PEER REVIEWED

YANKEE DRYERS

Yankee dryer development through stress analysis

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ABSTRACT

Stress analysis has guided the development of the Yankee dryer since its infancy in the mid 1800s. This paper traces the development of the Yankee dryer from a primitive drying cylinder to a modern pressure vessel of extraordinary productivity. The review of Yankee history also tracks the evolution of stress-analysis concepts and techniques from rudimentary equations to computer-based finite-element analysis. Stress analysis is shown to be an indispensable tool for understanding and resolving operating problems and, most importantly, for providing a methodology for running this critical piece of equipment safely.

Application:

Examines the role of stress analysis in the evolution of the Yankee dryer and its importance in operating and maintaining existing Yankees.

THE YANKEE DRYER IS THE corner-stone of the crepe tissue process. In less than a single turn of this large cast-iron pressure vessel, the tissue web is pressed, dried, and creped. Today, the surface speeds for tissue-making can exceed 2000 m/min. **Figure 1** is a photograph of a modern Yankee dryer.

The history and development of the Yankee dryer is a specialized subcategory of cylinder drying. Hunter *(1)* provides interesting background on early paper drying, noting that papermaking drying cylinders were first patented in England in 1820. Prior to this time, the paper-drying process was a no-tech affair, with workers simply hanging handsheets out to dry.

Drying cylinders, particularly in the United States, were first heated with charcoal fires. In 1839, the improved "Ranson" dryer was patented, and steam gradually became the common method of heating. In 1830, George Spafford of Phelps and Spafford built the first known U.S. drying cylinders *(1)*.

Early Yankees, more appropriately referred to as machine-glazed (MG) cylinders, were a natural extrapolation of the small-diameter multiple "can" dryers. Early paper-machine speeds were painfully slow; and some machines were actually hand driven *(1)*. For slow speeds—and in particular for lightweight, permeable sheets—increasing the dryer diameter (and thus the drum surface area) allowed the simple elegance of drying on only one cylinder.

Large cylinders started with designs similar to the smaller dryers. Before 1850, enhancements in foundry practices, machining, and handling capabilities permitted the production of large cast-iron structures. Hill, who wrote a history of papermaking in Britain *(2)*, claims that in 1843, a cylinder-mold machine with one large dryer above the vat was built, similar to the one depicted in **Fig. 2**.

(Large-diameter drying cylinders were not commonly installed in England until the 1880s, after they had become popular in America—a plausible origin for the name "Yankee." This is buttressed by an account of the first paper machine in Michigan in 1839. The paper machine was approximately 9 m long, and the paper was dried on a 10-ft-diam. cylinder heated by charcoal *(1)*. There are several anecdotes relating to the name "Yankee," so it is virtually impossible to establish a definitive origin of the term. The earliest record of a large drying cylinder in TAPPI's Yankee database is 1884 *(3)*. Little documentation has been found on the transition of Yankees from charcoal to steam.)

Machine speeds, nip pressures, and steam pressures were initially very low. Early speeds ranged from 1 to 100 m/min. Cylinder pressure expectations were typically less than 200 kPa. Note that the early cylinder in Fig. 2 does not appear to have a center stay, which has become one of the hallmarks of a Yankee. Records before 1900 show dryers of 7-, 8-, 9-, and 10-ft diam. were produced, and widths were mostly less than 2.5 m *(3)*.

DRYER-SHELL DESIGN CALCULATIONS

Designing cylinders to support charcoal fires was not particularly challenging. However, designing vessels for steam containment was quite another matter. The successful steam boilers entering the 19th century were capable of 7–15 kPa, and the engines developed power by collapsing steam to a vacuum *(4)*. Vessels of higher pressure were considered extremely unreliable, and explosions were not uncommon. The efficiencies of higher pressure awaited a workable design theory.

In 1833, French engineer Gabriel Lamé published the classical relationships between pressure and radial and tangential stress in cylinders *(5)*. Hoop stress provides a fundamental criterion of pressure-vessel design and has become the backbone of modern codes. Longitudinal stresses at mid-shell are a lesser con-

1. Yankee dryer

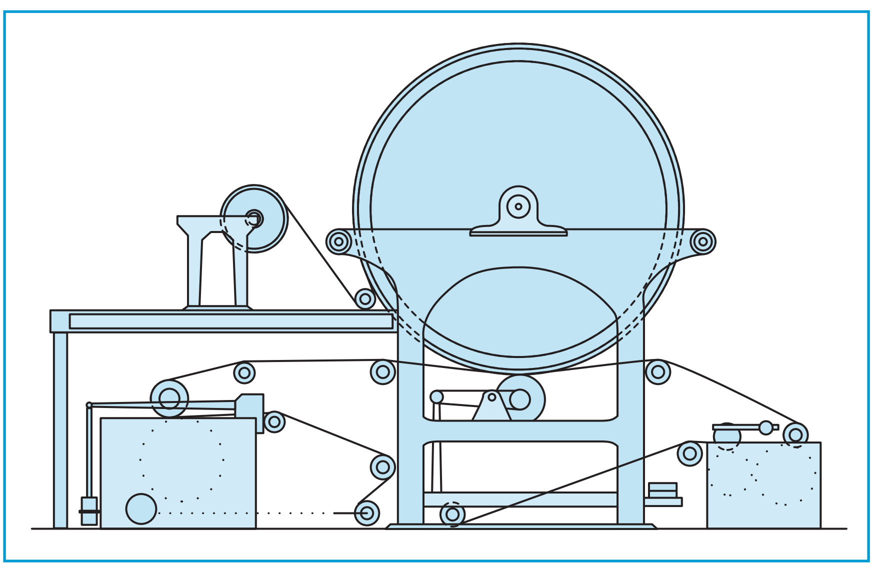
cern, since they are only half as large as the hoop stresses. The tangential stress distribution is parabolic through the shell, peaking on the inside diameter, as seen in **Fig. 3**.

For vessels of large radius-to-thickness ratios, such as can dryers and the larger Yankee cylinders, the pressure-induced hoop stress through the wall distribution is essentially flat. A simple formula for determining hoop stress is derived from a free-body diagram cutting a cylinder lengthwise, as seen in **Fig. 4**. The wall reaction force necessary to balance the projected pressure load is then divided by the cross-sectional area of the wall. The result is Eq. 1.

$$S = \frac{Pr_m}{t} \qquad (1)$$

where

S = hoop stress
P = pressure
r_m = mean radius of the cylinder
t = shell thickness



2. Pre-1850 Yankee machine

Since 1914, the ASME Code has attempted to maintain the simple nature of the hoop-stress formula, but some adaptations have been added to stay abreast of cylinder evolution. In Section VIII-1 Appendix 1 of the ASME Code *(6)*, Eq. 2 is offered for cylindrical shells.

$$t = \frac{Pr_o}{SE + 0.4P} \qquad (2)$$

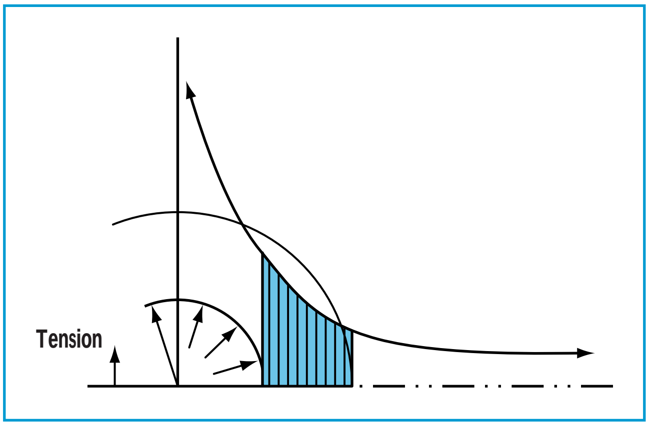
where

t = minimum required thickness, in.
P = internal design pressure, psi
r_o = outside radius of the shell course, in.
S = maximum allowable stress value, psi (10% of the tensile strength for cast iron)
E = longitudinal joint efficiency or ligament efficiency between any openings (1 for integrally cast shells)

