About the Author

D. Carl Yackel is a semi-retired Consulting Engineer. He is a 1950 ChE graduate of Iowa State University and a TAPPI member since 1960. He has lived in Lake Oswego, Oregon since 1974.

Mr. Yackel's 40 years experience in mixing and agitation began at Mixing Equipment Company in 1950 and has included such companies as Beloit-Jones, Prochem Ltd. and Voith-Morden, Inc. where he resigned as Vice President of Engineering in 1979 to enter the consulting field with the engineering firm of Jeddeloh, Hays, Inc. Retired since the Fall of 1988, he has maintained contact with this field and has been sought after for many agitation problems.

He is a member of the Process Engineering Committee, through which he has given papers on agitation and under machine pulping at TAPPI engineering conferences. He is a registered professional engineer in Oregon and New York State.

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ISBN 0-89852-503-9
TP R-177

Pulp and Paper Agitation:
The History, Mechanics, and Process

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TAPPI PRESS • Atlanta, GA
1990

Preface

It has been 40 years since a young chemical engineer, fresh from years of flying Army Air Corps machinery in World War II and three years of finishing college, slid wonderingly into a world of mixing technology and agitation. Someone once said, "Few engineers end up in their original discipline." This was never more correct than in my case. My dreams of running a chemical plant or being a research chemist (my first love) vanished in the study of mixing miscible fluids, dissolving solids and suspending solids to extreme concentrations. The first company I associated with was young, eager, and soon began to investigate the agitation of pulp slurries.

The centuries-old paper industry was still reveling in the "new science" of midfeathers and vertical circulators. Mixing technology changed all that, but, like the paper industry, changes came slowly due to the lack of process data and the secretiveness of suppliers. The industry was slow to accept what the "old guard" regarded as blasphemy. But time healed the wounds and "mixing" was renamed "agitation" as a salve to the "seniors." When was the last time you *purposely* designed a straight-shell high-density chest using mining nozzles and a toy agitator to dilute high-density stock?

In this book, we will trace this history and attempt to reveal the "secrets" of agitator applications for the paper industry of the present. Other advancements will some day make us the "old guard," but for now—here is how it is!

D. Carl Yackel
May 1990

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Chapter 1:

The Birth of the Modern Agitator

Sometime back in the 19th century, a rather complacent, non-visionary soul in the U.S. Patent Office thought it was time to close up shop because there was really nothing else that could be invented. He was the kind of person satisfied with the "green cheese" theory of the moon and felt that no human being could possibly withstand the forces involved if able to travel at greater than 20 mph. For too many years, the same kind of regressive thinking has affected our industry, at least up to the last four decades. Even now, the most generally accepted method of making paper consists of spreading a thin slurry of fibers on a moving wire mesh, pressing it between felt-covered rollers, drying it over steam-heated drums and winding it on a spool. Allowing for differences in size, speed and geometry, this is not conceptually different from the Fourdrinier Brothers' wonderful machine of 1801! (Fig. 1-1.) But more about that later.

The agitator is that sometimes forgotten device that keeps a pulp slurry in motion and in various degrees of uniformity, depending upon application. It has gone through an even greater evolution than our paper machines, albeit at a snail's pace when compared with such widely accepted advances as the "Wright Brothers' Folly" or Henry Ford's flivver (Fig. 1-2). Less than 70 years occurred between a flight the length of a football field and man's first visit to the moon.

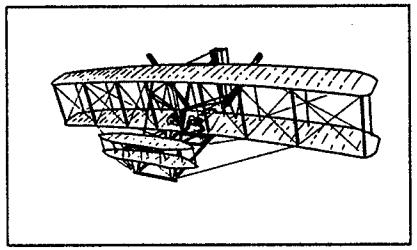


Figure 1-2. Wright Brothers airplane.

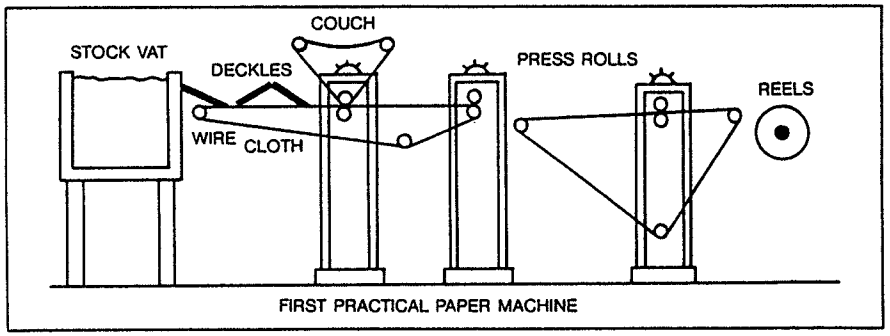


Figure 1-1. Early paper machine.

Early papermakers soon realized the cellulose fibers they had so carefully extracted from rags, plants and logs would not stay uniformly suspended when in contact with water. In fact, the fibers found the most troublesome places to collect and form dense plugs of immovable mass, such as square corners in any container and, of course, any available pump suction area.

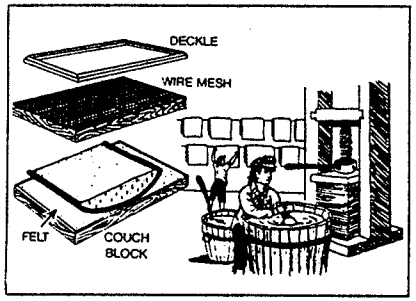


Figure 1-3. Wooden vat with hand paddle.

These propensities gave birth to a plethora of agitator concepts, some so primitive as to be laughable; others still laughable but unfortunately remaining with us. The first agitator design of the 15th century must have consisted of a wooden vat, a Dickens-type street "urchin" and a hand paddle (Fig. 1-3). The urchin paddled resolutely while sundry "papermakers" dipped their sheet molds, hand couched a wet sheet, pressed them for further water removal and then draped these over hair ropes, five sheets to a row, to dry. With the ongoing advancement of water wheels, the steam engine, and finally the electric motor to drive machinery, other methods of agitation evolved.

Let us consider some of these concepts and early attempts to produce and maintain uniformity in pulp slurries.

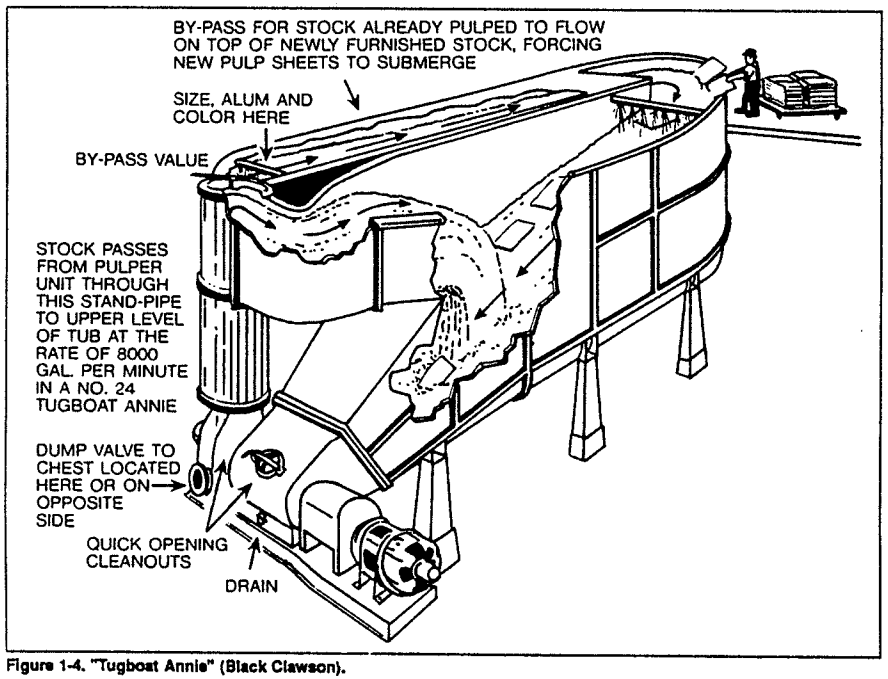


Figure 1-4. "Tugboat Annie" (Black Clawson).

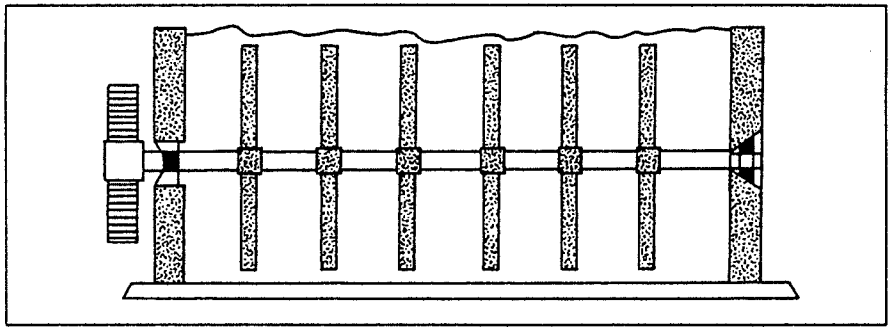


Figure 1-5. Horizontal paddle agitator.

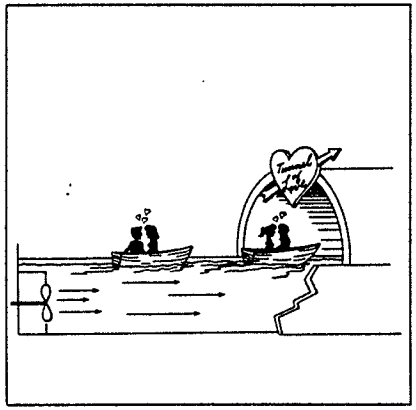


Figure 1-6. "Tunnel of Love."

Pump Recirculation

A pump was used to transfer "stuff" from one container to another. As long as the slurry was moving, the fibers stayed suspended. Papermakers decided to use the pump to take the slurry from the vat and dump it back in, thus maintaining that motion that seemed to keep the fibers from settling. With this concept, recirculation was born. No manufacturer is more closely identified with this type of agitation than the Black Clawson Co. of Middletown, Ohio, immortalized in what became affectionately known as "The Tugboat Annie." (Fig. 1-4).

First recirculation attempts quickly revealed that *only* where there was velocity would the fibers continue to move. When attempting to recirculate in long, rectangular chests, fillets of stock collected in the corners and along the boundaries of the flow stream back to the pulp suction causing a build up of rotting stock. Designers began building in those fillets with less costly materials than the hard-won cellulose fibers. Shaped wooden fillets, metal, concrete, and finally smooth file facing, was used to allow the stock slurry to stay in motion. The design of fillets became almost a "cult," with manufacturers clinging to their own set of corner ratios, bottom curvatures and pump suction feed channels.

Meanwhile, other ideas were being considered to lessen the cost of pump horsepower being diverted to recirculation.

Horizontal Paddle Agitators

Early experience with pump recirculation led some designers to seek a way of imparting motion to the *whole* chest rather than a narrow flow stream going back to the pump suction. Someone, whose name is lost in history, thought of the urchin with the hand paddle and conceived of using a number of large diameter two-bladed paddles on a horizontal shaft, rotating through the stock slurry at a very slow speed, reducing the opportunity and space available for the fibers to accumulate and settle (Fig. 1-5). The energy requirement of only 2 or 3 rpm was very low and, with a semi-circular bottom, greater amounts of

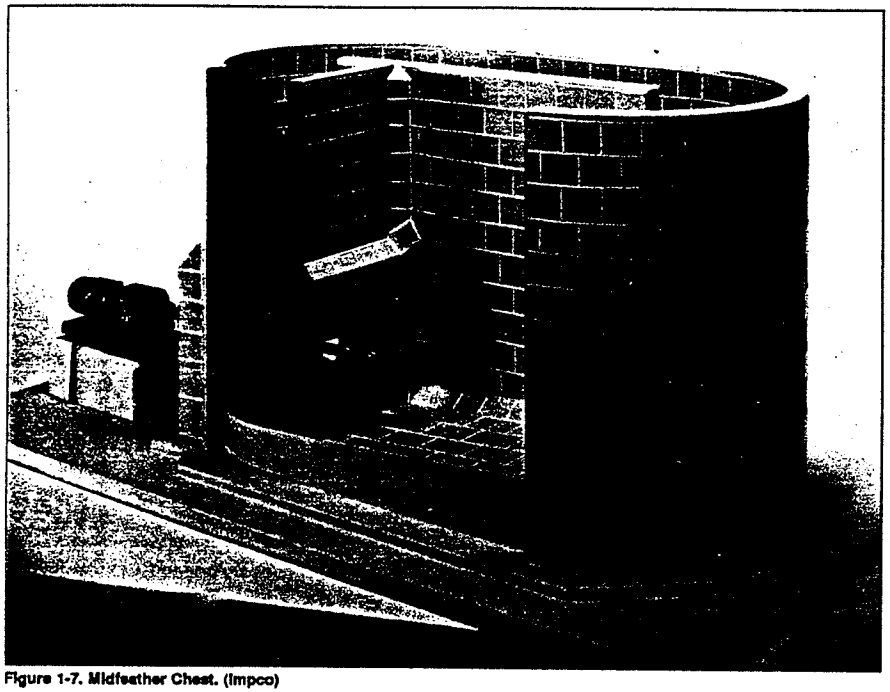


Figure 1-7. Midfeather Chest. (Impco)

stock could be held in storage. There were other problems associated with this design that became apparent, but we will discuss those in a later chapter.

Propeller Midfeather

Many were still convinced some type of continuous recirculation was the key to the least expensive and most efficient method of maintaining a stock slurry in suspension. Perhaps some less structured designer had a vision while watching the boats in the "tunnel of love" at an amusement park. (See Fig. 1-6.) "Why not keep that recirculation path entirely within the chest and use a flow generator that is more efficient than a standard centrifugal pump?" The midfeather stock chest was born. This revolutionary design by such men as A. M. Hurter of the Stadler-Hurter Co. and "The Commander" Arthur Whitesides of Improved Machinery Co., consisted of a long rectangular chest, well filleted at the ends and along the bottom sides. A vertical wall down the center of the chest ended some distance from each end. At one end of this center wall, a cross wall was constructed and a three- or four-bladed propeller was installed at close clearance to the circular hole in the cross wall. The propeller is essentially a low head pump with a wide open suction and, when rotated at some predetermined speed, will "push" or "pull" (depending on rotation) the stock slurry around the "racetrack" created by the midfeather wall (Fig. 1-7). With the success of this simple device, additional channels were later added to increase the volume of this low head storage chest. Soon every paper mill basement was partitioned off in some fashion to provide large storage chests for satisfying the increasing hunger of larger and faster paper machines. Fillet design became even more important to maintain suspension and continuous flow.

