during winding — Fig. 9 shows the result of increasing the cylinder pressure to about 1100 psi about half way through the winding of the coil. Unwinding is carried out at a cylinder pressure of 1500 psi as before.

It can be seen that the slip ratio is prevented from dropping below about 2 by the increase in cylinder pressure, and the peak compressive value of Q_o is reduced to 200,000 psi, still fairly high. There is still no tendency for loss of radial pressure toward the end of unwinding, so this method appears to be a useful way of satisfying the theoretical requirements for winding a good coil.

Other operating conditions — The calculations have all been made for the worst condition—maximum tension force and maximum tension stress. At lighter gages the stress will be the same but tension force will be lower, so the slip ratio will generally be higher giving better conditions. At heavier gages the tension force will be the same but the stress will be lower. The radial pressure on the mandrel will remain substantially unchanged as this is determined by the pressure in the rotating cylinder. However, the compressive hoop stress will not become so high as less collapsing is required to relieve the lower initial stress. The slip ratio will also be higher.

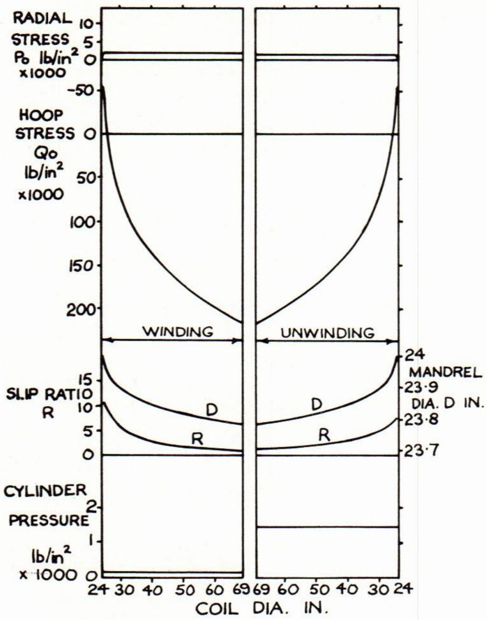


Fig. 8 — Operation with reduced cylinder pressure during winding.

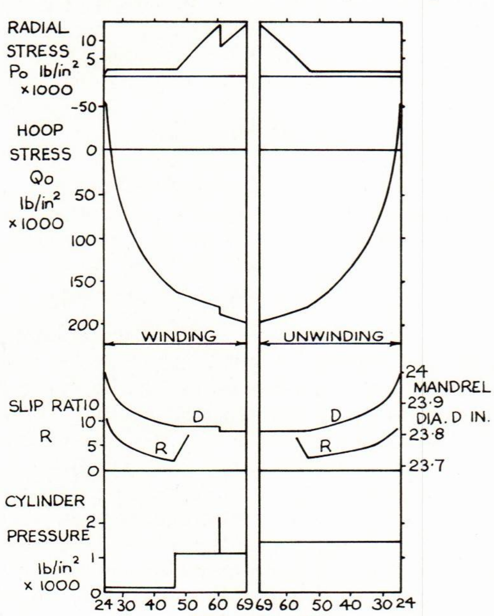
In summary all other operating conditions are better than the one for which the calculations have been made.

Practical operation — From the results shown in Figs. 7, 8 and 9 it is clear that the loss of radial pressure which occurs toward the end of unwinding can be prevented by using a lower cylinder pressure during winding, which also tends to keep the radial pressures during winding and unwinding more nearly equal.

However, if the cylinder pressure is too low during winding, the compressive hoop stress in the inner wraps of the coil becomes too high, thus increasing the danger of buckling if the coil is stripped from the mandrel. [If the coil is not stripped, but is rewound, this is unimportant.] There is also some danger of slippage of the inner wraps of the coil in this case.

The best operating pressure during winding should be established by practice, and practice will also determine whether the two pressure stages during winding will result in any benefit.

Fig. 9 — Operation with two pressure stages during winding.



Future development

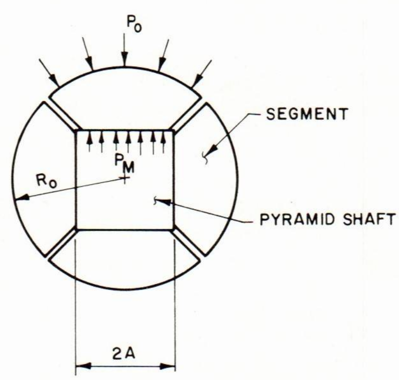
One limitation of the performance of the mandrel is that, because of the relatively high friction between segments and pyramid shaft, the radial pressure during unwinding is limited to about 1500 psi, with reasonable size of rotating cylinder.

If the friction coefficient could be reduced by a factor of 10 or more, a smaller rotating cylinder could be used, and the radial pressure during unwinding and winding could be made substantially equal and sufficiently high to give reduced compressive hoop stresses with higher slip ratios.