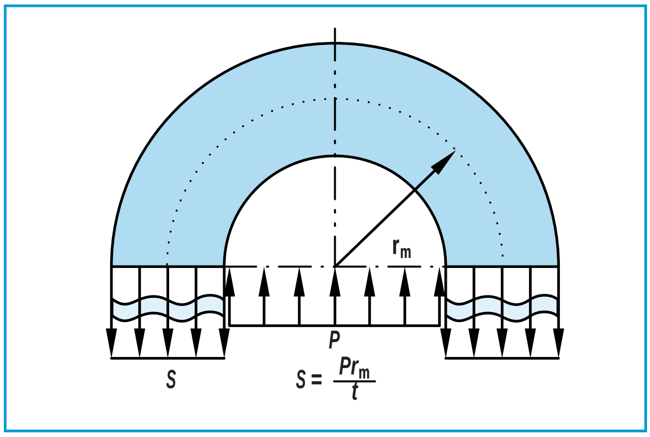
Equation 2 is solved for t but is also solved for P in Section VIII-1 Appendix 1 of the ASME Code. Using outside radius r_o in the calculation is more convenient than using the inside radius version of ASME Code UG-27 or than computing the mean radius r_m from the inner and outer radii. The denominator term $0.4P$ results from an adjustment of the mean radius outward by $0.1t$ to conservatively approximate the peak stresses of the Lamé formulation.

Equation 2 is valid for thicknesses up to half the radius of the shell. For typical Yankee diameters, the peak and mean stresses are well within 0.5%. For plain-shell Yankees, t was—in those days—limited to about 2 in. by the practicalities of casting and heat-transfer limitations. By the close of the 19th century, field experience showed that this thickness generally was satisfactory in accommodating the other loadings a Yankee is called upon to bear, although at that time, the stresses imposed were unfathomable. There is a tradeoff between P and r_o to hold stress S to a reasonable level.

Roughly speaking, the maximum tensile strength of cast iron increased by 70 MPa per decade from 1910 to 1960 *(7)*. In other words, 70 MPa was a practical maximum before 1910, and by 1960 the fully pearlitic irons could reliably give 420 MPa (i.e., Class 60) tensile strength. This hoop-stress guideline suggests that pre-1910 Yankees



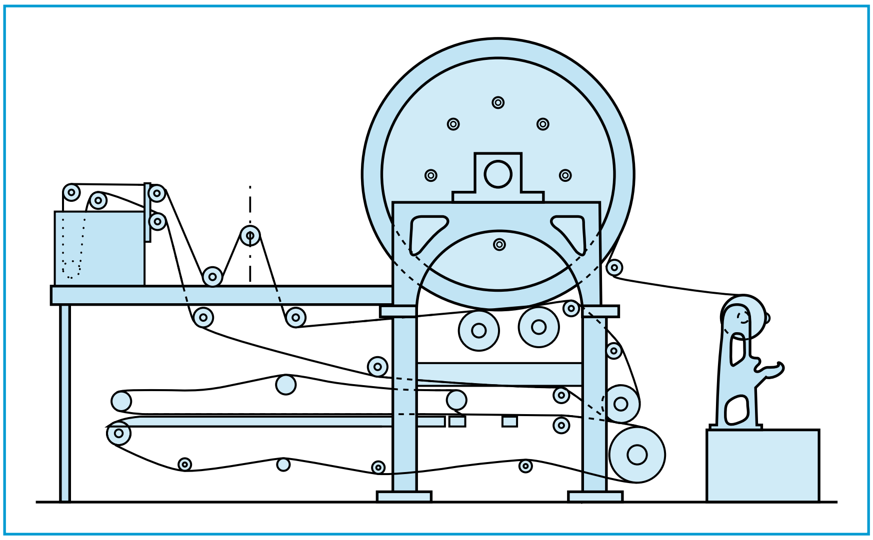
3. Tangential stress in internally pressurized cylinder



4. Average pressure-induced hoop stress

would be pushing their stress limits going to 12 ft in diameter, and indeed, 10 ft was the limit for pre-1920 dryers.

Tensile stress documentation for the early dryers is generally absent and has to be determined by destructive testing or by nondestructive tests such as hardness or ultrasonic evaluation. As can be imagined, the early castings could also be considerably nonuniform. Typically, the microstructure was very mixed, having some pearlite, ferrite, and various carbide and sulfide structures, all occasionally interrupted by gross porosity and voids.



5. Yankee fourdrinier circa 1900

DRYER-HEAD LOADING CONSIDERATIONS

Pressure forces, bending moments from supporting journals, and torsional reactions are the primary head loads on Yankees. In early vessels, lacking a center stay, the shell would be called upon to absorb these stresses. Early on, it became apparent that the shell was going to need help supporting these loads in a vessel of large diameter. However, too much space and material would be sacrificed to contain end stresses efficiently in heads made into hoop or dome structures. Unstayed flat heads needed considerable thickness or reinforcement to prevent unacceptable stress and deflection of the head.

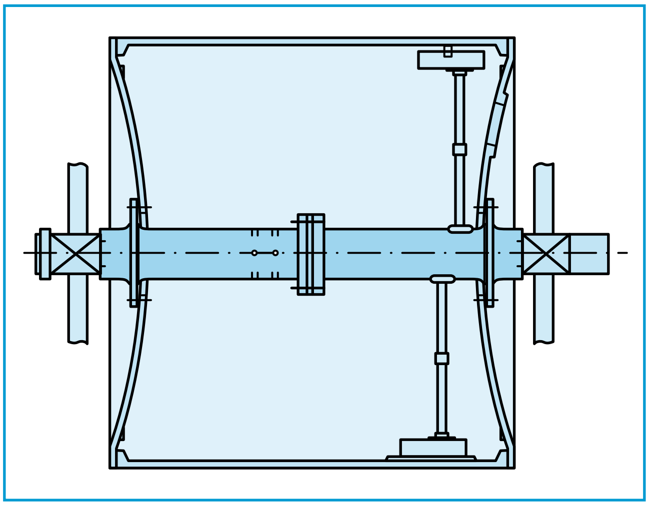
The early designers turned to the redundant stay or tie rod. Stays were age-old devices for supporting the flat ends of boilers, usually a natural result of having installed tubes. **Figure 5** shows a Yankee adapted to a fourdrinier from about 1900 *(2)*. The builder appears to have been wrestling with an end-load containment problem, since the ends of tie rods are visible on the heads.

Ultimately, designers would learn to control radial and circumferential positioning of the stay to limit the flexural stresses in the flat heads. Tie-rod redundancy solved this problem for early designers. Separate stay bars are still a part of one modern Yankee design, but bending stresses from centrifugal loads on the bars do impose limits to speed in this case.

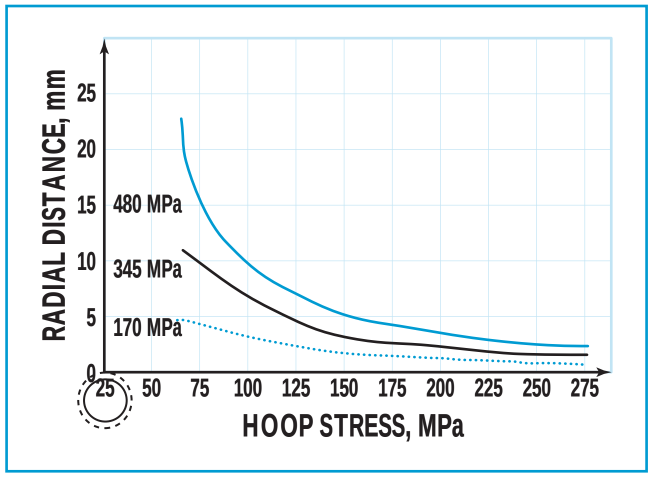
Unlike boilers, which generally are heavily insulated, Yankee external shell surfaces are cool relative to the inside. This presents a stay prestress issue to the designer, who must correctly anticipate differential thermal expansion. Threaded tie rods are appealing from an assembly adjustment standpoint, but sealing, centrifugal flexure, and thread fatigue issues have rendered them obsolete.