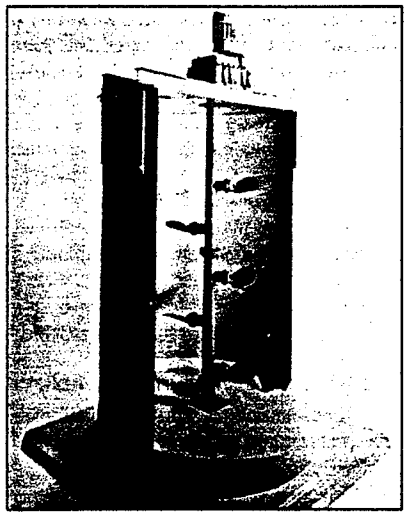


Figure 1-8. Vertical Circulator. (Impco)

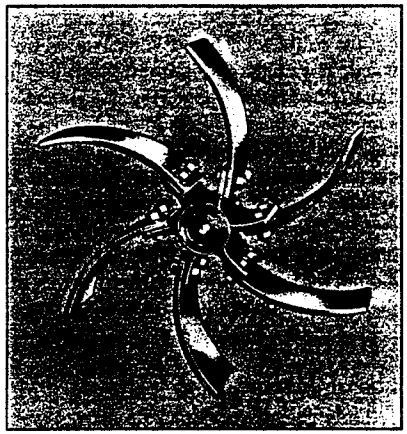


Figure 1-9. Spiral Backswept Turbine. (Lightnin)

Vertical Circulators

As the capacity of paper machines continued to increase, it became apparent the storage capacity provided by midfeather chests required excessive amounts of real estate. Basements became confusing alleyways as more and larger vacuum pumps and other auxiliary equipment clogged the increasingly narrow passageways under the machines.

Some designers recognized the economies provided by tall vertical chests which required much less area and could extend through two or three floors of the mill or be installed outside the mill. This could free the machine room for other equipment, but how could this design avoid the old problem of dead and rotting stock? Since the multiple paddle had been successful in horizontal chests and the propeller had proved itself in the midfeather; why not try a vertical chest with multiple propeller blades? Only Darwin could appreciate the evolutionary advance, although the modern agitator requires many more "missing links" than Darwin's jump from ape to man.

The vertical circulator included a long, heavy vertical shaft mounted centrally in a vertical cylindrical chest. Many arrangements of propeller blades were used, but a typical system featured a three-bladed propeller mounted just above a bottom steady bearing, followed by several single blades which gradually spiraled up the full length of the shaft to just below the normal stock level. This style of agitator, irreverently referred to as the "Christmas tree," has survived to this day, even outliving the midfeather designs that had become so popular in the first half of this century (Fig. 1-8.) Its capacity to keep very large volumes of stock in motion, if sized correctly, was unquestioned. Its ability to produce uniformity is quite another story which we will look at later.

Vertical Shaft *Turbine/Propeller* Agitator

In the early 1950s, Mixing Equipment Company, Inc., a manufacturer of mixing equipment for the process industries, became interested in pulp agitation. Under the able direction of Dr. James Y. Oldshue, the company refused to be hampered by a long history of "you can't improve on grandfather's design" and began

an intensive study of fluid mechanics. The result was the first installation of a large vertical agitator using a single spiral back-swept turbine near the bottom of the stock chest (Fig. 1-9). This design produced a continuous suspension of fibers and continuous uniformity to the pump suction for immediate use as uniform furnish to the process. Because of the high torque requirements of the radial flow turbine, this was quickly followed by a single-propeller design with even better results at low horsepower (Fig. 1-10). Many such units were developed for large and small chests, spanning the application spectrum from

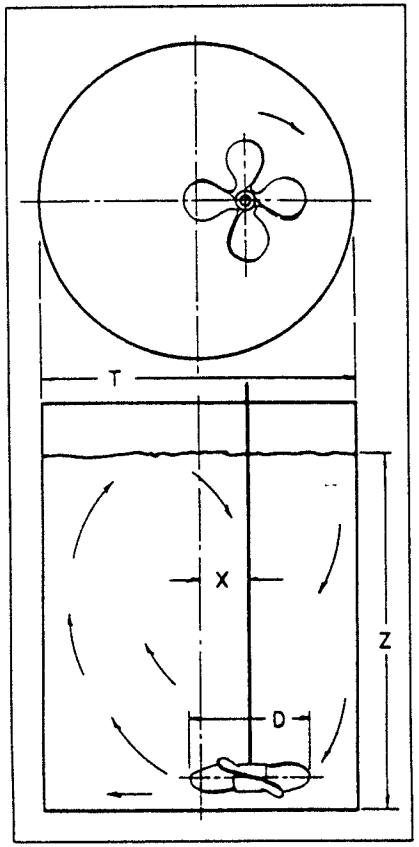


Figure 1-10. Vertical-single Propeller. (Lightnin)

outside storage chests to blend and machine chests, even in flash tanks. But there were still stumbling blocks to overcome—vertical units with gear boxes were expensive; bottom steady bearings were a maintenance man's nightmare. "Where can I put 100 tons of stock while I replace a steady bearing bushing?" Some applications developed disastrous shaft vibrations at certain levels that were not completely understood. Though these problems were eventually solved or at least assuaged, something more efficient must be available.

The Horizontal Shaft Propeller Agitator

With mounting pressure from field offices, researchers at Mixco discovered basic relationships that would allow a single horizontal shaft-mounted propeller to produce a continuous suspension of paper pulp in *any* shape stock chest. The empirical gibberish of midfeather walls and all but the simplest of corner fillets was eliminated. The unitized side-insert propeller agitator had come of age (Fig. 1-11), and soon other manufacturers joined the parade into the 20th century. As paper machines grew bigger and faster, old paddle agitated couch pits were causing machine tenders to grow old before their time. The initial breakthrough of the vertical agitator did not quite fit the papermaker's dream at the wet end; sheet slitters were more practical at the rewind stand! (Fig. 1-12.)

Age of Understanding

Early designers of stock chests and agitators were not oblivious to the shortfalls of their designs. They did the best they could with the resources available to them. Think of the scenario of a Bronze Age metalsmith trying to construct a turbine wheel for a modern jet engine. Methods of measurement were hardly refined. Fluid mechanics were just beginning to be understood. Inadequacy of the agitation was still good enough for the slow-speed, low-production paper machines and the quality of paper was acceptable to end users

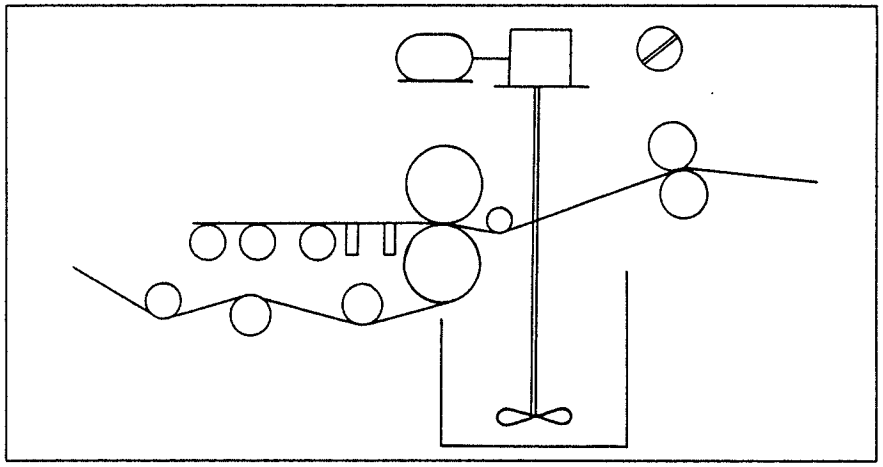


Figure 1-12. Vertical agitator through couch.

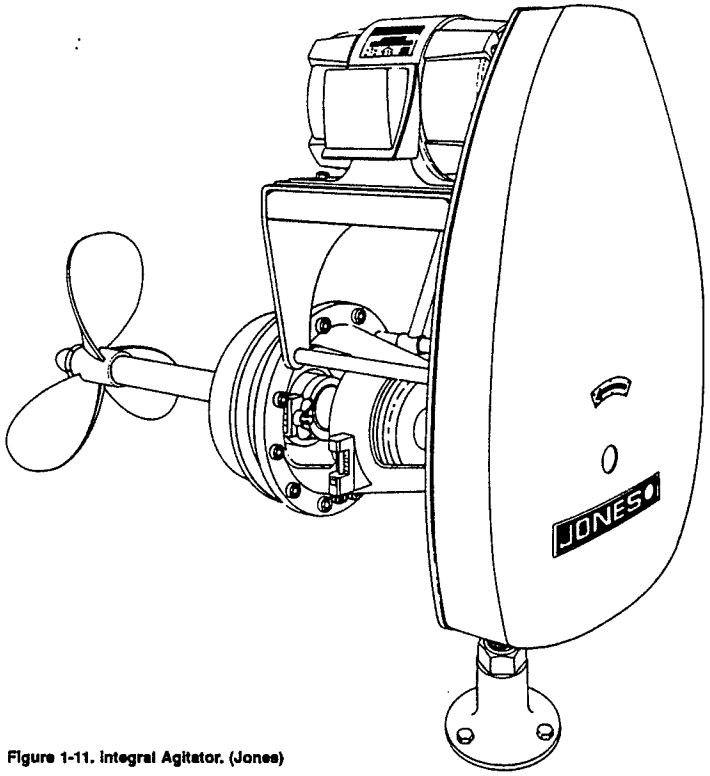


Figure 1-11. Integral Agitator. (Jones)

whose equipment advances paralleled the paper industry. "When you only have iron, you only make iron plows, not carbide-tipped blades!"

So the lack of process data, or insufficient grasp of what these data might mean, hampered the understanding of agitator design. In the earliest days, consistency as a specific value meant very little. One only knew that "it should be thick here and thin there." If an experimenter in 1583 had an oven in which to dry a stock sample, he certainly lacked a proper balance to prove a consistency of 3.561%. But after the invention of the paper machine and the need for increasingly larger stock chests, what did our best designers from the old line paper machine manufacturers measure, even as late as the middle 1950s? Well, they measured consistency and accurately too, although the concept of "air dry" always being 10% less than "bone dry" somewhat confounds this aging chemical engineer. (Remember, in 1950, one had to look long and hard to find a Chem E in a paper mill.) Designers recognized some differences in flow properties between bleached stock and brownstock. One "good old boy" in a southern kraft mill once told me, "You ain't never tried that thing in southern kraft, boy; it just don't act the same as that yankee kraft you bin foolin' with." The main criterion in picking an agitator size was velocity of surface motion and horsepower per ton (or cubic ft.) of stock in the chest. In a midfeather-style chest, if you could measure 25-to-30 ft/min. of surface motion, you had it "made in spades." The stock might be as stagnant as a cesspool two feet below the surface, but no one knew it until black smelly stock began to show up someplace else. One could look at the surface of a vertical chest fitted with a "Christmas tree" agitator, squirt a stream of tobacco juice into it to improve the Mullin, and state that the surface motion looked "real good."

The countdown for combining machine performance with process requirements was underway. Meanwhile, back at Beloit, Black Clawson, Rice Barton, and all the machine builders, the new breed of fourdrinier machines was getting bigger, faster and more demanding of quality at the headbox. That oft-forgotten "fitting" on the side or top of the machine chest (or you-name-it chest) the "agrivator," was becoming increasingly important. The stock prep supervisor wanted 100 tons of good pulp in storage, not 50 tons of black goop and 50 tons of tainted furnish. He wanted 5% pumped to his refiners, not 3½ to 6½ in five-minute swings. Tour bosses began to wonder why one stock chest took 30 minutes to level out a pH adjustment or color additive, while another was ready in three minutes.

"By your deeds you shall be known" and so it was, as agitator design improved, so did process results, from the blow tank to the paper machine and on to the discharge from the broke chest. But why the horizontal agitator? I'm not ready to close the Patent Office yet, but at this time of writing, the horizontal shaft agitator is the best thing we've got going. Tomorrow may be another story. At one time, we thought we had the mixing world by the tail and along came some guy with a "wiggly worm" that didn't move! The static mixer was born, so just settle back and listen to how it is, for now!

The propeller is an axial flow impeller. It draws in fluid on one side and discharges it on the other side. It acts in a similar manner to a jet stream—after discharging, entrainment occurs though decreasing velocity, flow is increased, resulting in a high-volume turnover. The vertical shaft propeller agitator, though superior in torque requirement to the vertical shaft turbine, is still an axial flow impeller. Because of its location close to the bottom of the chest, its axial component has to be immediately changed to a radial component to sweep the bottom and entrain flow from the stagnant volume above it. This reduces the efficiency of the configuration. The horizontal shaft propeller, on the other hand, gives the discharge stream the entire chest diameter in which to expand and entrain flow. The upward helical flow pattern generated envelopes the entire chest.

It's always been inevitable that, "We've done it in this chest, but now the boss wants to double the size." Scale-up is not confined to the paper industry. Suppose my wife makes small teddy bears as a gift for a baby shower, and now she wants to make a large one for her garden club. "How big should the eyes be?" she thinks. Geometry from 10th grade shows its worth—geometric similarity! A geometric series is one in which all ratios are maintained equal. If it is a turbine-type impeller, the height and length of the blades in relation to the diameter are maintained. In a propeller, the same rules must prevail. The developed area ratio (DAR), blade width, hub diameter to blade swing and other constants must remain in the same ratio. A square pitch marine-form propeller is a good example. You wouldn't expect similar results from a larger propeller on a larger boat if, aside from the larger diameter required, you also changed its pitch. Geometrically similar is the key phrase in scaleup, or as we say in the "trade", "a homologous series." That's a euphemism for "make it like the other one, only bigger!" Geometric similarity makes it possible to predict power response and process results.

Design of the Unit

Though we will cover specific design procedures later, now let's discuss basic unit design.

Early experience with the never-before-encountered shaft vibrations of vertical agitators revealed the phenomenon of fluid force. Because of the hydraulic inequities associated with an axial flow impeller, there was a force exerted upon the impeller in an adverse fashion. This force yielded to examination and was, included in the calculation relating to shaft design and bearing life. Prior to this, the only shaft considerations we observed were critical speed and torque. Today, no manufacturer familiar with all of the forces designs a side-insert unit at greater than 5000-psi calculated combined stress, nor less than 100,000 hours B10 life for shaft bearings. The packing box or shaft seal has always concerned maintenance people and the manufacturer. A packed seal operates successfully by the fluid film between the packing and the shaft sleeve. (A more erudite explanation I will leave to packing manufacturers). The packing box on a horizontal agitator must be able to supply this fluid film before fiber slurry can sneak into the box. Therefore, most legitimate agitator suppliers provide a packing box in which the inboard section contains a close fitting throttle bushing which feeds lubrication at no more than 5 gal/h back through a four- or five-ring box and allows a drip leakage of clear water equal to a fraction of that flow. Sealant flow should be controlled in volume and in pressure no more than 10-15 psi above the chest head.

Materials of construction for the agitator wetted parts are usually dictated by the client. In the paper industry, we don't usually deal with many exotic chemical mixtures, and if it wasn't for the water we use in alarming quantities, carbon steel would be satisfactory for most of what we do. However, rusty toilet paper might upset some of our customers, so usually stainless steel, T304 or T316, is acceptable and meets 99% of our needs. In a bleach plant or in other chemical areas, more exotic metals may be required. The paper mill is more aware of their needs than the equipment supplier.

Chapter 2:

Uniformity—The Key to Success

In the chemical industry, when we begin to make product "C" which we intend to sell at a profit, we may start with chemicals "A" and "B," put them into close contact in some sort of vessel while they combine, and we have created material "C." If we have studied all of the costs, material "C" can be sold for a higher price than the cost of "A" plus "B" plus all the direct and indirect processing costs. Perhaps that is an over-simplified version of the ingenuity of our coworkers in the process industries, but no one makes more work out of a single raw material than the paper industry. Some of us use some pretty exotic formulations to coat our product and some add a wet-strength chemical, which leads to more "hair-pulling" by those of us building pulpers, but let's discuss the basic, mainstream process of making a sheet of paper from a tree.