However, the only technique which could be used would be the use of rolling contact elements between segments and pyramid shafts. This approach is under investigation at the present time and if this method is economically attractive, a number of mandrels of this type will be going into operation over the next few years.

Appendix 1
Theory of Controlled Collapse Winders

Equivalent tubular mandrel



Force balance on unit width of one segment:

$$\int_{-\frac{\pi}{4}}^{\frac{\pi}{4}} P_o \cos\theta \, R_o d\theta = P_M 2A$$

$$P_o R_o [\sin\theta]_{-\frac{\pi}{4}}^{\frac{\pi}{4}} = P_M 2A$$

$$P_M = \frac{P_o R_o}{2A}\left[\frac{1}{\sqrt{2}} + \frac{1}{\sqrt{2}}\right] = \frac{P_o R_o}{\sqrt{2}A} \qquad (1)$$

This is the same pressure which would arise if the pyramid shaft were circular in section (radius $\sqrt{2}\,A$) therefore, assume deflection is similar. For the solid central section radial stress = tangential stress = P_M throughout (assuming no axial constraint). Radial shrinkage of central section

$$U_c = \frac{\sqrt{2}A}{E_1}(P_M - M_5 P_M) = \frac{P_o R_o}{E_1}(1 - M_5) \qquad (2)$$

For the segment of internal radius $\sqrt{2}\,A$ and outside radius R_o at some intermediate radius r

$$\text{Stress} = \frac{P_o R_o}{r};\ \text{Strain} = \frac{P_o}{E_1}\frac{R_o}{r};$$

$$\text{Compression} = \frac{P_o}{E_1}\frac{R_o}{r}dr$$

Radial compression of segment

$$U_s = \int_{\sqrt{2A}}^{R_o} \frac{P_o}{E_1}\frac{R_o}{r}\,dr = \frac{P_oR_o}{E_1}[\ln r]_{\sqrt{2A}}^{R_o} = \frac{P_oR_o}{E_1}\ln\frac{R_o}{\sqrt{2A}} \tag{3}$$

From equations 2 and 3 radial displacement of mandrel outside surface

$$U_1 = U_c + U_s = \frac{P_oR_o}{E_1}\left[\ln\frac{R_o}{\sqrt{2A}} + (1 - M_5)\right] \tag{4}$$

For a tubular mandrel of outside radius R_o and inside radius A_o under an external radial pressure P_o, Lamé's equations for the radial and hoop stresses at outside surface:

$$\sigma_r = P_o\frac{R_o^2}{R_o^2 - A_o^2}\left(1 - \frac{A_o^2}{R_o^2}\right) = P_o$$

$$\sigma_t = P_o\frac{R_o^2}{R_o^2 - A_o^2}\left(1 + \frac{A_o^2}{R_o^2}\right) = P_o\frac{R_o^2 + A_o^2}{R_o^2 - A_o^2}$$

Radial displacement of mandrel outside surface

$$U_1 = \frac{R_o}{E_1}(\sigma_t - M_5\sigma_r) = \frac{P_oR_o}{E_1}\left[\frac{R_o^2 + A_o^2}{R_o^2 - A_o^2} - M_5\right] \tag{5}$$

For equivalence, from equations 4 and 5

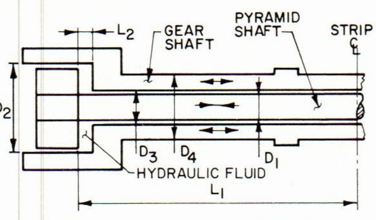
$$\frac{R_o^2 + A_o^2}{R_o^2 - A_o^2} - M_5 = \ln\frac{R_o}{\sqrt{2A}} + 1 - M_5$$

$$\therefore R_o^2 + A_o^2 = R_o^2A_2 - A_o^2A_2 + R_o^2 - A_o^2$$

$$\therefore A_o^2(2 + A_2) = R_o^2A_2$$

$$\therefore A_o^2 = R_o^2\left(\frac{A_2}{2 + A_2}\right) \quad \text{where } A_2 = \ln\frac{R_o}{\sqrt{2A}} \tag{6}$$

Axial stiffness of mandrel



Under action of tension F_1 in pyramid shaft

Stress in pyramid shaft	$= 4F_1/(E_3D_1^2)$	$= B_1F_1$
Pressure in hydraulic fluid	$= 4F_1/(E_3(D_1^2 - D_2^2))$	$= B_2F_1$
Stress in gear shaft	$= 4F_1/(E_3(D_4^2 - D_3^2))$	$= B_3F_1$
Elongation of pyramid shaft	$= F_1B_1L_1/E_1$	
Compression of hydraulic fluid	$= F_1B_2L_2/K$	
Compression of gear shaft	$= F_1B_3(L_1 - L_2)/E_1$	