Figure 6 shows a Yankee from about 1904 *(7)*. The center stay had arrived, and further progress had been made in cylinder end closure. Extending the journal shaft through a rotating vessel is a very simple method of staying heads and absorbing journal bending loads at the same time.

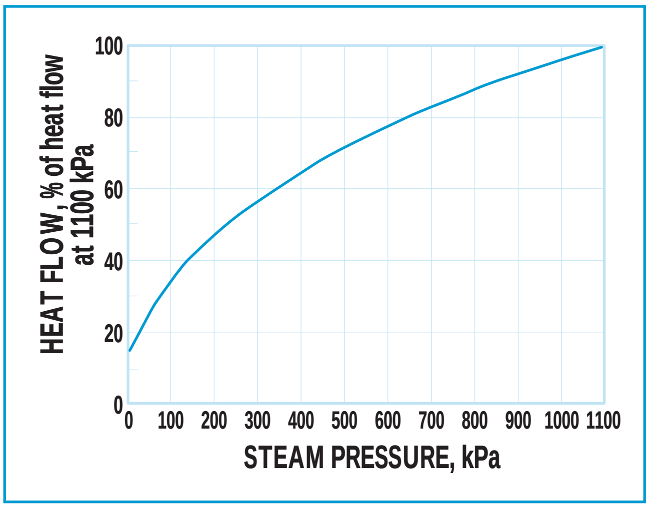
In the earliest Yankees, the center rod, if used at all, was probably a convenient hanger for a basket of charcoal. When vessels went over to steam, the center stay became part of the plumbing for admitting steam and removing condensate. The early stay was a 14-in. cast pipe-like tie rod



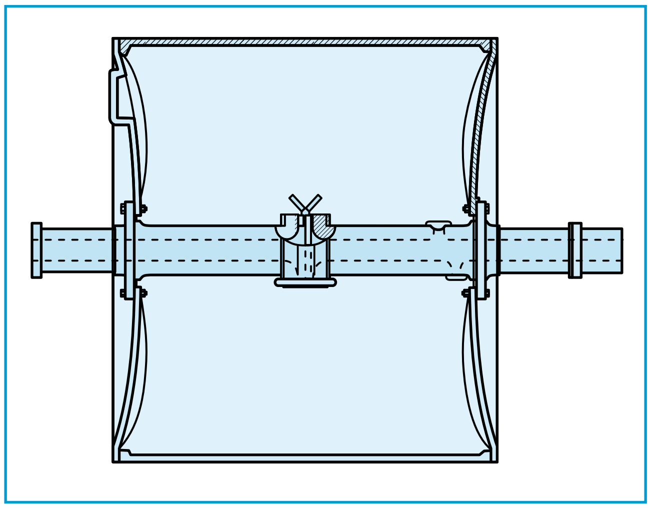
6. Yankee dryer (10-ft diam.) circa 1904



7. Hoop stress around 1-in. tapped hole for three levels of fastener prestress



8. Heat flow rate vs. steam pressure



9. Yankee dryer (12-ft diam.) with ribbed heads

between the heads that was blanked at the center to provide for steam inlet and condensate outlet at opposite ends.

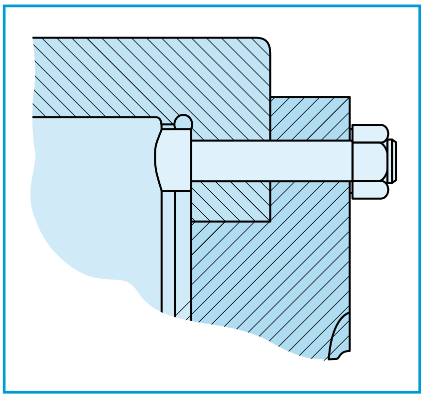
Stress analysis of the stay as a beam with bending stress, MC/I, superimposed with tension stresses, P/A, was quite straightforward. Taylor-Forge published a stress analysis technique in 1937 for sizing integral flanges, considering them to be a moment-loaded ring *(8)*. The flange analysis methodology was gradually absorbed into the ASME Code.

Historically, Yankee stays have grown in size with dryer diameter to properly balance the head span. Structural problems with stays are rare. Nevertheless, inspections can be useful to confirm the integrity of older dryers as a benchmark. Thereafter, it is prudent to check at five-year intervals to see that no damage is occurring. The following are suggested inspection steps.

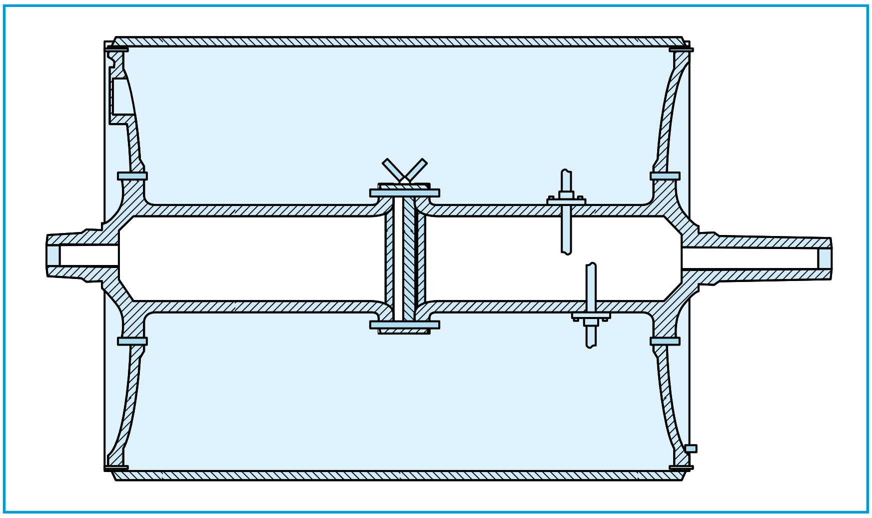
- Visually inspect for erosion. A stay can be eroded inside from steam and condensate activity. Internal access can be restricted, but modern flexible fiber scopes can make this check fairly easy.
- Test fasteners/rod bolts for stress corrosion cracking. Torque checks and ultrasonic inspection are frequently used.
- Use magnetic particle testing to rule out cracking of flange fillets.

SECONDARY STRESSES

The stresses discussed up to this point are considered to be primary and are thus protected in pressure-vessel design by large safety factors. Primary stresses are directly proportional to applied loadings such as pressure. Secondary stresses, in contrast, are "developed by constraint of adjacent parts or by self constraint of a structure" *(9)*. Secondary stresses tend to be localized in nature and self-limiting, but they can be of considerably greater magnitude than primary stresses. In static vessels, secondary stresses are not of great concern, but in rotating vessels such as Yankees, fatigue consequences elevate their importance.



10. Through-bolted joint



11. Yankee dryer (12-ft diam.) circa 1930

Examples of secondary stresses

Thermal gradients. Thermal stress is a self-balancing stress produced by a nonuniform distribution of temperature or by differing coefficients of expansion. The purpose of a dryer is to dry paper, which requires heat flow from inside to outside the shell. The hotter insides are expanded relative to the outside of the shell, creating a tensile stress state on the outside diameter (OD) balanced by compression on the inside diameter (ID). Thermal boundary conditions dictate the amount of heat flow and resulting thermal gradients. Examples of the secondary thermal stresses considered for Yankee dryers include (a) axial and circumferential shell stresses resulting from radial temperature distributions in the shell and (b) axial stresses resulting from the differences of mean temperatures between the shell and the stay.