Maybe you've never thought of it this way. We take logs cut from trees in our forests, saw them into suitable lengths and then grind them into rough fibers against a rotating stone or slice them into chips and cook them in a huge pressure cooker. Next, we add a large quantitity of water to make a pumpable slurry, and then we get rid of the water by a vacuum device, sometimes called a decker or, at this stage, perhaps a brownstock washer. The "blanket" of fiber coming off this device (Fig. 2-1) is then—you guessed it—diluted again with water, to get back to the same pumpable slurry we had before, only now it's "clean." This initial treatment isn't 100%

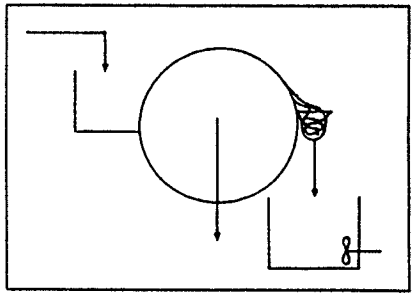


Figure 2-1. Decker with sheet to repulper.

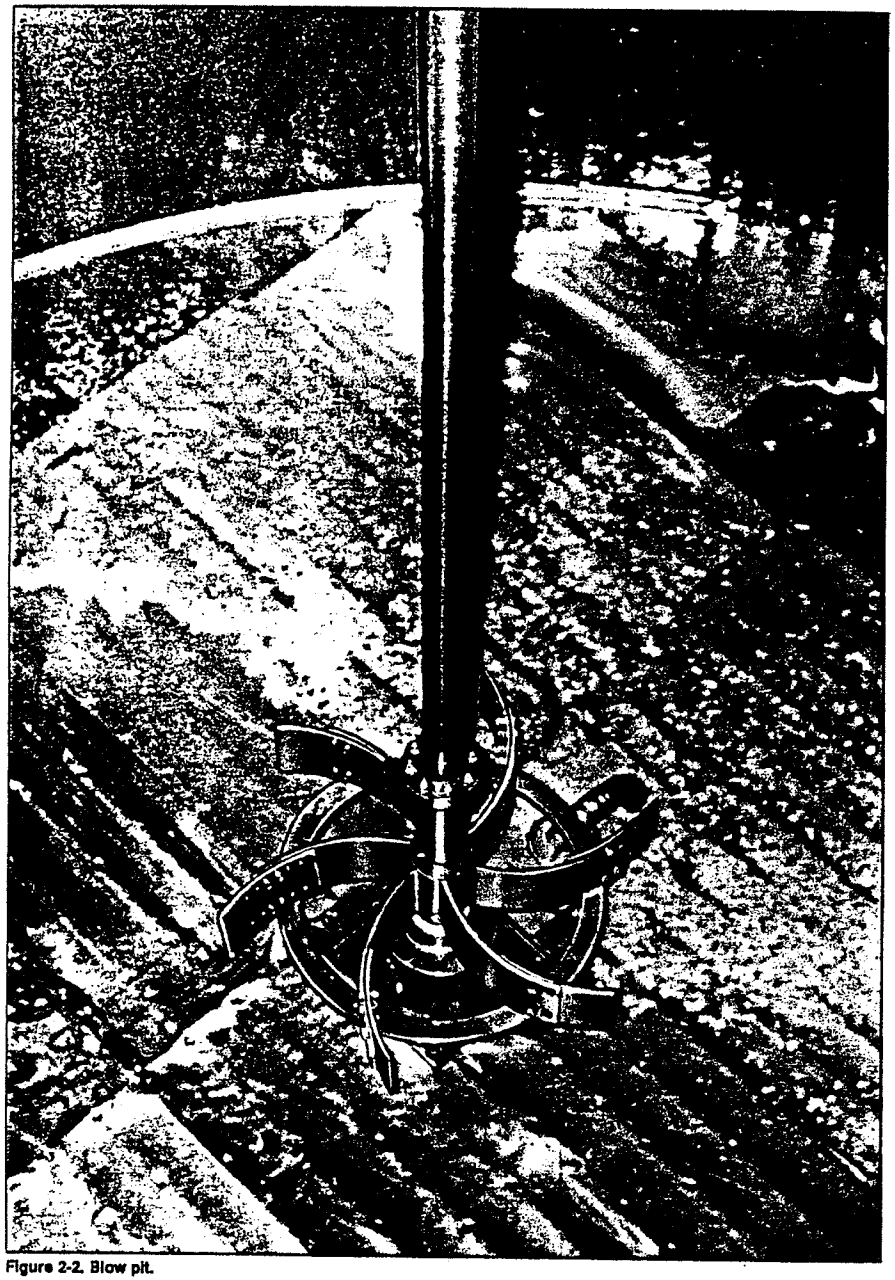


Figure 2-2. Blow pit.

effective—there are still oversized materials present and so we "strain" the flow through screens after more dilution and do the "water removal bit" all over again. Unless you are new in this wonderful industry, you know we keep on with this "concentration and dilution" until we get out of the pulp mill and into the paper mill. Here we have similar pieces of equipment, now called savealls, and fine screens and high-consistency refiners that require this tedium. Chemical "A," our primary material, has never encountered Chemical "B" but "Oh, what we've been through!"

The manufacture of paper pulp consists of an interminable series of dilution, concentration and re-dilution. Let's look at some of the agitator applications involved in the "torture treatment."

The Digester

There are few mills today which use the batch digester as the primary source of raw furnish. Time was, though, when the warning horn would send everyone scrambling from the digester blow area as one or more of these "cookers" was ready to "blow." In a sulfite mill, we had wooden blow pits with drainer bottoms (Fig. 2-2). In a kraft mill, we had blow tanks. In both of these, the function was the same: The cook would open the blow valve and, under digester pressure and high temperature, the cooked chips would discharge at high velocity against a "target plate" on the wall of the pit or tank. This served two purposes: the high velocity impact helped to break up knots and clumps of stock; the hardened metal plate protected the walls of the vessel.

This was especially true for older sulfite digesters—their brick linings often came out with the stock, and the wood-stave blow pits were ill-equipped to handle that kind of bombardment. (That's why I grabbed my hard hat and ran whenever I heard that blow horn go off!)

Once in the blow pit, after the hot liquor was drained off, a series of washings took place with water. In the early days, a rake-type agitator attempted to blend the high-consistency pulp with wash water. In more recent years, this has been replaced with a vertical turbine agitator, giving a controlled washing resulting in nearly uniform stock.

The batch blow tank following a sulfate (kraft) digester generally was (and still is in many mills) a vertical, domed-top pressure vessel with a steep conical bottom (Fig. 2-3). Black liquor from the first-stage washers was introduced into the bot-

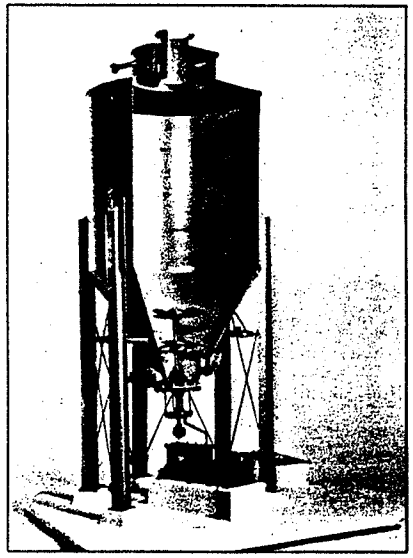


Figure 2-3. Blow Tank. (Impco)

tom cone, and a bottom-entering vertical agitator with multiple two-bladed flat paddles, of increasing diameter, operating at very low speed (20 rpm is typical) attempted to reduce the high-consistency blow to about 3½%. Since control of the black liquor dilution valve was signaled from the agitator drive motor power response, consistency was often erratic over quite a wide range. (We will see in an en-

suing chapter that power response varies slowly over a wide range of consistency.) As the parameters controlling the agitation of pulp slurries became more common knowledge in the late 50s and early 60s, it became apparent that this style of blow tank agitator with its "motor load controlled" dilution was more closely related to a giant viscometer than an agitator. One major supplier, Improved Paper Machinery Co. (IMPCO), marketed a vertical bottom-entry propeller unit consisting of two different diameter propellers with opposed pitch settings (Fig. 2-4). This unit,

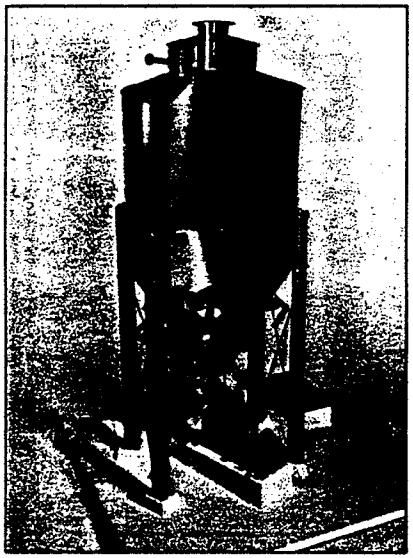


Figure 2-4. Blow Tank Agitator. (Impco)

though an improvement over the slow-speed paddles, still only controlled consistency over a wide range because of dilution controlled by motor power response. In later years, a reduced bottom tower configuration was used with a side-insert agitator designed for accurate consistency control in a controlled dilution zone with a consistency regulator controlling dilution.

In all of these various configurations, the purpose was to control consistency so a uniform stock slurry could be fed to the next step in the process. In the case of the blow pits and blow tanks, the next step was further dilution for screening and washing. The use of properly designed agitation equipment allowed more complete washing of stock in the sulfite blow pits and, in either case, provided accurate control of dilution for the stock screens and washers.

Other Critical Processing Steps

As the newly separated cellulose fibers, now washed and freed of knots and uncooked chips, begin a treatment cycle culminating at the headbox of the paper machine, accurate control of consistency becomes paramount to the proper treatment of the pulp. Although there are few batch digesters being installed today compared to the almost universal use of sophisticated continuous units, the remaining elements of stock preparation remain the same. Entering the paper mill from the pulp mill, the pulp still must be stored in feed chests, sometimes at high consistency to conserve space. The fiber's characteristics must be adjusted for the sheet being made, which means refining in old conical jordans or modern disk refiners. After diluting again, blending with other furnishes, adding dyes or pH adjustment chemicals in a blend chest, we adjust consistency once more for feeding to a machine chest. From here, we make further major dilutions before banks of centrifugal cleaners and then screen the furnish through pressure screens before dilution at the basic weight valve prior to the headbox.

Every one of these steps involves a piece of equipment designed to process a particular volume of slurry at a specific consistency. Variations from the design consistency affects the efficiency of that piece of equipment, often to catastrophic results. If the consistency is too low, the

production rate (*tons/day*) decreases. If we are feeding a washer or a saveall, we have increased the chance of overflowing the vat as well as decreasing the consistency of the pulp coming off the drum or disc. If the consistency is too high, we may plug the vat, possibly resulting in structural damage. If this were a pressure screen or a centrifugal cleaner, a higher consistency would plug the unit and shut down that part of the stock flow. Refiners, especially fine-tuned disk refiners, accomplish a specific treatment of the fibers for a particular grade. These may be rated at different tonnage rates, inlet consistencies and pressure drops or pressure rises, depending on the disk pattern installed for a particular grade. Too thin a concentration may allow the plates to break through and burn; too thick or insufficient fiber treatment can result (leading to off-spec sheet on the paper machine) in frequent breaks or actual plugging and damage to the refiner.

Consistency Control

The only solution to consistency variations and poor results problem is accurate control. We need consistency regulators with "space age" accuracy in measurement. Perhaps "consistency regulator" or "consistency controller" is a misnomer. It is only a measuring device. It reads the resistance flowing through it which is a value of consistency and then, with an established set point, it signals a dilution valve to open or close. If the variation in the stock stream is greater than the range of that valve, we've got a problem. The "consistency controller" can't add fiber to the stream, it can only add or subtract water at the controlled dilution point. (An exception might be a bale pulper in which the control element can start or speed up the conveyor, but this addition of fiber is several minutes away from the measuring point). But let's assume that the consistency variation being read by the controller is within the range of the dilution valve say, full open to closed. Without help from some other check point, we are now in the classic "hunting" mode with the controller struggling to find a midpoint and only succeeding in sending a "sine wave" of consistency variations on to the cleaner, screen, refiner, blend chest or whatever is downstream of this point.

That "other check point" is a properly agitated chest upstream of the measurement. A midfeather chest agitator, "Christmas tree" agitator, or horizontal shaft paddle chest can't level out variations on a continuous basis. Only a chest equipped with a properly sized propeller or turbine, impeller providing random top to bottom motion can. It can limit variations in consistency to ± 0.25% and deliver stock to the pump suction with instantaneous variations as low as ± 0.1% from average consistency. This doesn't mean that because of conditions upstream of this chest, the consistency or any other variable, might not increase or decrease over a period of time. It does mean the random agitated chest *will* blend those variations to minimum instantaneous variations so the changes are gradual over a long time period, allowing the "controller" to smoothly adjust the dilution valve without frantic hunting over its entire range.

The midfeather agitated chest and other "circulators" mentioned can only, if sized correctly, keep stock in suspension. Upsets to these agitation configurations move through these chests in virtual plug flow without perceptible change. When one considers that the whole purpose of stock preparation is to deliver a properly treated fiber at the exact consistency required to the headbox of the paper machine, we must maintain uniformity of the total furnish throughout the preparation process. Each piece of processing equipment, interacting with a fiber slurry not of the specifications for which it was designed, contributes some poorer quality to the furnish. This, more often than not, produces off-spec paper at the reel, or worse yet,

more broke in the broke chest. It was once said, "There are two methods to lose money in the paper industry. One is to make broke 24 hours a day, and the other is to not run the paper machine at all. Given the choice, the latter is by far the most economical."

Chapter 3:

Uniformity—A Hard Goal to Attain

As we have learned from Chapter 1, the design of an agitator that best suited our goals was a long time in coming because early designers were striving toward the wrong goal. By maintaining suspension, and dealing with dewatered stock, designers thought this would automatically solve all other glitches in the process. This was a fantasy that plagued the industry from the development of the first paper machine in 1801 all the way through the early 1950s. *(21)*

Let's look at these two eras of our industry's history in more detail:

The first era, lasting into the early 1950s, produced tremendous growth in the pulping processes and in the design of larger and faster paper machines, though there was limited improvement in the handling of stock slurries.

The second, just now 40-years-old, has represented the biggest single change in agitator design-thinking, as well as fantastic changes in paper machine design and other stock prep equipment. One can only wonder what a paper mill will look like 20 or even 10 years from now. If you have a problem trying to visualize some of these changes that have occurred in your lifetime, think for just a moment of a 1500 T/D linerboard machine and wonder how you would:

a. Refine at all positions with conical refiners;
b. Handle a full machine break with an off-machine broke beater;
c. Do all screening, from the pulp mill to the fan pump, with flat screens or open rotaries; or
d. Store enough machine furnish pulp in midfeather chests to keep the machine running through a 12-hour shutdown in the pulp mill.