Displacement of pyramid shaft with respect to segment, at strip center line

$$= F_1B_1L_1/E_1 + F_1B_2L_2/K + F_1B_3(L_1 - L_2)/E_1$$

Axial stiffness of mandrel $K_6 = F_1/\text{displacement}$

$$K_6 = 1/(B_1L_1/E_1 + B_2L_2/K + B_3(L_1 - L_2)/E_1) \tag{7}$$

Effect on radial and hoop stresses at bore of coil of winding one lap

In reference 2 the coil is treated as a thick walled elastic cylinder

From reference 2 equation 22 hoop stress =

$$E_2'\left(\frac{U_2}{r} + M_6\frac{dU_2}{dr}\right) \quad \text{where } E_2' = E_2/(1 + M_6)$$

From reference 2 equation 24 $\dfrac{U_2}{r} = C_2 + \dfrac{\overline{C_2}}{r^2}$

From reference 2 equation 25 $\dfrac{dU_2}{dr} = C_2 - \dfrac{\overline{C_2}}{r^2}$

$$\therefore Q_1 = E_2'\left[C_2(1 + M_6) + \frac{\overline{C_2}}{r^2}(1 - M_6)\right]$$

where

$$C_2 = \frac{1}{E_2'}\,\frac{1}{1 + M_6}\,\frac{1}{R_1^2 - R_o^2}(P_2R_1^2 - P_1R_o^2)$$

reference 2 equation 28

$$\overline{C_2} = \frac{1}{E_2'}\,\frac{1}{1 - M_6}\,\frac{1}{R_1^2 - R_o^2}(P_2 - P_1)R_1^2R_o^2$$

reference 2 equation 27

hence

$$\text{Hoop stress} = \frac{1}{R_1^2 - R_o^2}\left[P_2R_1^2 - P_1R_o^2 + P_2R_1\left(\frac{R_o}{r}\right)^2 - P_1R_o^2\left(\frac{R_1}{r}\right)^2\right]$$

$$\text{Hoop stress} = P_2\frac{R_1^2\left[1 + \left(\frac{R_o}{r}\right)^2\right]}{R_1^2 - R_o^2} - P_1\frac{R_o^2\left[1 + \left(\frac{R_1}{r}\right)^2\right]}{R_1^2 - R_o^2}$$

At coil bore $r = R_o$, hoop stress $= Q_1$

$$\therefore Q_1 = P_2\frac{2R_1^2}{R_1^2 - R_o^2} - P_1\frac{(R_1^2 + R_o^2)}{(R_1^2 - R_o^2)} \tag{8}$$

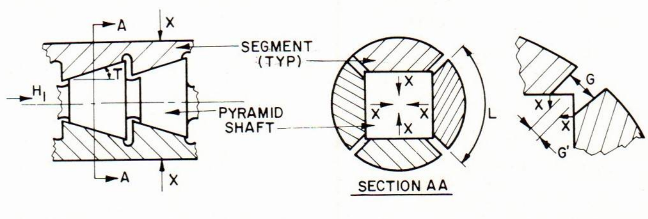
From reference 2 equation 7 $P_2 = S_1S/R_1$ (9)

From reference 2 equation 33

$$P_1 = \frac{2\dfrac{R_o}{E_2}\dfrac{R_1^2}{(R_1^2 - R_o^2)}P_2}{\dfrac{R_o}{E_1}\left(\dfrac{R_o^2 + A_o^2}{R_o^2 - A_o^2} - M_5\right) + \dfrac{R_o}{E_2}\left(\dfrac{R_1^2 + R_o^2}{R_1^2 - R_o^2} + M_6\right)} \tag{10}$$

Using equations 8, 9 and 10, the changes in radial and hoop stresses at the coil bore due to winding one lap are calculated.

Relationship between rotating cylinder piston displacement and mandrel diameter



Let D be the dia of curvature of the outer surface of segments (true circle diameter of mandrel).

G = size of gap between segments at the true circle dia
H_1 = axial movement of pyramid shaft relative to segments
x = radial movement of segments = $H_1 \tan T$

L = Length measured circumferentially along one segment

$$L = \frac{\pi D}{4} - G$$

For expansion or contraction of mandrel relative to true circle dia

$$\text{Circumference:} \quad C = 4L + 4G'$$

$$\text{where} \quad G' = G - \sqrt{2}x$$

$$\text{Effective diam of mandrel} = C/\pi$$

$$= \frac{4}{\pi}(L + G - \sqrt{2}x)$$

$$= \frac{4}{\pi}\left(\frac{\pi D}{4}\sqrt{2}x\right)$$

$$= D - \frac{4\sqrt{2}}{\pi} H_1 \tan T$$

i.e., if H_1 changes by H_2, mandrel dia changes by

$$\partial D = -\frac{4\sqrt{2}}{\pi} H_2 \tan T$$

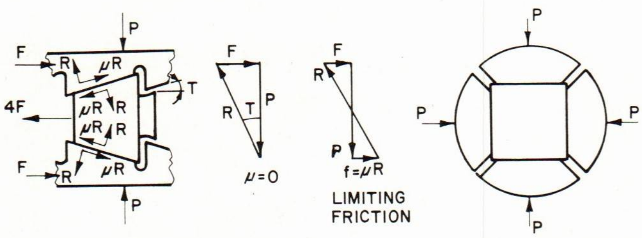
In general, movement H_2 will result in a change in the elastic strains and displacements of mandrel and coil. Assuming coil bore is in contact with mandrel all the time:

$$\partial D = 2\partial R_o = -2(\Delta U_2 - \Delta U_1)$$

hence

$$H_2 = \frac{2(\Delta U_2 - \Delta U_1)}{\frac{4\sqrt{2}}{\pi}\tan T} = \frac{\pi}{2\sqrt{2}} \frac{(\Delta U_2 - \Delta U_1)}{T_1} \qquad (11)$$