Warmup conditions also introduce thermal gradients. A gradual warmup with the dryer rotating is always recommended to avoid thermal shock. Rapidly applied, temperature changes produce twice the stress of a uniform gradient. A concentrated water jet on a dryer surface can instantly transmit water temperature to the shell. Such thermal shocks should clearly be avoided.

Discontinuity stresses. Changes in a vessel section result in stress concentrations. Examples are shell corners and the bases of flange junctions or reinforcement junctions, such as around a manhole. The ASME Code recognizes that these stresses could approach four times the hoop design allowable in static vessels. Finite-element analysis has allowed closer control of these stresses in modern dryers. The important consideration is that discontinuity areas should be examined by nondestructive test methods when benchmarking older vessels.

Fastener insertion stresses. Threaded plugs or cap screws can develop very high local tensile hoop stresses in the material surrounding the shank during installation. Wedging occurs because of the inclined-plane behavior of threads. Moreover, the standard pipe plugs have a tapered shank.

Figure 7 shows that the hoop effect on a female thread is extremely localized, but stresses are high enough to act as a possible crack-initiation site with a small boost from the surrounding stress field. The thread at the surface of a blind tap is also the most highly loaded, which concentrates the critical splitting stress near the interface.

The clamped part in a cap-screw joint reinforces the highly stressed area, and the splitting force is highly protected in most applications and therefore is frequently ignored. In Yankee dryers, however, superimposed stresses can change the situation for the worse. Bending stresses can open a cap-screw joint and strip away the protective clamping. Thermal gradients also can put a high tensile field around a threaded plug and start a crack.

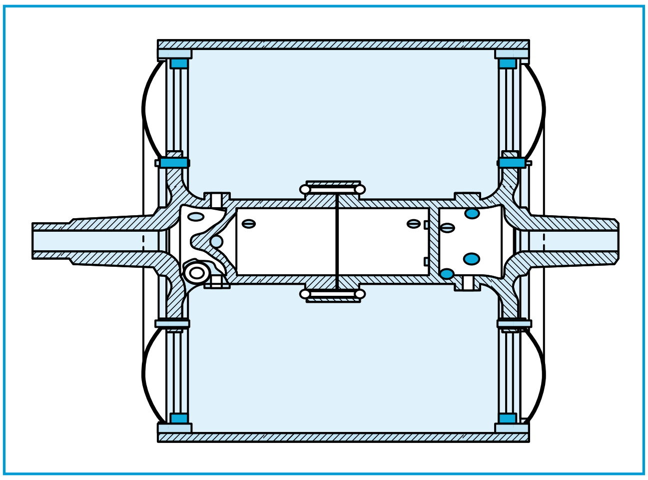
Blind taps have been used successfully in many areas of Yankees where stress is low. However, plugs and cap screws have been problematic when installed near free surfaces, such as external corners or the tops of ribs and, most importantly, when exposed to the high tensile fields on the outside of the shell.

Drive plugs and thermal sprays are common repair techniques that have been used very successfully to restore Yankee shell surfaces. Through-threaded plugs and metal-stitching-type fixes have been much less successful in shells and should only be used as a last resort.

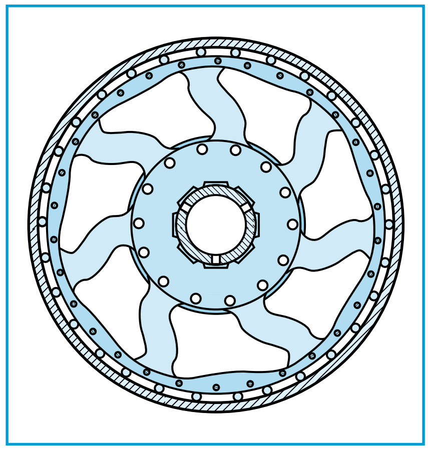
Head–shell cap screws near the shell surface likewise have been crack sites. Fastener wedging stresses have been dealt with historically by using buttress threads, eliminating blind taps, and using through-bolt flanges with separate nuts.

Peak stress

Secondary stresses do not cause problems acting alone, but they can be catastrophic when acting in combination with and superimposed upon one or more primary stresses. The resulting peak stress is of critical importance in a fatigue environment. Yankee designs have progressed to eliminate many built-in secondary stress concerns, and manufacturer derate data have provided



12. "Donut" head Yankee dryer (12-ft diam.) circa 1940



13. Donut-head Yankee spokes

wall thickness, and preexisting flaws were some of the contributors to early head problems.

A large flat disk is a fairly challenging casting to make. The ASME Code *(6)* offers an equation to guide flat-head design.

$$t = \sqrt{\frac{CP}{SE}}\, d \qquad (3)$$

where

t = minimum required thickness
d = head span
C = a constant depending on closure type (typically 0.25)
P = maximum allowable working pressure
S = maximum allowable working stress
E = joint efficiency (1 for a seamless head)

In UCI-23, the ASME Code notes that maximum allowable stress S for bending in cast iron is 1.5 times S for tension. Cast iron in compression is twice as strong as cast iron in tension. Graphite flakes impart a slit-like behavior to the cast-iron matrix, causing the lower tensile strength and stiffness.

Flexure situations that juxtapose compression and tension fields accent the material nonlinearity of cast iron, since the neutral axis shifts to accommodate the different stiffness. Cast iron is therefore notably more tolerant of a flexural environment than a tensile environment, which is fortunate, since the higher stresses brought to bear on dryers are flexural. The designer can exert control over head stresses by limiting pressure, increasing head thickness, shortening the span (such as by using a larger stay), and advanta-

substantive control over peak stresses. User recognition and avoidance of extraordinary loads and dubious alterations is also critically important. A knowledge of what can cause cracking and where the significant stresses can act is an indispensable part of designing a monitoring program to safeguard Yankees. The ensuing history of modern Yankee dryers is the story of the struggle to recognize and gain control of peak stresses.

CHRONOLOGY OF YANKEE DEVELOPMENT

Genesis

Roll-form toilet tissue made its appearance in 1871, and by 1899 *(1)* it had become a household standard. Odds are that it was originally neither creped nor very soft.

Much of the early manufacture of tissue was in the Paper Valley of Wisconsin, since the raw material for wood pulp was available and MG (machine glazed) technology was already established. Most major modern tissue producers can tie their origins to this place and time. The typical Yankee of this period was 10 ft in diameter and 3.35 m wide.

The fledgling crepe industry got off the ground during the decade of World War I *(2)*. Crepe paper demands for bandage material coupled with the product demands from an increasingly urbanized society were sparking a new industry. In addition, market demand was growing rapidly in nearby Chicago. The stage was set for the development of the Yankee dryer.

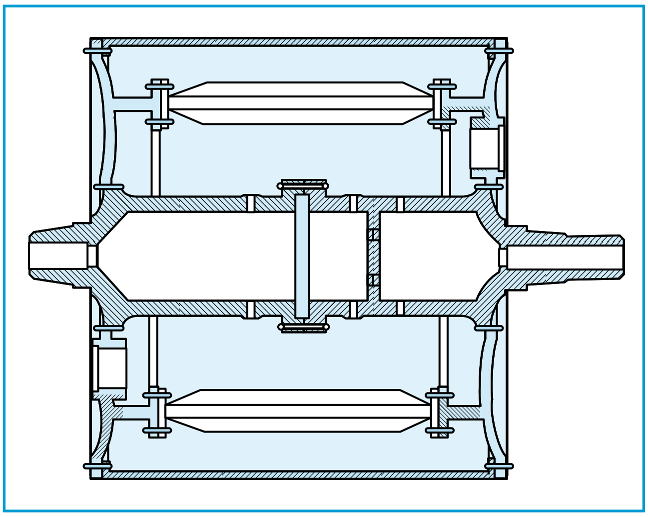
The 1920s

The 12-ft-diam. × 3.35-m-wide Yankee was developed during the "Roaring '20s." Production demands drove pressure ratings from 200 to 300 kPa by the end of the decade. **Figure 8** shows how heat increases through a plain Yankee shell with increasing steam pressure. Steam temperature is not linear with pressure, such that the heat flow at 200 kPa is nearly half that at 1100 kPa. The declining rate of heat-transfer increase with pressure, coupled with increasing runnability problems from surface and edge overheating, eventually drove Yankees to larger diameters rather than increasing pressures.