Our Heritage

Thanks to John Ainsworth's wonderful primer about our industry *(1)*, we know

the Fourdrinier brothers, Henry and Sealy, didn't invent the paper machine. They poured money into an invention by Nicholas-Louis Robert in France in 1799 that was brought to England about 1801. This latest threat to the "drip and slurp" of the sheet mold was an ornery device to get going, eventually leaving the Fourdriniers honorably broke but at least contributing two words that have stuck with the industry ever since. The first practical paper machine was started up in Herts, England, in 1804, designed by Bryan Donkin and operated by Merchant Warrell, the first machine tender. *(21)*

Early papermakers made paper with sheet molds and rags in the 1600s. When the Black Plague ravaged England in 1636, "furnish" was in short supply. The government, concerned with the spread of the plague, clamped down on the use of rags and ordered that all rags were to be burned. Even so, one of our intrepid ancestors in the industry tested the law by grinding rags in his mill. No record has ever been found as to the results of his trial. *(22)*

The hand paddle "stroked by a street urchin" epitomized a time when there was one vat between the grinding and the cooking of rags and the early paper machine. As we discovered that fibers were more abundantly available from trees, learned how to extract them by mechanical and chemical processes, more storage points, chests, were required. Each one, of course, required some kind of "stirring." Perhaps the supply of street urchins dried up or the street urchin's union objected. Anyway, various types of mechanical agitators emerged.

The horizontal shaft with multiple paddles was one of the earliest agitators. In relatively low-consistency stock, of 2% or less, the flat paddles, even at very low speed, did contribute some motion throughout the full length of the chest. However, as higher consistencies were necessary for greater storage capacity or to feed the first crude refiners, motion within the chest virtually ceased, except in the narrow area in the path of each paddle. One can imagine the consternation felt by those early "hands" as great quantities of slowly fermenting stock began to be found in those chests. Perhaps this unwanted fermentation inspired a few to go into the manufacture of an English version of "moonshine!" (Fig. 3-1.)

The first propeller agitators were developed. After a few futile attempts with long rectangular chests which created more settled and fermenting fiber at the far end and in the bottom corners, the midfeather circulator was born. This concept became "king of the agitators" for many years, for it was being used and proposed in new mills into the late 1950s. The midfeather circulator design was molded into a "near

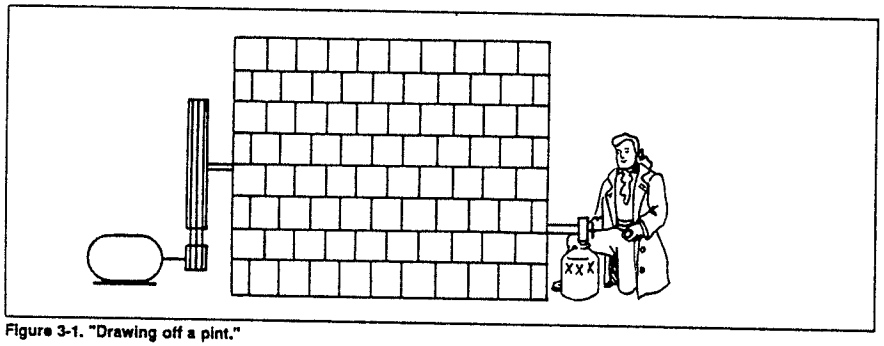


Figure 3-1. "Drawing off a pint."

science" by intensive examination and tests of various configurations. Data were developed that took into account the consistency of the pulp and the type of fiber, degree of refining, temperature and all the ratios of length, width, stock level, size of propeller and proper speed to develop the horsepower necessary for complete circulation. Had "science" had at last prevailed?

All data that went into the selection of this circulator, data that are equally important today in selecting the "modern agitator," were directed toward one goal, the velocity of the stock at the surface as it moved around this "merry-go-round" (Fig. 3-2). Upsets in the feedstream, consistency, freeness, or color moving through the chest in exact order of entry, like children playing "follow the leader," didn't seem to dismay the designers. In some mills, where such chests were, and still are, used on a batch basis for retention time, the stock would eventually become uniform after many circuits through the propeller, sometimes hours of continuous circulation. But, on a continuous basis, the incoming stock has one pass through the turbulent zone around the propeller before exiting to the pump suction. (This assumes the supplier was clever enough to require the feed to be on the upstream side and the discharge on the downstream side of the propeller.)

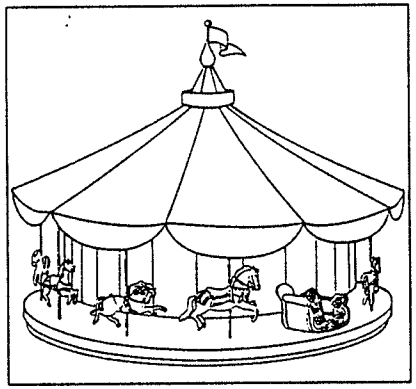


Figure 3-2. "Merry-go-round."

Some designers showed some recognition of this "minor" fault and installed "blending ports" at various positions along the midfeather wall. These were just large holes in the wall which allowed a portion of the stock to "short circuit" the full patch and thus break up the "follow the leader" game. Since there was no turbulence created at these ports, the idea of blending with them was little more than a "shell game" and kept the papermakers even more in the dark as to which "walnut shell" held the greatest stock variations. Of course, some oversized "blending ports" defeated the whole purpose by allowing the rest of the circulation path to slow down and eventually thicken and stagnate. Back to moonshine production again! (Fig. 3-3.)

A review of the A. M. Hurter paper (2) shows how thoroughly the empirical data

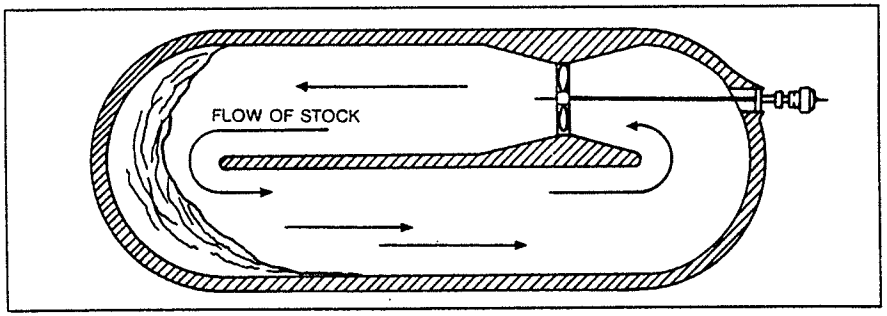


Figure 3-3. Stagnant Midfeather.

were studied for the design of the midfeather chests. The paper by Ryti *et al.* (*3*) studied the dampening effect on several different types of agitation and circulation. Their data implied remarkable results from a particular double agitator, triple-channel chest (although the single-channel design was very poor). A study of the various consistencies and retention times in each of the several chests presented, however, voided the comparison as similar volumetric and gravimetric values weren't followed.

The extreme area required for these relatively low-head midfeather chests and the increased storage required for the higher production paper machines, pushed stock prep designers to improve chest and agitator design in a way that was less sacrificial of mill area.

Let's take an example of a 500 T/D newsprint machine. The mill requires 30 minutes average retention in the machine chest at a stock consistency of 3½ %. An ideal single channel midfeather chest would require 1120 ft^2 of basement area with an overall height of about 16 feet. A vertical cylindrical chest, with the vertical shaft multi-bladed circulator could be designed in a couple of configurations. Placed just outside the machine room, a chest 55 ft high would require just 254 ft^2 at the base. Limiting the height inside the machine room to start at the basement floor and extend through the machine room floor, about 32 ft high, the chest would require 310 ft^2 at its base. As we will see later, the best design for a single side-insert modern agitator would be a chest only 24 ft high requiring 530 ft^2 on the basement floor with the top of the chest being accessible from the operating floor (Fig. 3-4).

The vertical "Christmas tree" had a number of advantages when compared to the midfeather design.

1. It required much less floor space for equal volume or, at equal floor space, could accommodate several times the volume.
2. It could be designed with a completely open, or loosely covered, top accessible from the operating floor, allowing for chemical additions and easy inspection of the furnish.
3. The multiple blades acted as a variable horsepower unit. When drawing down the chest, each exposed blade reduced the horsepower response.
4. Because of the smaller cross-sectional area, there was less unagitated stock on draw-down below the bottom propeller, allowing easy washout.

Disadvantages were a tradeoff. Both designs required internal bearings—the midfeather unit at the propeller in the cross wall and the vertical unit at the bottom of the chest. Both were inaccessible when the chest was full. Both designs acted on

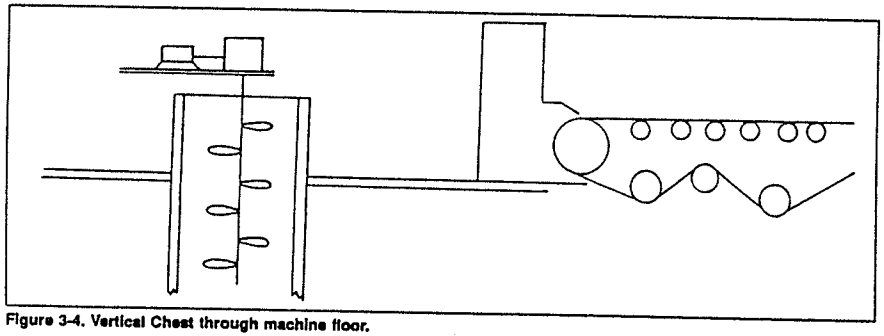


Figure 3-4. Vertical Chest through machine floor.

the principle of circulation. The midfeather must circulate the stock around the channels with sufficient velocity to prevent dewatering. The vertical unit must swirl the stock in a circular pattern for the same purpose. Rotary motion was the key to the success of the vertical "Christmas tree." As many variables were investigated to establish the empirical data required to properly size this design as were needed for the midfeather. The extent of the swirling motion imparted to the stock above and below each of the single blades had to be known in order to properly space them. The "swing" of the propeller blade to the diameter of the chest was fixed, and the rpm necessary to ensure the "swirl" included the periphery of the chest was determined with reference to consistency and type of stock.

None of these "circulators" paid much attention to throughput or the calculated residence time. As long as the surface velocity was achieved, it mattered little whether the stock paid a three-minute visit or took up residence for five or six hours. AND RIGHTLY SO—nothing happening in a rotational flow pattern could do much to upset what is primarily "plug flow"! It might have been convenient to know that a previous, vigorously mixed additive was going to stay in close contact in the same order of entry for two minutes or five hours, but any anticipation of significant dampening or blending of an upset was crushed by the fact that rotational motion afforded little or no random vertical motion. Plug flow through the chest prevailed. Problems that developed with the vertical circulator were similar to those encountered with the midfeather. Occasionally, incorrect sizing led to a beautiful swirling flow pattern that extended almost, but not quite, to the outer wall. Result: a foot or so of wide annular ring of stagnant stock (Fig. 3-5). Even when the initial selection was correct, process changes in the mill led to the need for a higher consis-

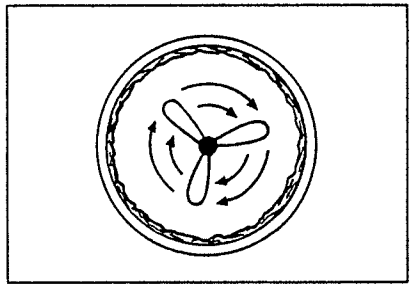


Figure 3-5. "Christmas tree" Agitator with ring.

tency or a different and more difficult stock to move, and that annular ring of dead stock was formed.

As we approached the fifth decade of the 20th century, agitation in most paper mills was either of the midfeather or vertical circulator type, with a few homemade paddles and gates which added an aura of antiquity. The accepted indicator of "good" agitation was still the velocity of surface motion, and every mill and agitator supplier had a little hand-held tachometer fitted with a calibrated paddle wheel which could read out in ft/min when held on the surface of the moving stock slurry.

The Theoretical Approach

The company that first applied mixing criteria to the handling of paper pulp slurries (*5, 6, 7*) learned some hard lessons in the early 1950s. The suspension of paper pulp did not conform to any of the usual criteria of solid suspension. There was no measurable settling velocity. There was dewatering at the surface, but one might just as well try to track a cloud through the sky as learn as much from that observation. The suspension of this fluffy solid in water seemed to disregard all the usual "rules of the game." Engineers were quite familiar with solid suspension, crystals in a dissolving medium, heavy clay slurries, and leaching of powered metals. Name a solid suspension operation in the process industries, and they could list dozens of successful installations.

Even more provoking was how easily the concentrations could be handled. They had produced uniform suspension of solids in 60 and 70% slurries. This innocent-looking fiber, when increased to only 6% concentration, created a mass that could be walked on! Attempts to use a standard mixing turbine either bored a hole in the center or put the whole mass into an uncontrollable swirl, slopping over the edge of the little glass tank. Standard mixing baffles only made things worse, as great clumps of fibers hung up on the vertical sharp edged baffles. It was impossible to measure the viscosity of the slurry, but there was obviously a "pseudo" viscosity that increased at an alarming rate with concentration. A most unusual problem! Solutions didn't come easy. Laboratory tests pointed toward one design only after it was installed in a mill representing a scale-up factor that even today would cause conservative management to shake their heads, was it determined this was an appropriate solution.

The suspension of cellulose fibers was flow-sensitive and, as such, the mechanism for agitation was the same as for fluid blending. This required an impeller with high flow characteristics and low shear. The first choice for a proper impeller was a turbine, but not just any kind of turbine. Earlier fouling experiences with a standard disc-type turbine had shown that something totally different was required: A turbine that was self-cleaning; one that had little structure beyond the hub to catch and hang up fiber. It had to have high flow characteristics and be structurally sound. The spiral backswept turbine was the result. This impeller included a large hub with six full-length blades, retreating backwards from the flow in a spiral pattern and, if necessary, requiring only one circular ring to augment the support given by the welds at the hub.

Next, the swirl component needed to be eliminated and true top-to-bottom random turnover created. Baffles were out; these had been tried with disastrous results. Why not an off-center position like those used for propeller mixers in the petroleum industry? The optimum off-center position was a function of normal level, and the slurry created its own fluid baffle. Predictable random motion was achieved. Process horsepower related to consistency in a remarkable way, being the cube of the change in consistency ($hp \alpha C^3$). The power number of the turbine was determined and the speed was calculated to absorb the required process horsepower. The scaleup of that first random-agitated chest was from an 18-in. diameter laboratory pilot tank to a 35-ft diameter storage chest—amazing! But the curtain had hardly begun to rise on this new era of pulp agitation.

The Nitty Gritty

There is an old expression in the trade that has become a part of the language of Americana: "Back to the drawing board." The "drawing board" in this case was the laboratory. The second installation in the same mill was in a 40-ft diameter chest, and the impeller was now a 9½-ft diameter propeller instead of a turbine. It was quickly realized that the lower power number of a propeller, lower torque for the same horsepower, meant a less expensive drive train and the propeller performed better in stock than did the earlier turbine. The question was then, why a vertical unit with the long, expensive shaft (and the vibration problem mentioned earlier), the costly speed reducer and the maintenance-prone steady bearing? "Why not a side-insert propeller?"

As further studies were made, some of the parameters were already established, but they needed refining. The effect of stock level in relation to diameter, Z/T, was easily established. It was striking! A variable speed propeller unit, installed in the side of a transparent pilot vessel was run at a speed just sufficient to promote motion across the bottom and turn over the entire contents at a minimum level (by definition "complete motion"). At this speed and horsepower response, the level was gradually increased until motion at the surface stopped. This proved to be a level equal to about 70% of the chest diameter, $Z/T = 0.7$. Above that level, additional speed (hp) was required to maintain motion, and at a ratio greater than 80% of the diameter, $Z/T = 0.8$, the increase became extreme. Rule Number 1, the most efficient design of a vertical cylindrical chest, for what was defined as complete motion, was a stock level of about 80% of the chest diameter or a Z/T of 0.8.