Condition for collapse of mandrel during winding



Mandrel starts to collapse when collapse forces P become sufficiently high to overcome static friction and cylinder force. At this condition $\mu = M_1$ and cylinder force $= 4F$

$$R \simeq \frac{P}{\cos T}$$

$$F \simeq P \tan T - M_1 R \cos T$$

$$F \simeq P \tan T - M_1 \frac{P}{\cos T} \cos T$$

$$F \simeq P(\tan T - M_1) \qquad (12)$$

$$P_M = \frac{P_o R_o}{\sqrt{2A}}(1); \; P = P_M W 2A$$

$$= \frac{P_o R_o}{\sqrt{2A}} W 2A = \sqrt{2} P_o R_o W \qquad (13)$$

At start of collapse, from equations 12 and 13 and putting $T_1 = \tan T$

$$F = \sqrt{2} P_o R_o W (T_1 - M_1) \qquad (14)$$

If cylinder force at start of collapse is F_1 then

$$F_1 = 4F = 4\sqrt{2} P_o R_o W (T_1 - M_1)$$

$$P_{o_1} = F_1/[4\sqrt{2} R_o W (T_1 - M_1)] \qquad (15)$$

If cylinder force at end of collapse is F_2, then, at end

$$P_{o_2} = F_2/[4\sqrt{2} R_o W (T_1 - M_2)] \qquad (16)$$

If $P = P_{o2} - P_{o1}$ and $\Delta F = F_2 - F_1$, then change in P_o during collapse is given by

$$P = P_{o_2} - P_{o_1} = \frac{1}{4\sqrt{2} R_o W}\left(\frac{-F_1}{T_1 - M_1} + \frac{F_1}{T_1 - M_2} + \frac{(F_2 - F_1)}{T_1 - M_2}\right)$$

$$P = \frac{-F_1(M_1 - M_2)}{4\sqrt{2} R_o W (T_1 - M_1)(T_1 - M_2)} + \frac{\Delta F}{4\sqrt{2} R_o W (T_1 - M_2)} \qquad (17)$$

Change in cylinder force, radial stress at coil bore and radial and axial displacements during collapse of mandrel when winding

$$\Delta U_1 = \frac{PR_o}{E_1}\left(\frac{R_o^2 + A_o^2}{R_o^2 - A_o^2} - M_5\right) = \frac{PR_o}{E_1}(K_1 - M_5) \qquad (18)$$

(K_1 thus defined. Reference 2 equation 32)

$$\Delta U_2 = \frac{-PR_o}{E_2}\left(\frac{R_1^2 + R_o^2}{R_1^2 - R_o^2} + M_6\right) = \frac{-PR_o}{E_2}(K_3/K_2 + M_6) \qquad (19)$$

(K_3, K_2 thus defined. Reference 2 equation 29, $P_2 = 0$ during collapse)

$$\text{From (11)} \quad H_2 = \frac{\pi}{2\sqrt{2}} \frac{(\Delta U_2 - \Delta U_1)}{T_1} = \frac{-\pi}{2\sqrt{2}} \frac{PR_o}{T_1} \times \left[\frac{(K_3/K_2 + M_6)}{E_2} + \frac{(K_1 - M_5)}{E_1}\right] \qquad (20)$$

Based upon observation that the collapse steps occur with great speed, the assumption is made that during each collapse step there is no time for oil to escape from the rotating cylinder.

The relative axial motion between pyramid shaft and segments is absorbed by elastic deformation of the mechanical parts and compression of hydraulic fluid in the rotating cylinder.

In this case $\Delta F = K_6 H_2$ (21)

From equation 17, define $K_9 = F_1(M_1 - M_2) \div [W(T_1 - M_1)(T_1 - M_2) 4\sqrt{2} R_o]$ (22)

From equation 20, define $K_7 = (K_3/K_2 + M_6)/E_2$ (23)

From equation 20, define $K_8 = (K_1 - M_5)/E_1$ (24)

From equation 17 and 22

$$P = -K_9 + \frac{\Delta F}{4\sqrt{2} R_o W (T_1 - M_2)} \qquad (25)$$

From equations 25, 21, 20, 23 and 24

$$P = -K_9 - \frac{\pi P R_o(K_7 + K_8)K_6}{2\sqrt{2}T_1 4\sqrt{2}R_o W(T_1 - M_2)}$$

$$\therefore K_9 = -P\left[1 + \frac{\pi K_6(K_7 + K_8)}{16WT_1(T_1 - M_2)}\right]$$

$$\therefore P = \frac{-K_9}{1 + \dfrac{\pi K_6(K_7 + K_8)}{16WT_1(T_1 - M_2)}} \qquad (26)$$

P is first calculated using equation 26 then ΔU_1, ΔU_2 and H_2 and F using equations 18, 19, 20 and 21.