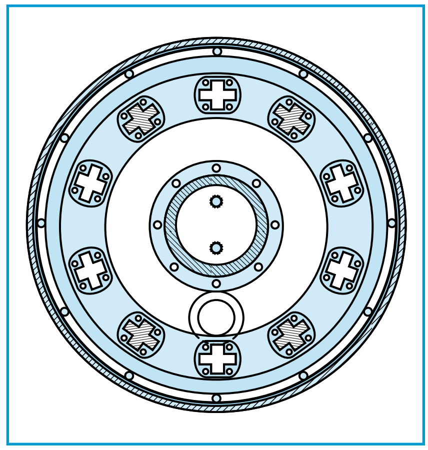
Some front-running paper machines had reached production speeds of 300 m/min by the opening of the decade. Basic Yankee construction remained the same, but cast iron was beginning to improve.

There were early signs that designers were grappling with the flat-plate behavior of the heads. Ribs appeared on the insides of the heads, as seen in **Fig. 9**. The growth in diameter had increased the head-to-stay span, and increasing pressures led to some early head-cracking problems. Low-tensile-strength iron, minimal geously adjusting redundant support of stay and shell.

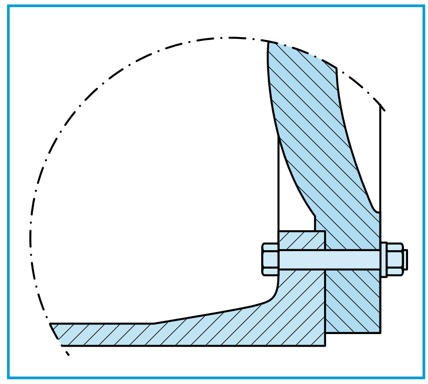
The 1930s
Yankee demand continued to grow in the Depression-era 1930s. Twelve-ft-diam. × 4.725-m-width, 520-kPa dryers were being produced at the end of the decade. The ASME Code was gradually becoming more accepted and became a standard for domestic Yankee construction in the early 1930s.



14. Stayed-head Yankee



15. Stay-bar arrangement



16. Tapered shell end

Some new Yankee problems became apparent. In addition to the aforementioned head problems, some shell edge cracking was seen, some of which was catastrophic. Since the cracking initiated at head–shell cap-screw cross sections, the wedging action of the tapered thread form was suspected as a contributor.

The through-bolted joint, illustrated in **Fig. 10**, was a solution and became common after the mid-1930s. The Yankee design had matured in cross section—with larger-diameter center stay, thicker heads, and anti-friction bearings—to a configuration that would remain typical into the 1960s. **Figure 11** shows a 1930s-type 12-ft Yankee. Important domestic Yankee suppliers at this time were Allis-Chalmers, Beloit, and Pusey & Jones.

The 1940s
The 1940s brought World War II and the attendant restrictions of the War Production Board, but tissue demand was high, and Yankee development continued. Despite restrictions, more than 20 Yankees were produced during the war years, and some old dryers were refurbished. At the close of the war, an important domestic producer of dryers, Newport News Shipbuilding and Dry Dock Company, began Yankee production.

Typical steam pressure at the end of the decade was 690 kPa, and widths went to 4.575 m. The closing years of the 1940s saw the first high-pressure 1100-kPa Yankee, the torus or "donut" head *(10)*, which is illustrated in **Fig. 12**.

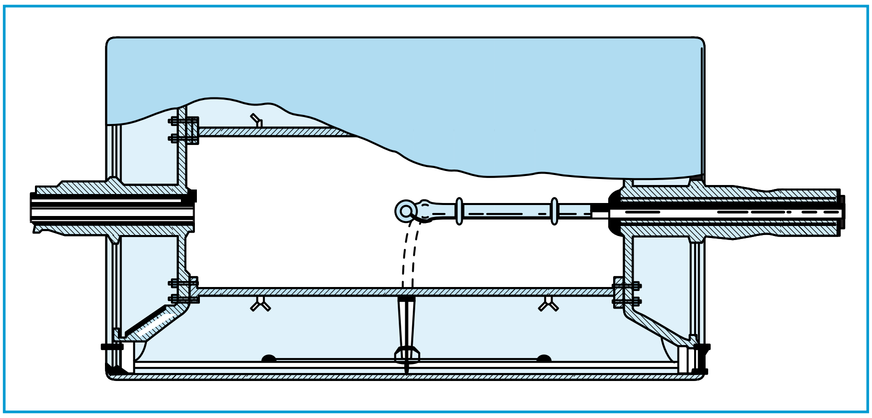
Steel toroidal heads contained pressure stresses in membrane fashion instead of bending and thus could be made thinner. Steel typically also has higher allowable design stresses. **Figure 13** shows the spokes that provided radial stiffness. Shells in these Yankees were early Class 60 iron, although shells made from iron with tensile strengths ranging from 275 MPa (Class 40) to 350 MPa (Class 50) were common.

The toroidal head was an understandable attempt to take advantage of higher-pressure and -temperature steam to increase heat transfer and production. However, excessive complexity and head protrusion—resulting in long journals and wide frame centers—limited the usefulness of this design.

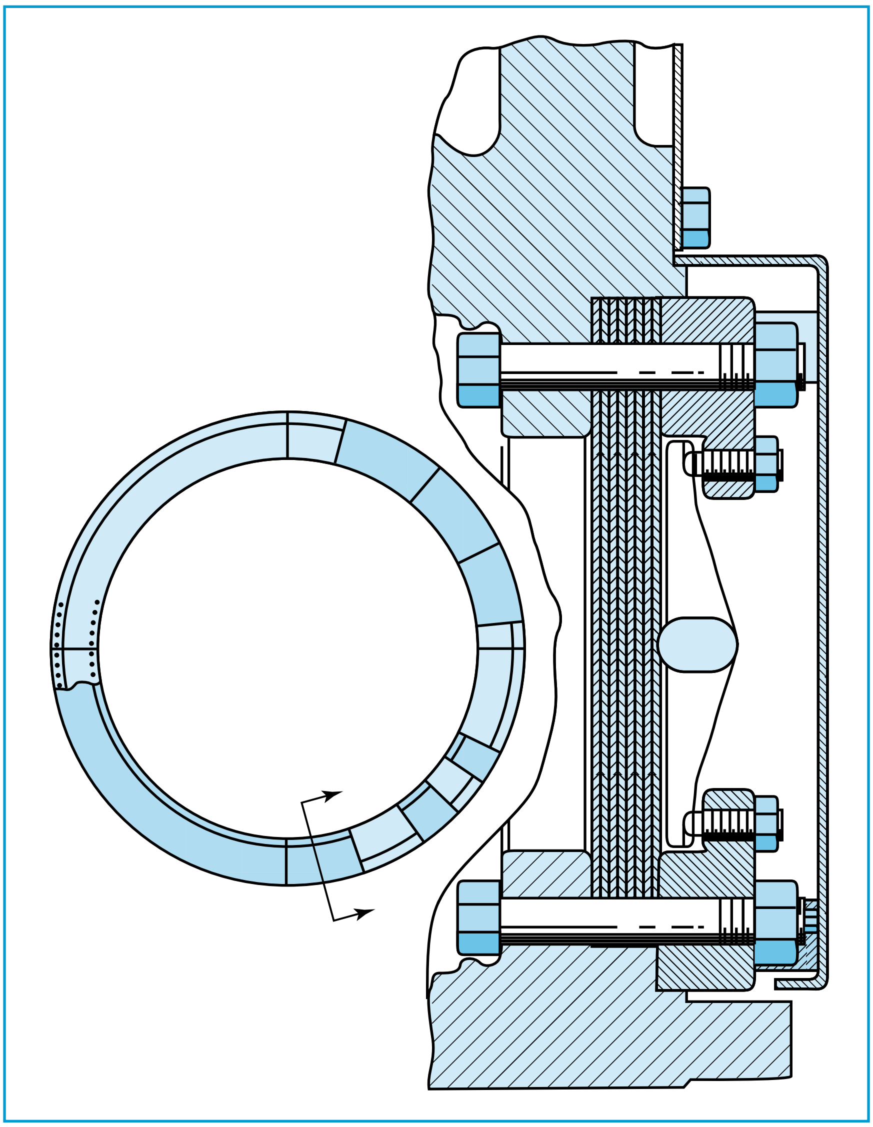
The 1950s
The 1950s was an era of intense innovation in Yankee dryers. The quest was on in earnest for better heat transfer and improved edge stability of the dryers. Much plate/shell analysis was going on using the methodologies of Roark and Timoshenko. There was a growing realization that square-cornered Yankee head–shell connections had edge-turning moments that required thicker shells for proper stress containment, and there were stability and heat-transfer-loss ramifications. The discontinuity stress in the shell corner was being looked at with interest.