It was also observed that larger impellers produced the same result as smaller impellers but at less required horsepower. This was consistent with the "flow-sensitive" assumption and the blending concept. So a relationship was established between process result and the impeller diameter, D/T. (22)

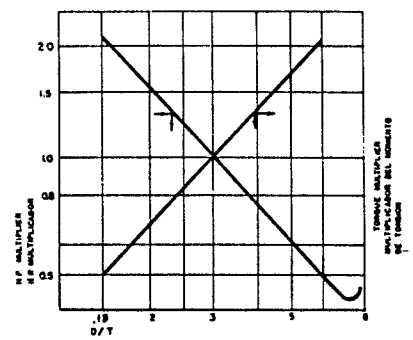


Figure 3-6. D/T vs. hp—Blending.

The D/T relationship was a useful way of getting at the flow (Q) and head (H) relationship of the impeller. Since the random agitation of paper pulp had been determined to be flow-sensitive, it was obvious that process horsepower would decrease with increasing impeller diameter, $hp \alpha (V_D/T)^n$. As in any flow-sensitive system, there was a point at which the ratio became so large that no amount of off-center position, or baffles, would keep the contents from swirling. A plot showing this effect is presented in Fig. 3-6.

With these data, the selection of impeller size became an economic balance between the cost of power at a particular mill site and the capital cost of the equipment. (High hp—low initial cost, Low hp—high initial cost.) More will be said about this relationship in a later chapter.

The investigation of side-insert propellers produced another unexpected revelation. Unlike liquid-blending systems which require an angular entry for the side-insert propeller in order to produce random motion, the pseudo-viscosity of paper pulp allowed an on-center mounting without producing rotational motion. But there were still some setbacks that plagued these upstarts in the paper pulp agitation business.

Initial laboratory work was done with repulped normal yield bleached sulfite pulp purchased from a local manufacturer of high-grade photographic paper. Gradually as reports of less than satisfactory performance came in from the first few installations, it was realized that "all cats aren't black." Groundwood pulp took more horsepower as consistency increased than did sulfite. Bleached northern kraft required more horsepower at all consistencies. Additional tests were required. Using bleached sulfite as a "standard," stock factors were established for the common pulps used by the few mills thus far penetrated, primarily the Northeastern United States and Eastern Canada. Then the roof fell in!

A large installation was made using vertical off-center propeller units in a large mill in Southwestern Washington State. The results of that installation almost caused this writer to become a new, but unemployed, resident of the Northwest, 20 years before a more pleasurable move. All the units were underpowered by 100% or more. A decision was made to replace

them with completely new machines: bigger shafts, heavier speed reducers and double the horsepower. What had gone wrong? The pulp was unbleached softwood kraft, permanganate number about 32. This required a higher stock factor than the company had ever experienced. More importantly, the pulp was made from 100% Douglas Fir. It was a costly lesson but one that had to be relearned only once; Slash Pine from the forests of Northern Florida and its bordering states held similar nightmares for the agitator supplier. As more was learned about this extremely long, coarse-fibered pulp, a crash program aimed at testing as many different pulps as possible was begun. Soon barrels of wet furnish began rolling into the lab from all corners of the United States and Canada. The costly setback was turned into a plethora of data covering a wide range of consistencies for virtually every class of virgin pulp used on the continent. Unbleached kraft pulp from various wood species and through the whole range of cooks, permanganate numbers from 18 to 42, later to include the even higher-yield cooks, were dumped into the lab test tank and evaluated. Another problem was solved because somebody had asked, "Why?"

High-density Storage

In the earlier years of this century, the production capacity of most paper machines was relatively low. Even in a multi-machine mill, the storage capacity of most low consistency midfeather chests and vertical chests was sufficient to keep the machines running through routine "downs" in the pulp mill. But as tonnage rates increased and space requirements became extreme to ensure uninterrupted feed to the machines, even during a minor shutdown in the pulp mill, the concept of storing pulp at higher, even unpumpable, consistencies became a necessity. A 50-ton storage chest at 4% consistency was a big chest, over 40,000 ft^3. Even as a vertical cylindrical chest, it would be on the order of 30 ft in diameter by 60 ft high. A midfeather chest would be almost unthinkable, requiring some 2400 ft^2, and yet a three-machine mill making only 500 T/D could run 2½ hours on that storage capacity, if the pulp mill supply were cut off for any reason.

If there was a way to store the pulp at 12% consistency and get it back out at some lower, pumpable consistency, that 40,000 ft^3 chest could store 150 tons, extending the running time to over seven hours!

There were undoubtedly many false starts in the design of a high-density storage concept; most were dismal failures lost in the annals of history. But one method was tried, picked up by all the old line suppliers, and stuck with for more years than we care to remember. The Mining Nozzle Concept!

The principle behind these early high-density storage chests was simplistic. Except for the operating problems that increased with the equipment age, it was quite effective. A typical chest for storage of 100 tons of pulp at 12—14% consistency would be 28 ft in diameter by about 45 ft high. The bottom was essentially flat, a shallow slope toward the pump suction. A typical agitator for this size chest was a side-insert unit, 36-inch diameter propeller driven by a 25- or 30-horsepower motor. At a pump-out rate of 500 T/D diluted to 3½ to 4% consistency, the bulk of the dilution water (80%) was introduced at about 40 psig, just above and behind the agitator. The agitator was installed just over or near the pump suction. The remainder of the required dilution water would be introduced in three separate locations, perhaps 135, 180 and 225 degrees around the chest from the agitator and located relatively near the bottom of the chest. Water introduced at these locations was through an oscillating nozzle, an orifice of about ⅜ ins., at 80 psig,

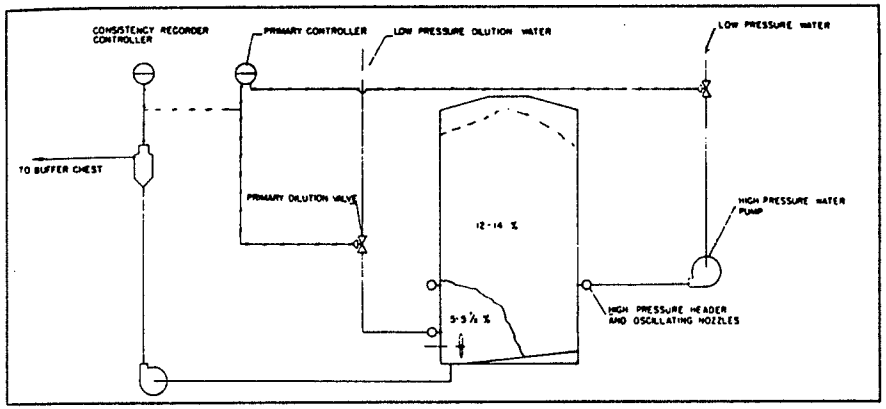


Figure 3-7. Hi-density with Mining nozzles.

from high-pressure booster pumps. The purpose was to "mine" the high-density pulp, cut away the cake and slush it toward the pump suction, thus giving credence to the term "mining nozzle" (Fig. 3-7).

Meanwhile, back at the agitator, a pumpable slurry was created from the "mined cake" and dilution water "sucked" into the suction side of the pump and discharged to a low-density chest. So far, so good. But anyone who has looked at a consistency chart plotting the performance of such a chest, or even sat and listened to the rising and falling whine of the discharge pump, knows that what was coming from that chest was far from uniform. We used to refer to it as "cabbage heads and water." The low-density chest, to which this flow was directed, became known as the "leveling chest." The reasoning was not hard to understand and the suppliers never intended the high-density storage chest to operate without a leveling chest. The agitator was entirely too small to ever create and hold a continuous "bubble" of low-density stock. Actually it alternately "grabbed" a large volume of water producing a big bubble of dilute stock which then collapsed as the pump discharge increased, allowing large clumps of thick stock to follow for a short time. The mining nozzles ensured there was a continuous drift of slightly diluted stock across the bottom toward this alternately increasing and decreasing diluted zone. The consistency variations were often from 0 to 6% actual consistency over extremely short intervals. Since the leveling chest was often a vertical "Christmas tree" circular, it generally was quite large in order to have any chance of dampening these violent swings into something usable.

This was only one of the "livable" problems plaguing the mining nozzle design for high-density storage. Other problems were much more serious.

Sometimes the blades of the adjustable pitch propeller would move: if to a high pitch, the unit would shut down because of the overload; if to a lower pitch, little motion would be imparted over the pump suction. In either case, the evacuation of high-density pulp would come to a standstill. Sometimes one or more mining nozzles would plug with fiber from too rich a white water used for dilution, with similar results. More water was needed, often meaning manual hosing from the top of the chest, thus reducing the consistency throughout the chest and defeating the whole purpose of high-density storage.

Sometimes a paper machine was dropped off the line for a period of time to

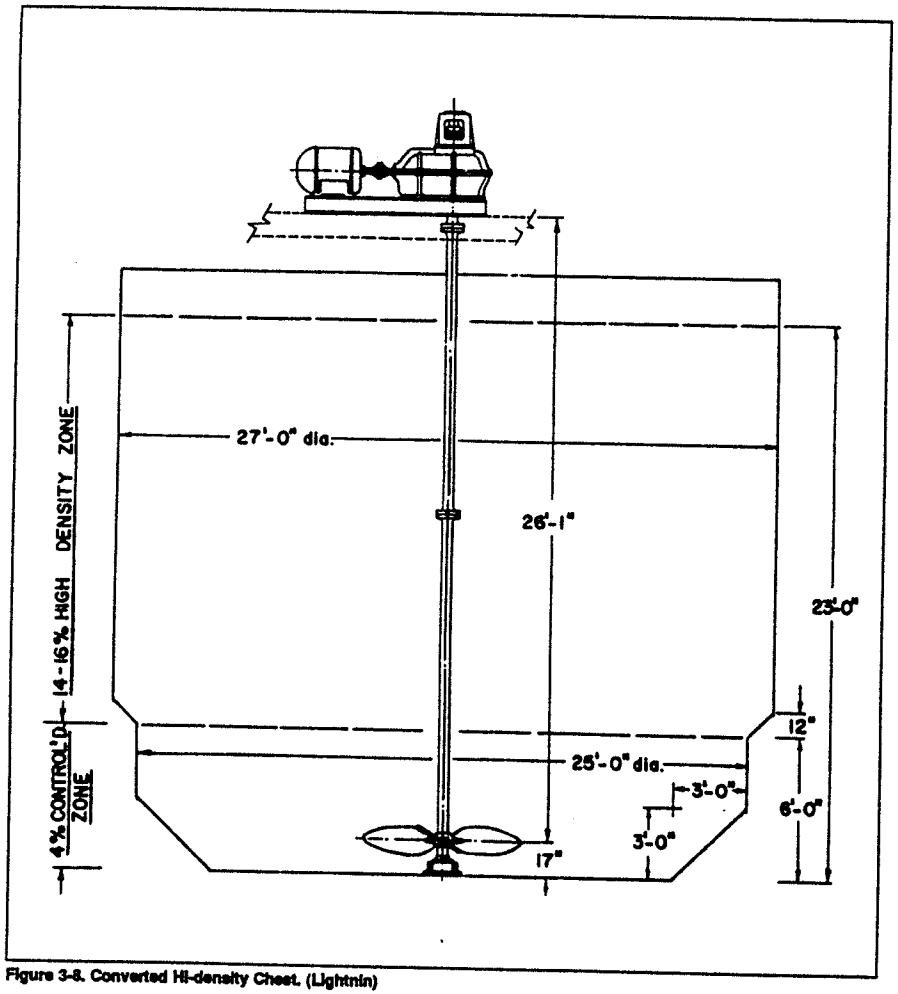


Figure 3-8. Converted Hi-density Chest. (Lightnin)

avoid a prolonged shutdown. Time is the worst enemy of wet pulp; excessive dwell time left large quantities of dense pulp lying on the bottom of the chest, 180° from the agitator. It didn't move very well at full flow from the swing nozzles; now it didn't move at all. The result usually was a pile of black, rotting stock that gradually broke off, bit by bit, and found its way into the paper mill as rotten furnish.

One of the worst catastrophes was caused by a combination of reduced withdrawal and steady dilution. Many of these chests had solid tops, grouted in or fixed to the tile-lined side walls. As the chest continued to fill at a rate greater than the outfall, abetted by the 80-psig pressure water from the mining nozzles, something had to give, and it usually was the cover, taking with it a portion of the side wall.

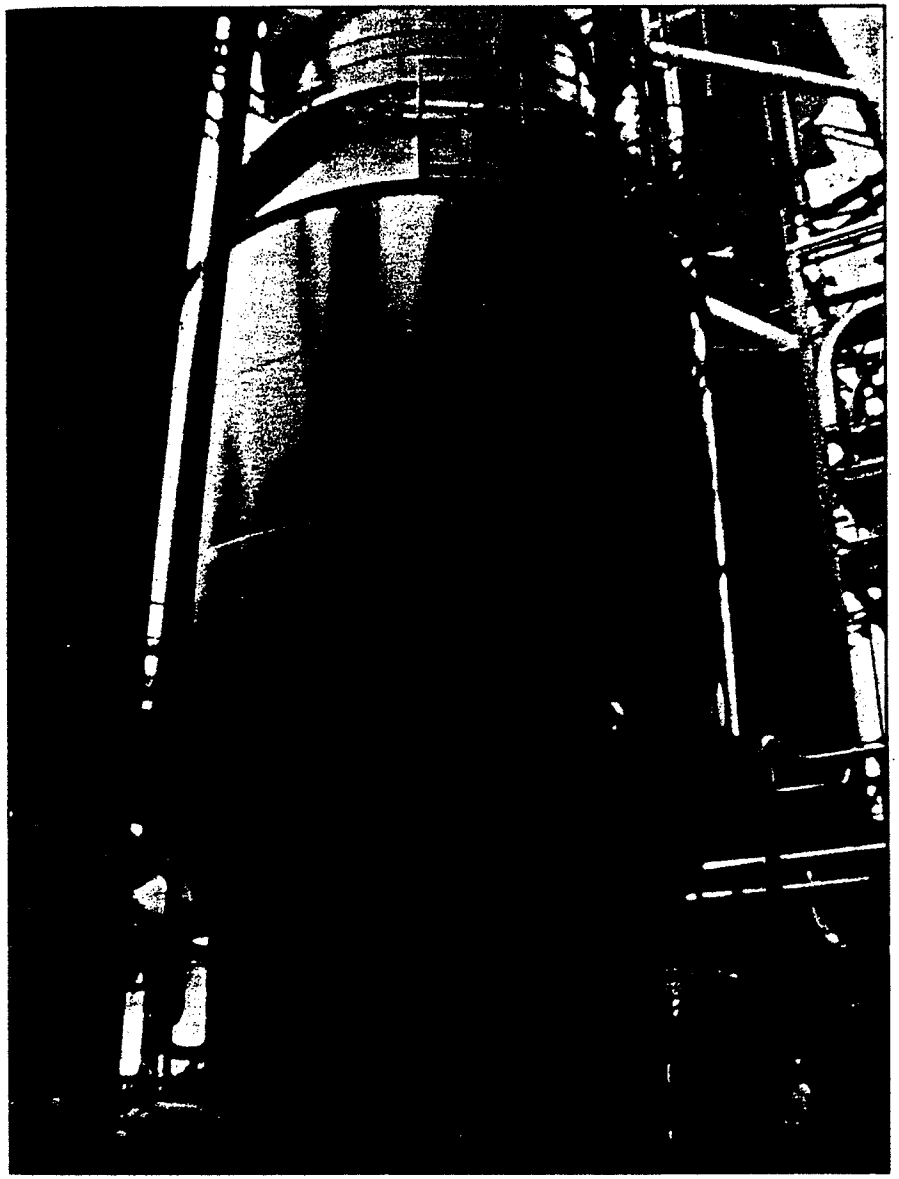


Figure 3-9. Reduced Bottom High-density Chest. (Lightnin)

The first universally accepted, high-density storage chest had "feet of clay!"