Change in hoop stress at coil bore during collapse when winding

Equation 8 is used, putting $P_2 = 0$ and substituting P for P_1 and Q for Q_1

$$Q = -P\frac{(R_1^2 + R_o^2)}{(R_1^2 - R_o^2)} = -PK_3/K_2 \qquad (27)$$

Change in radial and hoop stresses at coil bore and displacements when coil is removed from mandrel after winding

When mandrel is collapsed to remove coil, radial stress at coil bore drops to zero.

$$\Delta P_o = -P_o$$

From equation 18 $$\Delta U_1 = \frac{-P_o R_o}{E_1}(K_1 - M_5) \qquad (28)$$

From equation 19 $$\Delta U_2 = \frac{P_o R_o}{E_2}(K_3/K_2 + M_6) \qquad (29)$$

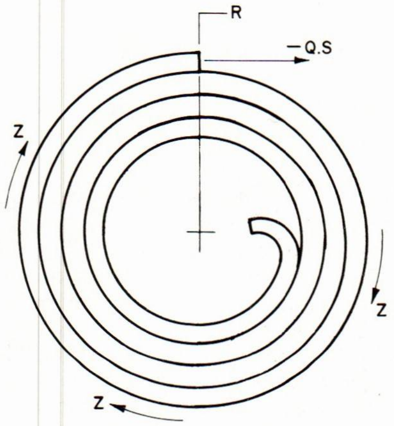
Piston stroke H_2 is then obtained using equation 11.

Definition of condition for validity of equations for coil

The equations which treat the coil as a thick-walled cylinder under internal and external pressure are only valid provided that no slippage occurs between laps of the coil.

In general, slippage will occur between coil laps if the torque being transmitted by the coil is greater than the torque which the coil is capable of transmitting by means of circumferential tension and friction between laps.

The slip ratio is defined as the ratio of maximum transmissible torque (no slip conditions) to actual torque. Only if the slip ratio is greater than unity throughout the coil will the coil retain its integrity as a thick-walled cylinder.



For unit width of coil; torque transmitted by coil; T_o = tension stress x thickness x coil outside radius

$$T_o \simeq S_1 S R_1 \qquad (30)$$

At any radius R inside the coil, if Z is the shear force on last lap up to radius R, and Q is the hoop stress (compressive +) at radius R

$$\text{Torque transmitted} = R(Z - QS) \qquad (31)$$

Maximum possible value of Z, if P_r is radial stress at R

$$Z_{\max} = M_3 Pr 2\pi R \qquad (32)$$

Maximum transmissible torque is T_o' where

$$T_o' = R(2\pi R M_3 Pr - QS) \qquad (33)$$

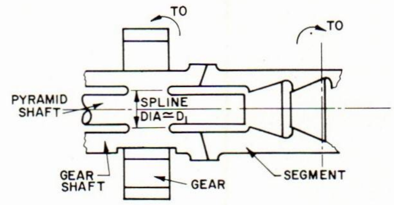
$$\text{Slip ratio} = T_o'/T$$

$$\text{Slip ratio} = \frac{R}{SR_1S_1}(2\pi R M_3 Pr - QS) \qquad (34)$$

This is generally a minimum when $R = R_o$, $P_r = P_o$, $Q = Q_o$ and this is the worst case

$$\text{Slip ratio}_{\min} = \frac{R_o}{SR_1S_1}(2\pi R_o M_3 P_o - Q_o S) \qquad (35)$$

Effect of frictional resistance to relative axial motion at drive splines and pyramids



Torque T_o is transmitted from coil, through segments to pyramid shaft, then through splines to gear shaft

$$\text{Torque force on splines} = 2T_o/D_1$$

$$\text{Torque force on pyramids} = T_o/\sqrt{2}A$$

assuming torque is effectively transmitted close to the corners of the pyramids. Resistance to relative axial motion is F_o where

$$F_o = \mu T_o(2/D_1 + 1/\sqrt{2}A) \qquad (36)$$

At start of coil collapse, if cylinder force is F_1 then

$$F_1 + F_o = 4F \qquad (37)$$

and equation 15 becomes

$$P_{o_1} = (F_1 + F_o)/[4\sqrt{2}R_o W(T_1 - M_1)]$$

where

$$F_o = M_1 T_o(2/D_1 + 1/\sqrt{2}A)$$

$$F_o = M_1 X_1$$

using equation 30 thus defining X_1. Hence equation 15 becomes

$$P_{o_1} = (F_1 + M_1X_1)/[4\sqrt{2}R_o W(T_1 - M_1)] \qquad (15a)$$

Similarly equation 16 becomes

$$P_{o_2} = (F_2 + M_2X_1)/[4\sqrt{2}R_o W(T_1 - M_2)] \qquad (16a)$$

Equation 17 becomes

$$P = \frac{-(F_1 + X_1T_1)(M_1 - M_2)}{4\sqrt{2}R_o W(T_1 - M_1)(T_1 - M_2)} + \frac{\Delta F}{4\sqrt{2}R_o W(T_1 - M_2)} \qquad (17a)$$