The stayed-head design *(11)*, illustrated in **Fig. 14**, used a ring of staybars (**Fig. 15**) in addition to the center stay to provide additional head support. Shell ends "matured" to a tapered configuration to help contain the shell end moment and provide a shell of minimum thickness for optimum heat transfer to the sheet. The tapered shell end is illustrated in **Fig. 16**.

Another design attempt to reduce shell end moments was the flex head *(12)*, illustrated in **Fig. 17**. A multiple-ply stack of stainless-steel segments separated by gaskets (**Fig. 18**) was installed in the head to dissipate end moments from pressure



17. Flex-head Yankee dryer (12-ft diam.)



18. Flex-head assembly

and differential expansion. Several dryers were built and ran many years, but leakage doomed the design. The construct did, however, permit a record 12-ft-diam. × 6.555-m-width dryer.

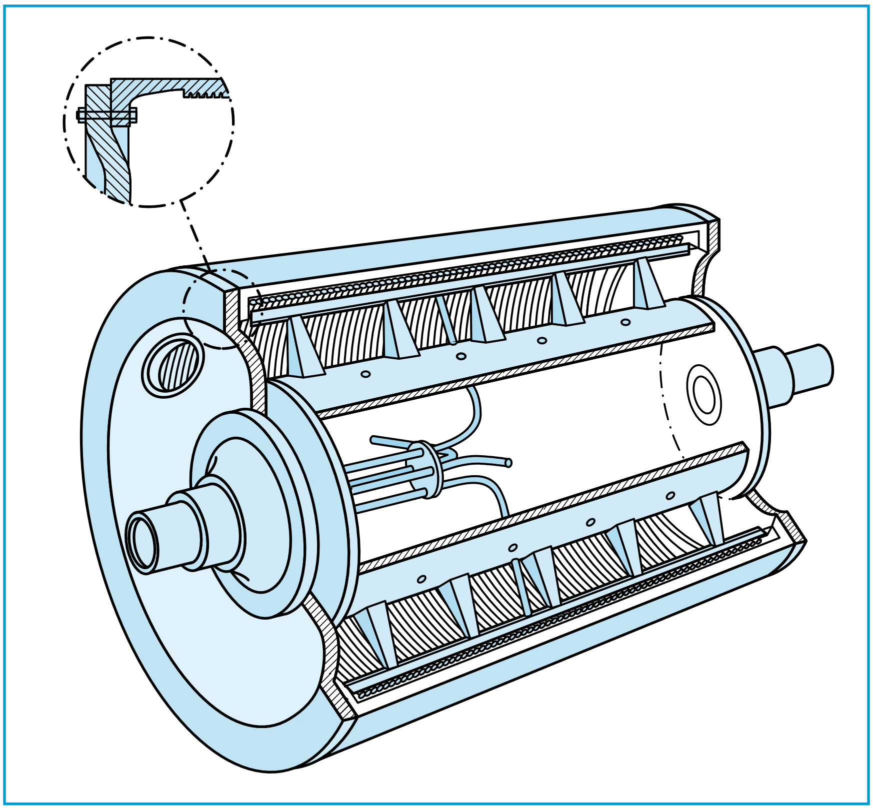
One dangerous attribute this design shared with most other Yankees of the time was the shell flange extension. The extension was conceived as a simple way to stretch the shell and provide a protective roof over the head–shell joint, not unlike the head flange common to can dryers that supported the rope groove. Operating experience revealed that the extension on Yankees behaved like an overactive cooling fin, particularly where edge sprays were used. Very high hoop stresses resulted, and catastrophic failures occurred *(13)*.

Additionally, and most ironically, some recent examples of hard-pack corrosion wedging have been discovered under some of the few remaining edge extensions. Wedging has been severe enough to cause radial distortion and very high additional hoop stresses. Again, a catastrophic failure has resulted *(14)*. The standard treatment for the shell extension has been to machine it off, first checking with the original manufacturer to ensure that sufficient rigidity remains without the flange.

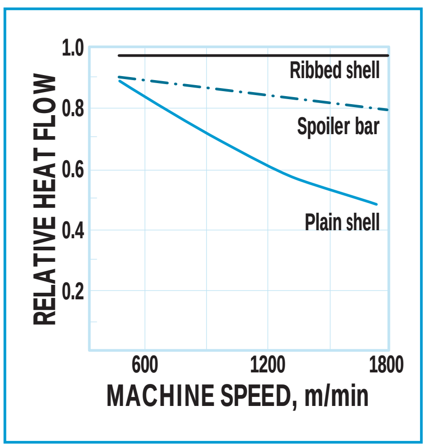
Other shell materials were considered to improve heat transfer, most notably steel and bronze. Though not perfect on all counts, gray cast iron—with its combined durability, stability, good heat transfer, and strong resistance to corrosion—presents a compromise of qualities difficult to improve upon.

The 1960s

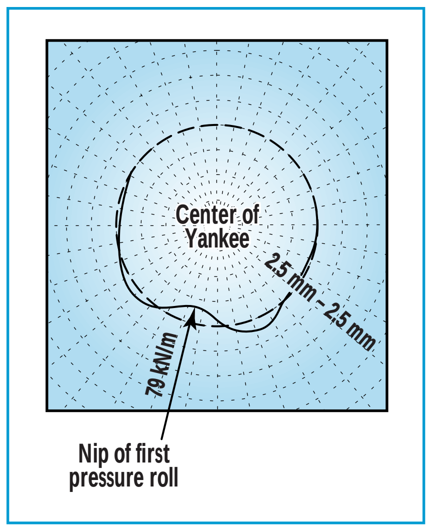
The 1960s saw revolutionary changes to Yankee design and technology. Diameters extended to 15, 16, and 16.5 ft, 5 m, and the ribbed shell, illustrated in **Fig. 19**, was developed as a means of increasing heat transfer while maintaining high radial stability. Ribs protrude through the condensate layer, reducing the insulating effect. Heat-transfer improvements with an optimized ribbed shell can be 35% better than a plain shell *(15)*, or even more when compared as a function of speed *(7)*, as seen in **Fig. 20**.



19. *Ribbed-shell Yankee dryer*



20. *Relative heat flow, Yankee comparison*



21. *Yankee nip deformation*

Stress analysis progressed, aided by the development of the digital computer, to produce the modern derate curve. In addition to pressure-induced hoop stress, the circumferential components from centrifugal, thermal gradient, and cyclical nip bending stresses were determined and included in a comprehensive safety factor. The fatigue nature of the vessel loadings was considered, and derate regimens were developed to preserve vessel integrity. Finite-difference techniques were used to calculate local vessel stresses and displacements, and considerable strain-gauge work was done to corroborate stress analysis.