Those people, referred to earlier as neophytes and upstarts, *(5, 6, 7)* had already successfully experimented with the zone

agitation principle. Several large vertical chests had been fitted with single-impeller vertical-shaft agitators, not to move the entire contents but to induce complete motion across the bottom and to a predetermined height. The remainder of the stock moved in a plug downward into the agitated zone. These installations were all done at 4-5% low-density. There were a few such conversions made to existing shallow high-density chests, but the horsepower was excessive to ensure a complete bottom zone in these large diameters. The minimum level produced was so high that it robbed the capacity of the chest. Even when a portion of the bottom was filled in to create a smaller diameter and reduce the horsepower required, the cost was prohibitive except in extreme cases of total dissatisfaction with the mining nozzle system (Fig. 3-8).

Something totally new was needed. The leveling chest was essentially a blending chest, and because its agitator was so inefficient, it had to be inordinately large. A blending chest with only 10- or 12-minutes retention could be completely satisfactory if equipped with a modern side-insert agitator. Why not move a properly-sized blending chest under the high-density storage chest? But 10-minutes retention at 4% for 500 T/D only required an 18-foot diameter chest, about 10 ft deep. How could you put that "pot" under a 28- or 30-ft diameter, high-density chest? Easy! Build a chest with two diameters, the major diameter 1.6 or 1.8 times the minor and join the two with a steep 60-degree transition section. The upper section can then accommodate the maximum storage capacity. Now an agitator could be designed to sweep the entire bottom of the chest with constant stripping away of the high-density pulp above the junction of the conical section and the straight shell of the lesser diameter. Dilution water, at low pressure, could be introduced just above and to one side of the propeller, and a simple control system could be programmed to allow 80% of the dilution to the diluted zone with the remainder added at the pump suction for trimming to an exact consistency level. Gone were the high-pressure pumps to feed the mining nozzles, and the leveling chest was no longer required. Properly controlled stock could be fed directly to refiners, paper mill storage or any other processing unit in the flow sheet (Fig. 3-9).

I would like to report instant acceptance by the industry, with a chorus of cheers from the "peanut gallery." But like all innovations, from the first paper machine to Fulton's Folly, the Wright Brothers' toy, and others too numerous to mention, we were laughed at! Who could design such a ridiculous looking chest, and who would guarantee it wouldn't collapse of its own weight?! "Reduced-bottom tower, indeed! Looks more like an upside down milk bottle!" So the "milk bottle" was born but remained a "one company" dream for many years. One courageous pair of mill engineers in Canada *(4)* caught the vision, and the first reduced-bottom controlled-zone chest became a reality although not used initially for high-density storage. It was a giant step forward with 200 tons of groundwood storage, but modest in shape. The ratio between the reduced diameter and the major diameter was only 1.35. The tank was of steel construction with outboard column supports. Gradually this concept became accepted in other mills and the major tile constructors designed free standing chests with ratios up to 1.8. Capacities were increased beyond even the most optimistic projections until chests of 300, 400 and 500 tons of capacity became commonplace *(5, 7, 15)*. Today it is virtually impossible to find a bid specification for a new high-density tower that doesn't specify the reduced-bottom design.

The controlled-zone principle allowed many other applications. A low-density broke tower feeding a fraction of its capacity back into the stock system, is efficiently controlled with only the bottom zone in agitation, thus reducing the capital and energy costs that would have been required to agitate the entire chest. The same principle was applied to long rectangular chests by agitating one end to continuous uniformity and allowing the bulk of the stock to feed into the zone by hydraulic gradient on a continuous basis—End Zone agitation (Fig. 3-10).

What made it all possible was an understanding of the minimum residence time required to blend cyclic upsets and maintain uniformity at a reasonable horsepower level. 10-to-12 minutes was established for the high-density design, but data were developed to allow lesser residence times at the expense of higher horsepower inputs. Exact residence time requirements for known cycles of variation could be calculated using a modification of the classic MacMullin, Weber curves *(14)*, which were presented years before the first computer or hand-held calculator in 1935. [An interesting side note, someone in the petroleum industry recalculated the relationships using a sophisticated computer, plotted the data, and found the curves looked like a tracing of the designers' work!]

The application of a mixing science developed in the process industries was being applied successfully to the paper industry. This is our next area of study.

Figure 3-10. End Zone Agitation. (Lightnin)

Chapter 4:

Agitation vs. Mixing

It was difficult for a young chemical engineer, trained in the scientific jargon of mixing technology with the chemical industry, to assimilate such terms as couch roll, breast roll, dandy roll, broke, slice and dozens of other colorful names permanently tagged to a centuries-old industry. Even a trainee on a paper machine knew the more earthy term for the squirts at the couch roll, much to my embarrassment, but to be told that mixing was mixing and agitation was agitation punctured my ego like a pin popping a balloon. "Son, a mixer is that little thing hanging on the side of that 55-gallon drum; an agitator is that big thing on top of that stock chest." Apparently mixers were associated with "tanks" and agitators with "chests." But even *that* wasn't always right. An agitator is defined in *Websters Dictionary* as "an implement or apparatus for mixing." That defense was met with, "Webster, ain't he the tour boss that got run off last week?"

I learned not to dispute the old hands, and even came to take a certain pride in the subtle differentiation. After all, a stock agitator was much bigger than a side-entering mixer on a crude oil-blending tank, even though that blending tank might contain over four million gallons (100,000 bbls) of oil.

The process industries also had their preferred generic handles. How many young graduates today can immediately describe, without a 1941 edition of Riegel, an Imhoff tank, a soap crutcher or a blunger? For that matter, how many young paper mill engineers can tell you the origin of "couch" as now applied to that suction roll and the pit beneath it? So we'll stick with "agitation," but let's see how closely the terms intertwine.

In discussing the gradual evolution of this unit operation and the equipment to perform it, two words that stand out are "circulation" and "random motion." The earliest agitators, all the way up to the 1950s, were all "circulators." They moved the pulp in such a fashion as to prevent de-

watering. But like the cinema cowboy hero whose shoulders and height disappeared with the removal of his padded jacket and high-heeled boots, the circulator was found lacking when it was finally understood that uniformity of concentration fed to another piece of process equipment was more important than some visible dewatering at the surface of a storage chest. Agitators that produce random motion, on the other hand, produce circulation and continuously mix (there's that word) pockets of varying concentration or freeness into a uniform mass. As we have previously shown, the sophisticated refiners, screens, cleaners and proportioning devices used today in a modern high-production paper mill rely on uniformity to perform at peak efficiency.

Flow Head Relationships

The rotation of any type of impeller in a fluid absorbs horsepower and produces two reactions; flow (Q) and head (H). (It also produces heat, but let's leave Mr. Joule out of this discussion). Different mixing (agitation) problems require different ratios of Q and H.

The following table portrays a rough spectrum of mixing operations beginning with requirements of high *head/low* flow and progressing to high *flow/low* head:

H-q solids dispersion
liquid liquid dispersion
gas liquid contacting
solids dissolving
solids suspension
heat transfer (in waterlike liquids)

h-Q miscible liquid blending

The agitation, as well as blending, of paper pulp slurries fits best in that last slot of the mixing spectrum. A large amount of flow is required to move the mass, with enough head (turbulence) to ensure random intermixing and create an homogeneous slurry.

The D/T relationship first mentioned in Chapter 3 became one of the most important tools in the early selection procedures for the random motion concept of stock agitation. It wasn't a new tool, because D/T had been used for years in selecting turbine mixers for the process industries. But in those applications, the need was to find the precise ratio that exactly met the process needs. It was restrictive when applied to such mixing problems as solid suspension, solids dissolving, gas dispersion and the dispersion of two or more immiscible liquids. With all the high-powered mathematics stripped away, most mixing operations could be reduced to a simple requirement of (a) a particular amount of flow (circulation) and (b) a particular amount of head (turbulence).

Mathematically, $hp \alpha QH$. But it is difficult, except under laboratory conditions, to measure finite volumes of flow and head, and it is equally difficult to assign specific values of Q and H to a particular mixing operation. Figure 4-1 is an example of how the D/T ratio was used to find the minimum horsepower level for several different operations. These are just typical curves and within any one category, such as solids suspension as in Figure 4-2, the minimum point might be shifted to the left

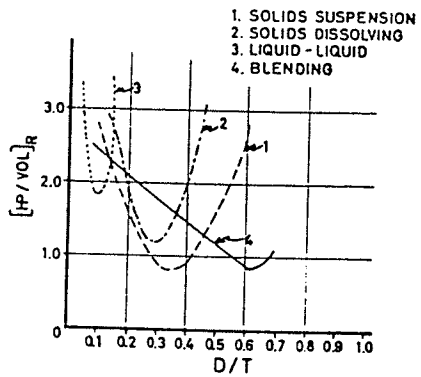


Figure 4-1. D/T vs. hp—Several applications.

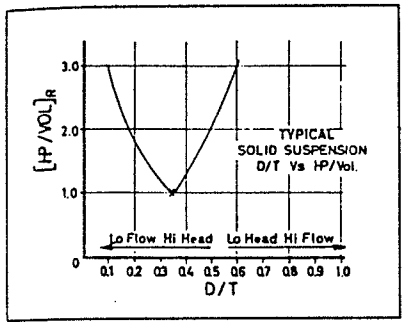


Figure 4-2. D/T vs. hp—Solids suspension.

or the right on the abscissa, up or down on the ordinate, depending on the physical characteristics of the solids to be suspended and the liquid acting as the carrier. Though the perihelion point on that curve represents the minimum horsepower requirements, it isn't necessarily the optimum selection. If the manufacturer didn't have a standard impeller of that exact diameter, the operating speed might be just at the break in a reducer size, and a lower *impeller/higher* speed could save hundreds or thousands of dollars. A larger impeller and slower speed might allow the use of a much smaller diameter and a less costly shaft.

It should be understood that any selection, made from any point on that curve, would meet the process requirement. The perihelion (minimum point) produces exactly the correct amount of turbulence (velocity) to overcome the settling velocity of the solids we wish to suspend. It also produces the exact amount of flow (pumping) to distribute the suspended solids in a uniform slurry throughout the vessel. If we were to move to the left but stay on the curve, this would result in a higher velocity component, more than enough to suspend the solids. The higher horsepower level is needed to produce the necessary flow to yield uniformity. If we were to move to the right but stay on the curve, the opposite is true. Now we have ample flow but need more horsepower to provide the velocity for suspension.

For an even more vivid example of why the minimum horsepower level isn't always economical, or even practical, let's go back to that 100,000 bbl oil storage tank mentioned earlier in this chapter. That tank would likely be 110 ft in diameter by 60 ft high. Whether used for oil or gasoline, the time allowed for blending would be perhaps 8-to-12 hours and the horsepower required could range from 25 to 75. This would normally be applied by one or more 25-horsepower, side-insert units with 28-in. propellers. Why should we use even 25 horsepower with that little propeller when a gasoline blending curve tells us the same result can be obtained with only 2 horsepower? All we need is a vertical unit with a 55-ft diameter turbine! The 28-in. propeller at 420 rpm represents 3750 inch pounds of torque. That 55-ft turbine would theoretically run at 0.35 rpm requiring 360,000 in.-lbs. of torque. The optimum D/T doesn't sound like the best idea! If it was a floating roof tank (and it likely would be), we'd have another interesting problem or we'd be drilling for oil in the center of a 100,000 bbl storage tank (Fig. 4-3).

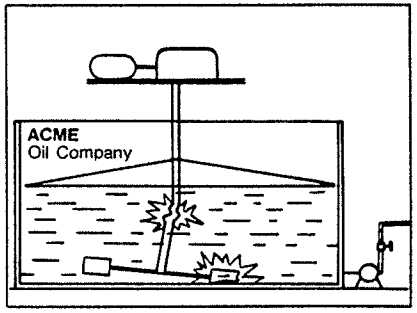


Figure 4-3. Vertical Agitator on floating roof.

The agitation of paper pulp is flow-sensitive and, as such, follows the horsepower versus D/T curve for blending shown in Chapter 3. However, the de-

signer rarely exceeds a D/T of 0.4 in making a selection. It is also difficult to ensure predictable results if the propeller selection falls below 15% of the chest diameter, a D/T of 0.15. Usually, several choices of propeller diameters are looked at, and the final unit selection is based on the overall economics of capital and operating costs. A typical example of these and other considerations is explained in the following case:

A 20-ft diameter dump chest has a normal stock level of 16 ft, but as it cycles between pulper dumps, the level is run down to as low as 8 ft before filling up again. A very low horsepower level could theoretically be applied if a D/T of 0.4 were used. But this would mean using an 8-ft propeller which would look ridiculous in this chest and vortex violently as the level dropped to 12 ft and be partially exposed when the level reached 8 ft. Excessive splashing and air incorporation would result. An optimum selection for this chest would be a 42-in. propeller. Its higher speed, lower torque would provide a much more economical choice, even at the cost of additional horsepower. Obviously, if only power savings were considered, we would be in an untenable situation of extreme capital cost and an unacceptable process result.

Chapter 5:

Impeller Horsepower Response

In earlier explanations and discussions, we frequently mentioned the energy absorbed by a rotating impeller produces various ratios of flow and head, $hp \alpha QH$. We have also referred to certain similarities between an agitator impeller and a centrifugal pump. Now let's define some of the differences lest we get too comfortable with what similarities do exist.

A centrifugal pump does pump fluid, but in a severely restricted manner due to the discharge size and the pipe line size to which the pump is rigidly attached. An agitator impeller isn't so rigidly restricted. Except in some very specialized designs, its discharge is free and allowed to entrain additional fluid, ultimately generating many times the flow initially produced at its source. It isn't surprising, therefore, that the affinity laws for an agitator are somewhat different from those you are familiar with for a centrifugal pump. Let's restate these for both devices *(9)*.