Defining

$$X_2 = K_9 X_1 T_1/F_1 \qquad (38)$$

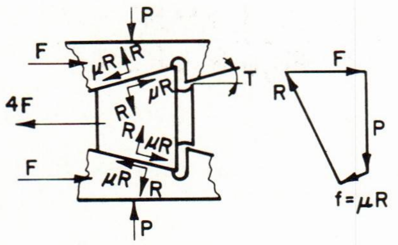
and noting

$$X_1 = S_1SR_1(2/D_1 + 1/\sqrt{2}A) \quad (39)$$

Equation 26 becomes

$$P = \frac{-(K_9 + X_2)}{1 + \dfrac{\pi K_6(K_7 + K_8)}{16WT_1(T_1 - M_2)}} \quad (26a)$$

Condition for expansion of mandrel during unwinding



Mandrel starts to expand when collapse forces P become sufficiently low to enable the cylinder force to overcome static friction and collapse forces. At this condition $\mu = M_1$ and cylinder force = $4F$

$$R \simeq \frac{P}{\cos T}; F \simeq P \tan T + M_1R \cos T$$

$$\simeq P \tan T + M_1\frac{P}{\cos T} \cos T$$

$$\simeq P(T_1 + M_1) \quad (40)$$

From equations 13 and 39 at start of collapse

$$F = \sqrt{2}P_oR_oW(T_1 + M_1) \quad (41)$$

If cylinder force at start of expansion is F_1, then, neglecting spline friction.

$$F_1 = 4F = 4\sqrt{2}P_oR_oW(T_1 + M_1) \quad (42)$$

$$P_{o_1} = F_1/[4\sqrt{2}R_oW(T_1 + M_1)] \quad (43)$$

Including effects of spline friction F_o which now resists cylinder force

$$F_1 = 4F + F_o = 4F + M_1X_1 \quad (44)$$

$$P_{o_1} = (F_1 - M_1X_1)/[4\sqrt{2}R_oW(T_1 + M_1)] \quad (43a)$$

If cylinder force at end of expansion is F_2, then, neglecting spline friction

$$P_{o_2} = F_2/[4\sqrt{2}R_oW(T_1 + M_2)] \quad (45)$$

Including effects of spline friction F_o

$$P_{o_2} = (F_2 - M_2X_1)/[4\sqrt{2}R_oW(T_1 + M_2)] \quad (45a)$$

If $P = P_{o_2} - P_{o_1}$ and $\Delta F = F_2 - F_1$, then, neglecting spline friction, change in P_o during expansion is given by

$$P = P_{o_2} - P_{o_1} = \frac{F_1(M_1 - M_2)}{4\sqrt{2}R_oW(T_1 + M_1)(T_1 + M_2)} + \frac{\Delta F}{4\sqrt{2}R_oW(T_1 + M_2)} \quad (46)$$

Including effects of spline friction

$$P = P_{o_2} - P_{o_1} = \frac{(F_1 + X_1T_1)(M_1 - M_2)}{4\sqrt{2}R_oW(T_1 + M_1)(T_1 + M_2)} + \frac{\Delta F}{4\sqrt{2}R_oW(T_1 + M_2)} \quad (46a)$$

Change in cylinder force, radial and hoop stresses at coil bore and radial and axial displacements during expansion of mandrel when unwinding

From equation 46 define $F_8 = F_1(M_1 - M_2) \div$

$$[4\sqrt{2}R_oW(T_1 + M_1)(T_1 + M_2)] \quad (47)$$

From equations 46 and 47

$$P = F_8 + \frac{\Delta F}{4\sqrt{2}R_oW(T_1 + M_2)} \quad (48)$$

From equations 48, 20, 21, 23 and 24

$$P = F_8 - \frac{\pi PR_o(K_7 + K_8)K_6}{2\sqrt{2}\pi 4\sqrt{2}R_oW(T_1 + M_2)}$$

$$F_8 = P\left[1 + \frac{\pi K_6(K_7 + K_8)}{16WT_1(T_1 + M_2)}\right]$$

$$P = \frac{F_8}{1 + \dfrac{\pi K_6(K_7 + K_8)}{16WT_1(T_1 + M_2)}} \quad (49)$$

This expression does not include the effect of spline friction

To include effect of spline friction

$$\text{defining} \quad X_3 = F_8X_1T_1/F_1 \quad (50)$$

Equation 49 becomes

$$P = \frac{F_8 + X_3}{1 + \dfrac{\pi K_6(K_7 + K_8)}{16WT_1(T_1 + M_2)}} \quad (49a)$$

P is first calculated using equation 49 or 49a then ΔU_1, ΔU_2, H_2 and F are calculated using equations 18, 19, 20 and 21. Hoop stress change Q is then calculated using equation 27.