A spin-off of the stress development was the ability to determine workable Yankee and pressure-roll crowns. The Yankee's response to a line load is not beam-like and varies around the shell, as shown in **Fig. 21**.

By the end of the decade, designers were contemplating Yankees that could operate at 1600 m/min.

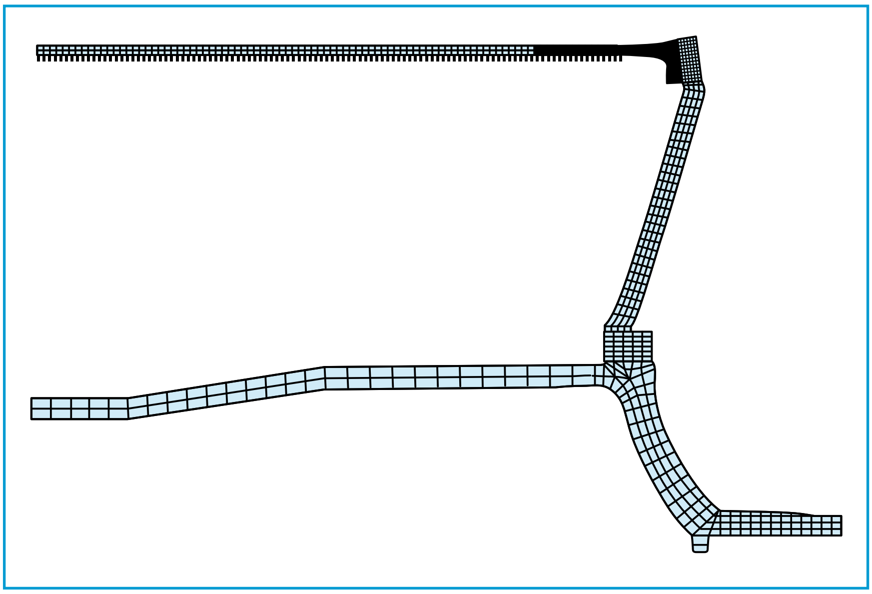
The 1970s

Finite-element analysis arrived in the 1970s and opened a completely new avenue for stress analysis. The graphics accompanying the analysis technique provide for quicker, more accurate analysis of stress and deflection. The formulation is inherently more flexible than finite-difference analysis. Thermal-analysis capabilities were greatly expanded.

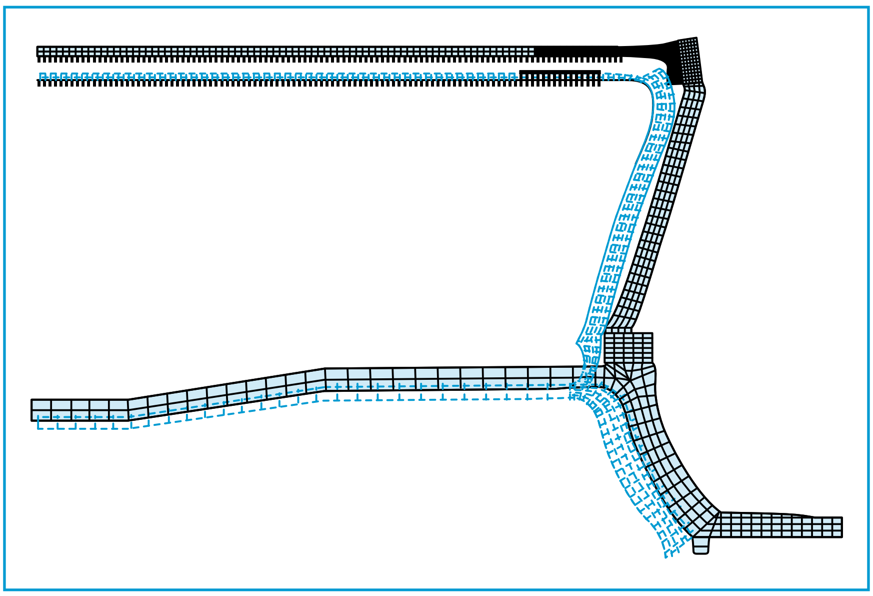
Finite-element analyses are now the heart of the calculations that determine stresses for ASME Code design approval and displacements for determining the compound crown for Yankees. First, the geometry is modeled by forming a mesh of nodes and elements, as seen in **Fig. 22**. Thermal boundary conditions are applied to the model, and temperatures are calculated throughout the vessel.

A structural evaluation is carried out from the temperature, pressure, and journal-support conditions. Finite-element analysis can determine primary stresses along with thermal gradient and discontinuity secondary stresses. **Figure 23** shows the deformed and undeformed shape of a Yankee during papermaking operation. Small pressure deformations and the more significant warm-up thermal strains cause the solid deformed model to grow from the dashed undeformed structure. The surface of the dryer under the sheet is shrunk radially relative to the heads because of being cooler.

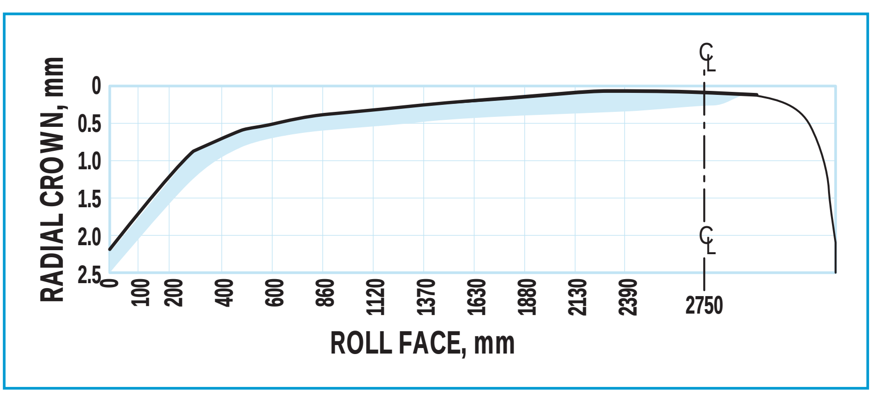
Applying compound-crown end relief to the Yankee shell helps prevent overloading of the pressure-roll covers and doctor blades on the ends, as seen in **Fig. 24**. The result is better runnability and smoother moisture profiles. At the end of the decade, ribbed dryers measuring 18 ft in diameter and 5.84 m wide were being produced to facilitate operation at speeds of 1800 m/min.



22. Yankee dryer finite-element model



23. Yankee dryer deformed matrix



24. Yankee dryer compound crown

Newport News left the paper equipment business in the mid-1970s after producing about 300 Yankees.

The 1980s

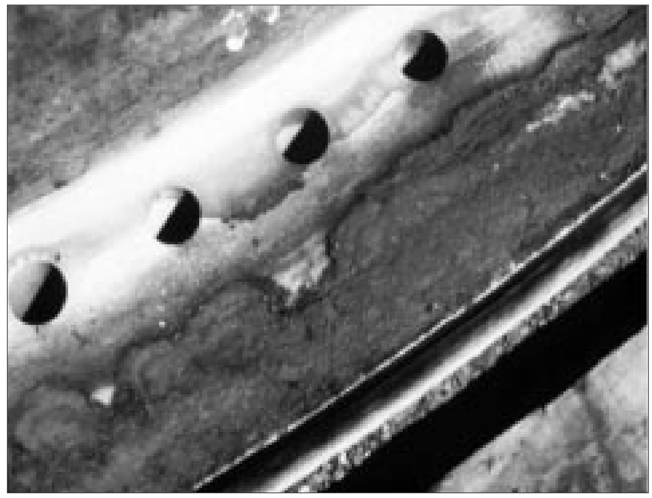
The 1980s saw 18-ft ribbed dryers with widths of 7.34 m and up to nearly 7.75 m wide.