In all cases, the following nomenclature will be adhered to:

hp = horsepower
H = head
N = operating speed (rpm or rps)
D = impeller diameter (ft. or in.)
Q = flow (gal/min or ft^3/min)

A. Affinity laws for centrifugal pumps
1. $hp \alpha QH$
2. $Q \alpha ND$
3. $hp \alpha N^3D^3$
4. $H \alpha N^2D^2$

B. Affinity laws for agitators
1. $hp \alpha QH$
2. $Q \alpha ND^3$
3. $hp \alpha N^3D^5$
4. $H \alpha N^2D^2$

It's obvious the proportional relationship for Q and hp *(2, 3)* are quite different for an agitator impeller than for a centrifugal pump. Given a 10% increase in diameter at constant speed, the pump will have a 10% increase in flow and the agitator will

have a 33.1% increase. The horsepower response of the pump will increase by 33.1%, but the agitator will increase by 61%. Now I know some of you are going to cry: "Foul—a pump isn't that simple!" And you're right. Those increases in flow and horsepower will occur only if the head is allowed to increase in accordance with the fourth relationship, $H \alpha D^2$, and if the efficiency remains the same. Those of you who have constructed new curves for changes in pump geometry know the problems that arise. But it isn't that complicated with an agitator. For one thing, we have geometric similarity. If you want to change from a 28-in. square-pitch marine-form propeller to a 30-in. propeller at constant speed, you can be certain that all the geometric ratios of the propeller are the same and the horsepower will increase by the fifth power of the diameter change. Because the propeller isn't in a restricted chamber and discharges freely, whatever head change occurs doesn't affect the horsepower response. Changing the diameter of a pump impeller doesn't involve changing the height of the vanes, so you don't maintain geometric similarity with the original diameter. This further detracts from the stated proportionality of D^3.

So let's get on with the power response of an agitator impeller. The correct horsepower relationship *(10)* is:

$$hp = \frac{k N_p \rho N^3 D^5}{g} \qquad (1)$$

where:

N_p = A power number specific to the type of impeller
ρ = Density of the fluid
N = Operating speed
D = Diameter of the impeller
g = Gravitational constant
k = Constant factor to convert units to horsepower.

When dealing with paper pulp slurries, we assume the density of the slurry is equal to water at 60°F. There is a correction factor for the pseudo-viscosity of pulp slurries, but at 4% b.d. consistency it is designated as 1.0. When using N as rps, D as ft., ρ as lbs/ft^3 and g as $32.2\ ft/s^2$, the equation reduces to:

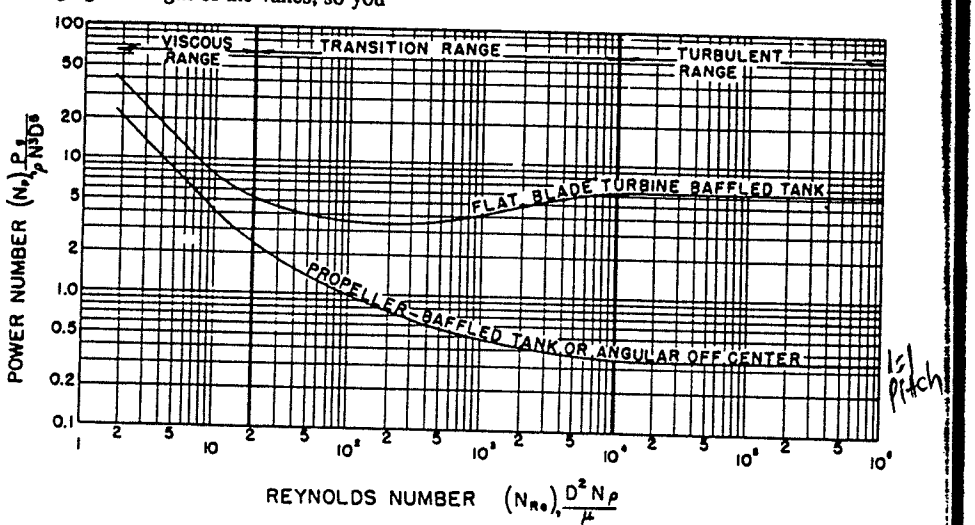


Figure 5-1. Reynolds Number vs. Power Number.

$$hp = \frac{N_p N^3 D^5}{283.814} \qquad (2)$$

If you have followed the text to this point, you know that several different types of impellers have been used to agitate paper pulp. Each one has a unique power number, N_p, which is constant regardless of size, as long as it has been scaled-up geometrically similar. Figure 5-1 is a typical plot of Reynolds Number versus Power Number for a radial flow turbine and a marine-form square-pitch propeller *(16)*. Since we account for an increase in viscosity or lower Reynolds Number by a simple multiplier, we use the Power Number associated with the turbulent range or flat portion of the curve. The standard three-blade, square-pitch propeller will have a Power Number of 0.36. The older spiral backswept turbine, though a little sensitive to D/T, would have a Power Number of approximately 2.9.

For those of you who enjoy "factor labels" and want to prove any constant before believing it, let's go back to that power Equation (1) and work it out:

where:

N = rps
D = ft.
ρ = 62.4 lbs/ft^3
g = 32.2 ft/s^2
$N\rho$ = dimensionless.

Okay, so the density of water at 60°F is 62.37! But the gravitational constant in Madison, WI. is 32.164 and the denominator would become 283.633. We've had our fun; let's settle for 283.8. If you can calculate power response closer than that, you don't need to read this book.

"In the beginning was the Logos" and also was the adjustable pitch propeller. (Notice I said adjustable pitch, not variable pitch. As an old B-24 pilot, variable pitch meant something you could change with a lever from the pilot's console, not something requiring a lockout tag, a 10-lb. sledge and an Allen wrench with a 24-in. cheater.) One was a blessing, but was the adjustable pitch propeller a panacea or plague? It will probably stay with us for a long time. It has some advantages. When the supplier "wasn't too sure" of the power response he had anticipated, there was a cryptic message attached to the installation instructions. "After determining initial power response, use the attached graph to increase or decrease power consumption by changing the blade angle (often called flare) to affect _____."

In other words, adjustable pitch was a crutch to alleviate the ignorance of the supplier. But there were good reasons, too. You, the user, weren't always perfectly honest with the supplier, and sometimes a unit sized to operate in 4% consistency might easily encounter 3 or 5%, resulting in an underload or a severe overload, regardless of the process result. So a change in pitch could bring the motor and drive to the correct loading level. There was another advantage. Many chests had restricted entries. An 18-in. manhole might have been the largest opening in the chest. Trying to install a 54-in. propeller might have presented a problem unless it were thrown on the floor while the chest was being built. But what if a blade broke during service? How would you replace it? So a propeller with removable blades was an obvious answer and if removable, why not adjustable? So, let's recap, the reasons for an adjustable pitch propeller are:

1. Ignorance on the part of the supplier;
2. Poor planning by the user;
3. Ability to change power response.

Well, that sounds sufficient. One of the problems encountered with adjustable pitch propellers was the propensity to change pitch while in service, often drastically and catastrophically. Most often (Murphy's Law) moving to a higher pitch angle resulted in extreme overloads and

sometimes complete "wrecks"—broken blades, bent shafts, rupture of the chest wall, etc. Some suppliers did better than others. Jones Division, Beloit Corp., developed a propeller that met the needs of possible changes with an eye toward exact pitch setting and minimum problems in the case of failure. The Jones propeller (Fig. 5-2) incorporated a blade design with a self-locking pitch. If the locking device did fail, the blade would move toward zero pitch, decreasing agitator load, thus avoiding the "wreck" usually experienced with feathering to maximum load. In addition, the propeller design included a machined pitch block for a specific angle which made pitch changes a simple substitution of a pitch block rather than an elaborate exercise with a straight edge and protractor.

ProChem of Canada took a different route. They believed their power response data were so accurate that the excellent performance of its "Maxflo" impeller would require only a fixed-pitch design specifically tailored to the process requirements. If it were too large for the 18-in. manhole, it required the chest should have an opening large enough to accommodate the impeller. With all the reasons for adjustable pitch propellers considered, it still remains primarily an advantage for the supplier.

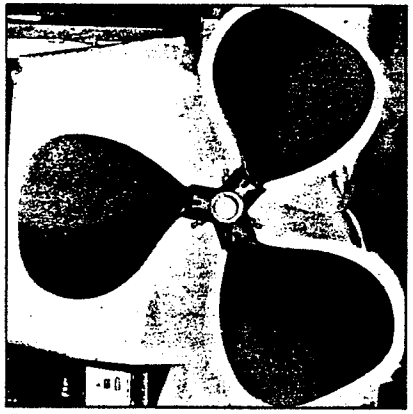


Figure 5-2. Adjustable Pitch Propeller. (Jones)

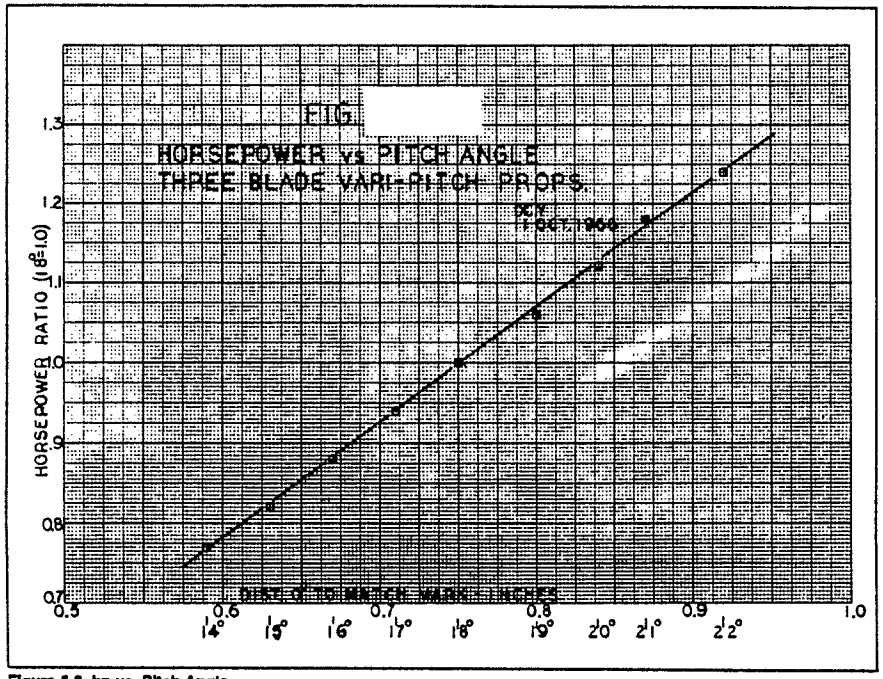


Figure 5-3. hp vs. Pitch Angle.

The plot shown in Fig. 5-3, describes the effect of pitch angle on horsepower. A power ratio of 1.0 is used for a three-blade marine-form propeller at 18-degrees, square pitch. Consistency also affects horsepower response. Most suppliers have adjusted their basic data to the basis of 18-degree pitch and 4% b.d. consistency. We are afflicted with the standard speed imposed by the American Gear Manufacturers Association (AGMA) if gear drives are used and by stock sheave diameters when we use the more amenable v-belt drives. Quite naturally, one of the standard speeds doesn't always allow the optimum efficiency if we are limited to square-pitch propellers. Obviously, some adjustment must be made. When using relatively low power inputs, say up to 50 hp, picking a speed for a little bit more than the process hp required and using the next size motor isn't a great penalty. This is quite common for propellers 36 ins. in diameter and smaller. But at higher power levels and greater diameters, this can become costly. Fig. 5-4 displays a plot of hp response vs. consistency. Let us go through some typical examples:

Before we get started, let's define the usual loading practice. It's good practice, with the hydraulic swings that occur with agitators, to load to 90% of motor rating. This allows a generous factor for unpredictable surges. We also allow 10% for drive and packing box losses; thus the impeller power response to match or exceed

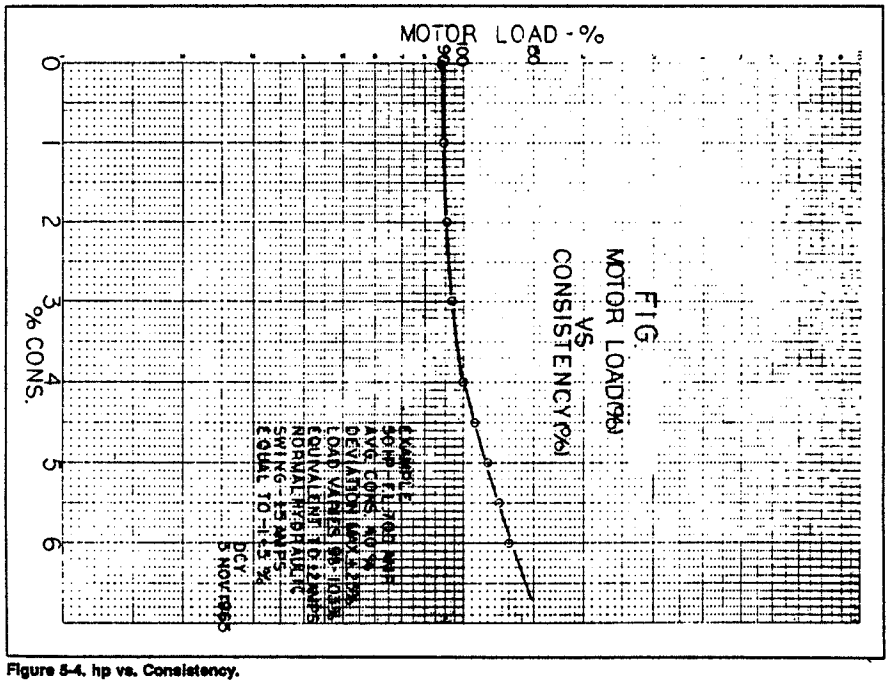


Figure 5-4. hp vs. Consistency.

the process requirement will be 80% of the nameplate rating of the motor.

Now, we have a decker chest which is to operate at 5% b.d. consistency. It has been determined that a 42-in. propeller, operating at a speed sufficient to absorb 40 impeller horsepower, will meet the process requirement. A 50-horsepower drive will be required, and we need to know the operating speed.

Using Equation 2:

$$hp = \frac{N_p D^5 N^3}{283.8}$$

$$N = \sqrt[3]{\frac{hp \times 283.8}{N_p \times D^5}}$$

Np = .36

D = 42" = 3.5'

$$N = \sqrt[3]{\frac{40 \times 283.8}{.36 \times 3.5^5}}$$

N = 3.916 RPS = 235 RPM

But Fig. 5.4 gives a factor of 1.13 for 5% consistency.

hp = 1.13 x 40
hp = 45.2
hp α N^3

$$N = 235 \times \sqrt[3]{\frac{40}{45.2}}$$

N = 226 rpm

With 5V-belts a standard speed of 231 rpm is available. Increasing the speed to 231 rpm from 226 rpm will yield:

hp = 40 x $[231/226]^3$
hp = 42.7

Is this safe? Well, 42.7/50 = 85.4%, but if we follow the rules we only have 45.0 hp to work with at 90% motor load and therefore, 42.7/45 = 94.9%. This is too much unless we want to crowd the service factor. Now, if we have an adjustable-pitch propeller, as in Fig. 5-3, 40/42.7 will require a factor of 0.94. We find 17 degrees will give a factor of 0.94.

Therefore, the selection would be the 42-in. propeller set for 17 degrees, operating in 5% b.d. stock at 231 rpm to consume 40 hp at the propeller and load the motor to not more than 90%.

Let's try another one:

A high-density tower requires 150 installed horsepower using a 54-in. propeller to operate in a controlled consistency of 3.5% b.d. consistency. Using the same rules as above, we are looking for the speed at which to absorb 120 hp at the impeller. The first calculation will give us 3.71 rps or 222.6 rpm. Now, at this horsepower level we might find an appropriate speed with a v-belt drive, but that would require a more expensive lower speed motor, 1170 or even 870 rpm. It is more likely that we would just use a standard parallel shaft speed reducer driven by a 1750-rpm motor. The standard AGMA reduction ratio shows an output speed of 230 rpm.

hp = 120 x $[230/222.6]^3$
hp = 132.4

But at 3.5% b.d., the factor (Fig. 5-4) is 0.97.

hp = 132.4 x 0.97 = 128.4

Now a pitch setting which will give us a factor for 120 hp (Fig. 5-3).

f = 120/128.4 = 0.93, 17 degrees gives 0.94
and hp = 128.4 x 0.94
hp = 120.7 acceptable.