REFERENCES

1) Sims, R. B. and Place, J. A. "The stresses in the reels of cold reduction mills" *Journal of Applied Physics, (British),* Vol. 4, July 1953, pp. 213–216.
2) Vater, M., Troost, A. and Wilkening, H. "Ermittlung der radialen Haspelbelastung beim Wickeln von ban förmigem Gut," *Bänder Bleche Rohre Dusseldorf,* 7, 1966, No. 3, pp. 135–141; No. 5, pp. 275–282; No. 8, pp. 479–490.
3) Wilkening, H., Doctoral dissertation. Aachen Technical High School, West Germany 1965, (same title as reference 2).

Discussion

presented by

Oberingenieur F. Lohmann, Demag AG, Duisburg, W. Germany

Dipl.-Ing. Dieter Leers, Plant Engineer, Stahlwerke Sudwestfalen AG, W. Germany

F. Lohmann: In the mid-1960's we put in operation, in three different plants, three similar ZR22 mills with winder blocks of the type shown in Fig. 4. These blocks performed very well for quite some time with strip tensions up to 83,000 lb. However, later on, one installation suffered fatigue failures of the segment hooks, (part 27, Fig. 4) but there were no problems in the others. The segments in the first plant were repaired by welding. Sometime later the same fatigue failures took place in the second plant and later on in the third. The failures were all clearly fatigue failures, and the question arose why were the times to failure so different for the three plants when the applied strip tensions and other conditions were very similar.

TABLE I **Mills having reverse pyramid winders**

Company	Mill type; matl.	Strip width, in.	Strip gage, in.	Rated tension × 1000 lb	Block dia., in.	Coil dia., in.	Top speed, fpm	Date winders installed	Builder
Allegheny-Ludlum Brackenridge, Pa.	22-50 Stainless	50	0.15-0.003	64	20	64	500	1974	Mull
Eastern Stainless Baltimore, Md.	22-49 Stainless	49	0.187-0.003	65	24	58	700	1974	Mull
DEW 2 West Germany	22-52 Stainless	52.8	0.275-0.016	79	24	71.3	660	1974	Demag
Tirgoviste Rumania	21BB52 Silicon steel	52	0.100-0.007	77	24	76.8	2460	1974	Demag
Acerinox 2 Spain	22B50 Stainless	50.4	0.236-0.008	88	24	78.7	1288	1974	Hitachi
Acerinox 1 Spain	22B50 Stainless	50.4	0.236-0.008	88	24	78.7	1288	1972	Hitachi
Krupp 1 West Germany	22-52 Stainless	52	0.196-0.012	77	24	68.9	820	1972	Demag
Alcan Arvida, Canada	4-HI 24/50 x 90 Aluminum	80 (Warm)	0.25-0.078	40	20	80	400	1971	Demag
Comsteel 2 Australia	21AA66 Stainless	69	0.315-0.018	30	20	72	400	1971	Hitachi
DEW 1 West Germany	23-43 Stainless	40.8	0.18-0.02	57	24	57.1	580	1970	Demag
DEW 3 West Germany	21BB61 Stainless	61.8	0.315-0.02	99	24	70.9	1640	1969	Demag
SWF 2 West Germany	21BB61 Stainless	60.2	0.300-0.02	99	24	70.9	1476	1969	Demag

Investigation established that the maintenance departments applied different lubrication practices, and lubricated at different intervals. After this, the mandrels were lubricated more frequently and with much more care. Also, in one plant, we installed indicators on the outboard bearings of the two winders so the operators could check the operation of the mandrels by observing whether the pyramid shafts shifted axially during winding as desired.

It was seen that on newly greased mandrels a uniform movement took place, but with continued operation, as the grease was squeezed out of the lubrication grooves due to the high pressure between the pyramid shaft and segments, the movement became abrupt and jerky and this condition produced excessive stresses on the segment hooks. The intervals between greasings were then reduced sufficiently to insure safe sliding of the pressure surfaces, and the segment hooks were strengthened as much as possible.

However, in spite of all of this, certain factors such as limitations of machining accuracy could result in one segment in a mandrel being more heavily loaded than the other three, and in such cases failures of the segment hooks sometimes occurred.

The operators learned to live with these problems, as these winder blocks were at that time of the best design available for use with alloy steel strip requiring very high winding tensions.

These difficulties led directly to the new reverse pyramid mandrel design.