A challenging corrosion problems was discovered in the mid-1980s. From time to time, surface corrosion had been evident on some Yankee shells, usually as a series of pits or grooves. During investigation of instances of serious surface runout problems, workers found several head flanges cracked nearly through from inside to outside, as seen in **Fig. 25**. The cracking was caused by the wedging of corrosion products. Corrosion had worked down between the head–shell joint flanges (**Fig. 26**), and the swelling of the corrosion products cracked the head flanges. Cracks have been discovered at the spigot interference and bolt pitch circle diameter, as seen in **Fig. 27**.

Fortunately, not all areas of Yankee dryers have been found equally subject to attack. The external area of the shell outside the sheet, including the head–shell joint, has been most at risk. Local high temperatures, process oversprays, tensile stress state, chemical accelerants, closed white-water systems, and, ironically, protective coatings on the shell appear to be risk factors.

Wedging stresses do not build up in a completely predictable manner, and the problem may appear to relent occasionally if corrosion products break out. However, the process has not been seen to stop once started, and it eventually proceeds to failure. Corrosion must be physically removed, and the resulting damage must be repaired before taking protective measures. An effective repair for the joints has been developed, and this can correct the problem if caught early *(16)*.

25. Crack in head spigot



26. Corrosion on head flange

Contact finite-element technology has been a valuable aid to understanding the behavior of the head–shell joint and the seriousness of the stresses resulting from corrosion. Contact finite elements provide a method of sensing whether element surfaces are in contact during an analysis and thereby able to transmit load. A model of the joint is shown in **Fig. 28**. Contact elements are distributed along the joint and fit surfaces. Bolt clamping loads are applied by truss elements. Assembly, pressure, and papermaking boundary conditions are also added, and numerical iteration proceeds until the contact element condition converges.

From this analysis, it was determined that during the papermaking condition—when the outside of the dryer is cool relative to the inside—the outside of the joint tends to gap, as seen in **Fig. 29**. This defeats sealing and protective coatings and encourages ingress of corrosion-supporting media. Using this information, it has been possible to redesign to increase the mechanical stability of the joint. However, corrosion protection also requires the companion strategies of coating protection and corrosion inhibition, which address the fundamental electrochemical nature of the problem. Enhanced mechanical design of the joint can give these strategies a chance to work.

DEVELOPING A SAFETY-ASSURANCE PROGRAM

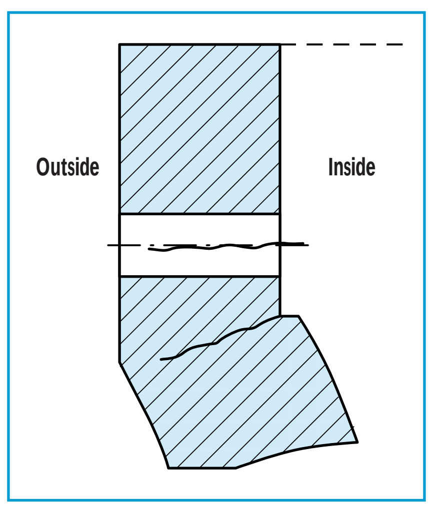
The history of Yankee development provides a perspective for designing a Yankee safety-assurance program. Here are some key suggestions for consideration:

- Review two TAPPI publications: "Guidelines for the Safe Operation of Yankee Dryers" *(17)* and "Yankee Dryer: Guidelines for Safety and Condition Assessment" *(18)*.
- Maintain critical records on the vessel. Examples of critical records include assembly drawing, manufacturer's data report, derate curve or chart, and records of grinds and repairs.
- Derate curves should be kept up to date. Current shell thickness should be available. Steam pressure, pressure-roll loading, and condensing load (where specified) should be maintained in accord with the derate data. Safety valves limiting steam pressure and pressure-roll loading should be properly set.
- Benchmark inspections should be made on a periodic basis. The recommendations of manufacturers, insurers, and TAPPI guidelines should be considered.

One of the simplest and most cost-effective inspections is a thorough visual external and internal examination to check for loose, broken, or missing parts; metal loss; distortion; leakage; and corrosion. Inspections for corrosion/erosion metal loss as well as corrosion-product swelling at joints are highly recommended. Vessel age, design, and condition may suggest other inspections with various nondestructive test methods.

SUMMARY

Concepts and techniques of stress analysis have guided the development of the Yankee dryer from a primitive drying cylinder to a modern pressure vessel of extraordinary productivity. From the most rudimentary equations governing stress and deflection to today's modern digital computing with nonlinear finite-element analysis, structural designs and operating strategies have evolved to optimize the Yankee dryer's safety, reliability, and productivity.



27. Crack locations on head flange

KEYWORDS
Development, drum driers, machine design, stress analysis, structural analysis, tissue machines, yankee driers.

A review of Yankee dryer history highlights the importance of stress analysis in the evolution of this remarkable piece of equipment. It also demonstrates that stress analysis can be part of an adaptive problem-solving/design-optimization strategy that can extend the limits of Yankee productivity in the future. TJ

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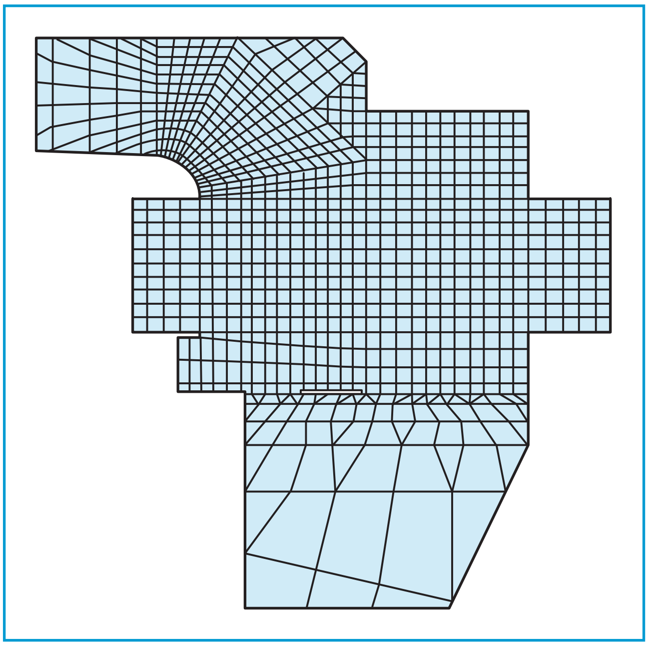
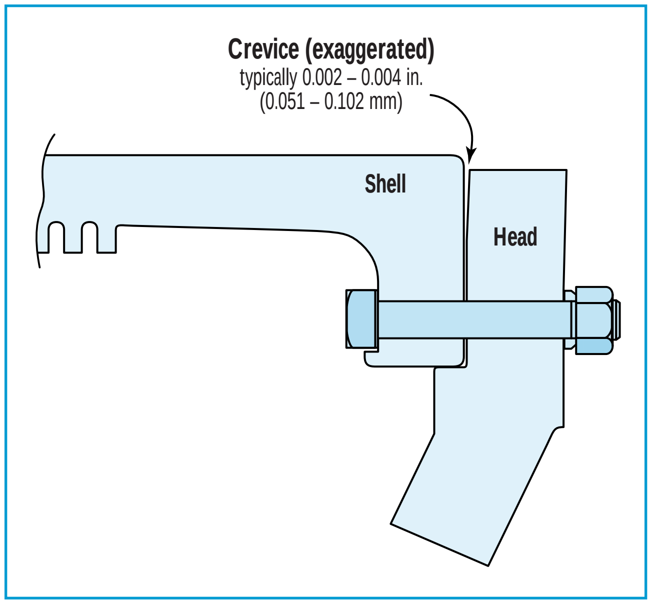


28. Finite-element model to analyze head-to-shell joint behavior



29. Gap in head-to-shell joint

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