In order to achieve the highest capacity, 30 Cr Ni Mo8 (closest equivalent to AISI 4330) was selected for pyramid shaft and segments, and this is probably the best steel for this application. Instead of previously used bolted-on bronze plates, bronze was deposited on the pyramid faces by welding, which allows optimal dimensioning of shaft and segment. Moreover, by virtue of close contact with mill users, many detail design improvements have been made since the first reverse pyramid winders at Deutsche Edelstahlwerke (DEW) in Krefeld and at Stahlwerke Sudwestfalen in Geisweid, which have been completely successful.

The reference list (Table I) shows that the older German mills have been retrofitted with the new reverse pyramid winders, and further improvements based on experience have been incorporated.

The new winder mandrels are now, apart from lubrication, maintenance-free, therefore, allow the mill users higher utilization and hence higher production.

One other interesting special development in which some of the reverse pyramid winder design principles were applied, and were well proved in practice, was the downcoilers for hot rolled aluminum strip in the new fully continuous aluminum strip casting-rolling mill installation at Alcan in Arvida, Canada. Here there are two mandrels which have been in operation since mid-1971.

Dieter Leers: This discussion covers the subject of maintenance of the controlled collapse winders. At Stahlwerke Sudwestfalen greasing is performed three times a week on the winders.

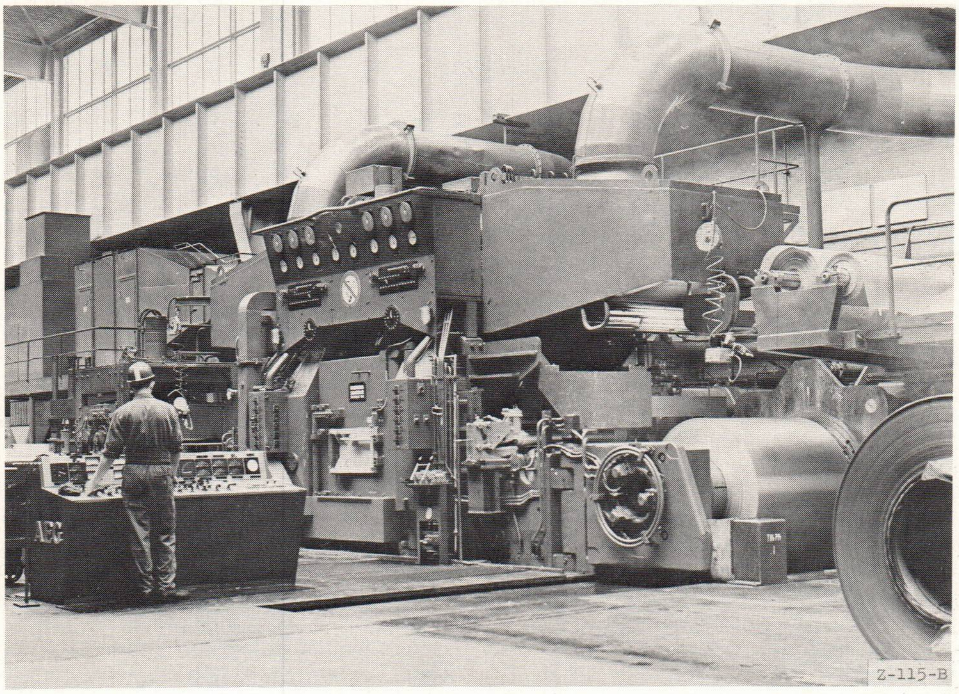
This is particularly important for the block segments. After 18 months of three-shift operation, the sliding surfaces between segments and pyramid shaft are scraped. The need for this becomes evident because the increase in friction coefficient arising from wear of these surfaces results in breaking of segment retaining T-keys during block collapse. The lips of the dovetail keyways (where segment retaining keys engage) have yielded on several occasions. They have been repaired by welding and on one occasion the shaft adaptor was replaced.

Max life of outboard bearing sleeves is two years in three-shift operation. The strip gripper has a max life of six months in three-shift operation (gripper teeth break). Strip gripper cylinders seal elements are replaced after six months as the packings become brittle resulting in leakage.

In conclusion, the winder block has satisfactory life. Apart from the limited endurance of these parts no other problems have occurred so far.

J. W. Turley: The contributions by Mr. Lohmann and Mr. Leers clearly confirm the point that it is absolutely vital to grease the collapsible mandrels frequently in order to insure min wear and reliable operation.

Mr. Leers discusses the problems associated with the segment retaining T-keys and associated keyways in the gear shaft adaptor. These keys were incorporated in the original designs at DEW and Sudwestfalen as a safety device and were intended to fail in the event that the mandrel was greased too infrequently. It is significant that these failures on the Sudwestfalen mandrels did not occur on the identical DEW mandrels, where greasing was more frequent. We understand that Sudwestfalen now grease more frequently and this is no longer a problem.



Sendzimir mill ZR21BB-61 with reverse pyramid controlled collapse winders installed at the Krefeld, W. Germany, works of Deutsche Edelstahlwerke AG

Printed in the U.S.A.