Most suppliers wouldn't have to go through these individual calculations. However, the advent of the pocket calculator in late 1972 made these calculations child's play compared to the laborious slide-rule manipulations. Nomographs have usually been designed to incorporate the consistency and pitch corrections to the proper motor load that we have just done in several steps. Some propellers may have slightly different blade width ratios within their geometric series and therefore a different basic power number, all designed into the nomograph. However, the basic marine-form propeller does meet the constants we have used, and it is important to understand how these can be handled in the field.

What we have just covered is "horsepower response." In each example, the process horsepower requirement has been given. It cannot be stressed too strongly that horsepower response has nothing to do with process horsepower requirement. *It's only* the reaction to operating a particular diameter impeller at a particular speed in a particular fluid. Whether the expended energy is sufficient to perform the necessary action is an entirely different study, and that is the real "meat" of *The Selection of Agitators for Paper Pulp Slurries.*

Chapter 6:

Process Horsepower I

The last chapter concluded with the admonition not to confuse horsepower response with process horsepower requirement. Though this sounds simple enough, no single mistake is more often made, leading to extreme frustration by both the client and the vendor.

The classic case, apocryphal in nature *(17)*, is concerned with a little ½ horsepower portable mixer on a dye tank. The bag of additive dumped into the tank did not disperse as quickly as the client wished. His first complaint concluded with the comment, "I think I'll pull that ½ horsepower motor off and put a 2 horsepower motor on it." Assuming the 2 horsepower motor was the same speed as the smaller one removed and he did nothing to the propeller size, no change in dispersion would occur. However, he would have provided a very light load for a 2 horsepower motor. The hp response was related to the diameter of the propeller and the driven speed. The process hp requirement was greater than the original installed capacity, but he would have done nothing to change that. Fortunately, we caught that situation in time to make the proper correction in propeller diameter and absorbed horsepower.

The horsepower response of a particular agitator is related to the speed and diameter of a particular type of impeller. The agitator only absorbs the reaction to that speed and diameter. It doesn't know, or care, whether it is installed in a 1000-gal. tank or a 20,000-gal. stock chest. You might say that I become exacerbated over this unwarranted confusion, but unless we fully understand the difference between "response" and "requirement," we will always have difficulty in determining the correct action to be taken.

First Considerations

Chest shape: In the early days of my career, I was often fascinated by the various and strange shapes of stock chests. Especially in the northeastern part of the

United States and in Eastern Canada where I spent my "apprenticeship." There were old mills along the St. Lawrence River where the walls of some stock chests seemed to conform to the shoreline of the river. These were some of the last mills run by water power (Fig. 6-1). Some

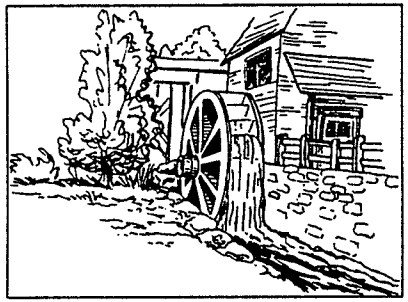


Figure 6-1. Mill by River with Water Turbines.

stock chests had "L" shapes and looked like an abandoned office complete with a "side-mounted wash room." More often than not, that's exactly what they were. How some of these chests were successfully agitated, I'll never know. Most likely, they weren't!" I remember a huge vertical cylindrical chest in the northwest, that was divided into pie-shaped quadrants. I tried to agitate it on two separate occasions and with two different suppliers.

Now that there is some common-sense engineering applied to chest design, I feel a certain nostalgia for the uniqueness and ingenuity of our forebearers. The "giants" of the era of midfeathers and vertical circulators mentioned earlier were "seven-league boots" ahead of what the founders of our industry had to contend with. The "Hurters" and "Whitesides" were quite specific in the length, width and height ratios required for good midfeather circulation. There was a "camaraderie" that existed between suppliers and clients, a shared responsibility that implied, "We're all in this together." But as a recent piece in the *Tappi Journal (18)* said, the "guinea pig (a mill willing to accept a piece of equipment on trial and report the results) is becoming an endangered species."

The design criteria for those original circulators had such qualifications as:

1. Single midfeather chest
   a. Channel width can be 1½ times the propeller diameter.
   b. Stock height can be 1½ times the propeller diameter.
   c. Maximum length can be 35-to-40 ft.
   d. Horsepower required can vary from 5 to 10 hp/1000 ft^3.
2. Vertical chest with multi-bladed circulator.
   a. Propeller swing diameter should be about 45% of chest diameter.
   b. Lower propeller should be 2 ft off bottom.
   c. Individual blades to be 2-to-3 ft apart..
   d. Three-blade clusters to be 4-to-5 ft apart.
   e. In low consistency, less than 4%, vertical shaft should be 1-to-2 ft off center so stock will ROTATE. In higher consistencies, shaft should be on center.
   f. Horsepower can vary from 8 to 30 hp/1000 ft^3 depending on size, usage and consistency.

Many other specific restrictions or requirements were exact ratios for corner fillets in midfeather chests, ratios of channel widths when multiple channels were used with a single circulator and maximum chest diameters for a particular propeller size. Most of these restrictions were arrived at empirically over many years of mill trials and remained as dictates for these types of agitators because the basic hydraulics and fluid mechanics of moving pulp slurries with a mechanical agitator hadn't been studied. When a particular midfeather installation showed great areas of stagnant stock, the vendor and the mill

agreed to increase horsepower or even install a second circulator. The particular "failure" and "fix" was then added to the growing list of empirical rules. I remember my first experience trying to agitate raw Bagasse and watching, to my horror, great islands of almost dry fiber circulating around the chest, not even being aware of the additional submerged "reefs" of thick stock that never moved at all. There was much to learn!

The first concentrated efforts to design stock chests for random circulation of paper pulp slurries were directed toward the simplest of vertical cylindrical containers. We already mentioned preliminary work with the success of the vertical turbine design, advanced to the vertical propeller and finally the horizontal-insert propeller unit. Even though these investigations were years before the OPEC crunch and our present concern with energy costs, the theme of the first trials was, "How can we handle the most stock with the least amount of horsepower?"

We were trying to prove to an entire industry that what they had been using for 150 years was wrong. There wasn't a long line of customers at the receptionist's desk, waving purchase orders for our new ideas, especially when it became apparent that even with optimum chest design, the cost of producing random motion and uniform consistency was considerably more than that required for midfeather or vertical circulator designs. This wasn't unexpected; after all we were doing more work to "produce a better product." The client had to be convinced that he needed this "better product," and he had to be willing to pay for it. We had a sales problem like trying to sell a deluxe Cadillac station wagon to a prospector who really wanted a four-wheel drive Jeep to get into the back country. How do you compete with a 50-hp agitator against a 25-hp circulator and build your credibility?

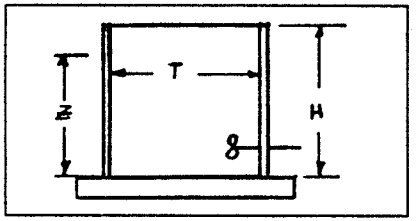


Figure 6-2. Vertical Cylindrical Chest.

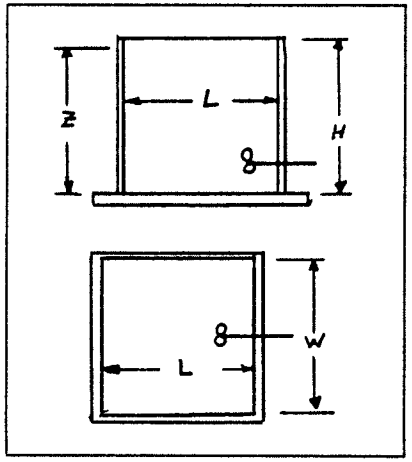


Figure 6-3. Square Chest.

Well, we let the sales department fight that battle while the engineers fought to provide the data. We borrowed the nomenclature from the process industries and, with tests to back up our conclusions, presented the ideal vertical chest, (Fig. 6-2).

These data proved that to attain our definition of compete motion, the stock level to chest diameter ratio should be 0.7 (Z/T = 0.7) for minimum horsepower. When plotted on log log paper, there was little premium paid for a Z/T of 0.8 (but this was an exception). The ideal dimensions for a vertical cylinder chest became a Z/T of 0.8, a stock level equal to 80% of the chest diameter *(8, 12, 20)*. The penalty in increased horsepower above this ratio greatly exceeded the increased volume attained. Once established, the company turned its

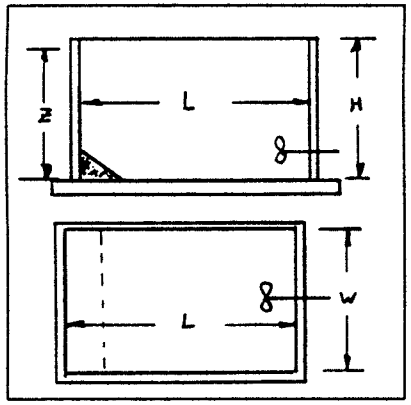


Figure 6-4. Rectangular Chest.

attention to rectangular chests. Why? Cylindrical chests are wasteful of space, but are also less expensive to construct. Hoop stress in cylindrical vessels allows thinner walls! But, again, if several rectangular chests can be nested together, that factor becomes negligible. To everyone's surprise, the optimum shape for a rectangular chest, for complete motion, was a CUBE! (Fig. 6-3)

With $L/W = 1.0$ and $Z/W = 1.0$, we could install even less horsepower for the same volume as in a vertical cylindrical chest with a Z/T of 0.8. But you can't build a free-standing chest of this design for even twice the money required for an ideal cylindrical chest of the same volume. So any thought of using rectangular chests must be tied to several in a row with common walls. Well, how far can we stretch these ratios to make this option attractive? Let's look at Fig. 6-4.

If we increase the length of the chest for a single agitator, we can go as far as an $L/W = 1.5$, but maintaining a $Z/W = 1.0$, without any great penalty in energy required. In fact, because of the geometry involved in some sizes, it might even come out to a little less energy requirement. We might be onto something here; nesting a bunch of these shaped chests could save a bundle! But there is a "bearcat hiding in the bushes" here. Dropping the level below that Z/W of 1.0 at first lowers the process requirement, but then increases it dramatically as the agitator begins to vortex while trying to drive the flow pattern to the end of the chest. At L/W ratios greater than 1.5, we will need to use two agitators as shown in Fig. 6-5.

The two units would be sized as though each is in a separate chest. We simply use $W/2$ as the "W" for each unit and design the chest for an L/W and Z/W of 1.0. A typical example for this type of chest is a medium width couch pit under the wet end of the paper machine.

All of these applications for chest design have assumed the need for complete motion to the normal stock level. What about the controlled-zone type applications?

These can be divided into two categories:

(1) Low-density storage, e.g., broke storage.
(2) High-density storage, e.g., pulp mill or broke.

Broke storage is a touchy subject with many paper machine superintendents. Many of them wish they had more than they do. On a full machine break, you can't use all of it back at the blend chest

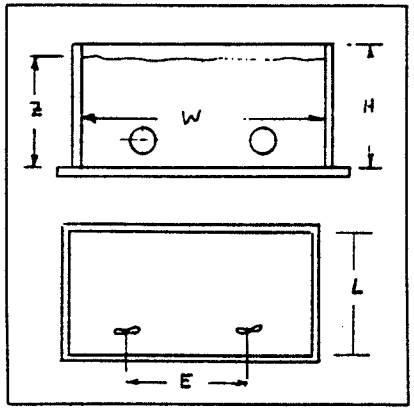


Figure 6-5. Rectangular Chest with two units.

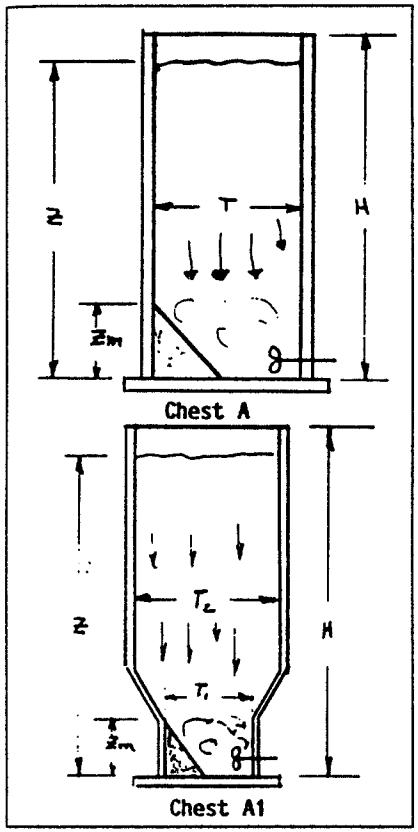


Figure 6-6. Controlled Zone Chests - Straight Shell & & Reduced bottom.

so most of it has to be held someplace else. At a production rate of 500 *tons/day* (21 *tons/hour*), only about 75 T/D (3 *tons/hour*) can be immediately reused. It doesn't take a wizard to know that the remainder, 18 *tons/hour*, will fill up most available chests pretty quickly. It has always been amusing to me, while trying to sell an under-machine broke pulper, to be asked, "Will your pulper take a break for 24 hours?" My answer is usually another question, "How much broke storage do you have?" When I hear the usual answer of one or two hours, I can truthfully say, "My pulper will handle the break as long as you have a place to put it." Some

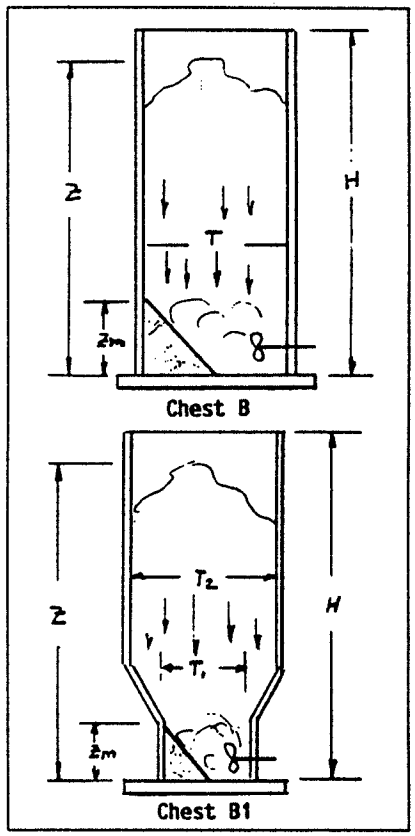


Figure 6-7. Hi-density Reduced Bottom Chest.

newer mills have provided amply for broke storage, but as we said earlier, making broke 24 hours a day is not really what we set out to do. Most of the mills I've had anything to do with feel well-endowed if they have two hours at full production, and many have much less than that. Of course, on prolonged breaks, one can always slow down the machine, cut the sheet back to the couch or—horrors—take the sheet off the wire!

But let's deal with a two-hour capacity and a machine production of 500 T/D. The broke chest would see a net production of about 18 *tons/hour* and therefore a chest of 36-40 tons capacity will do nicely. At an

average consistency of 3½% coming from the broke pulper, this will require a vertical cylindrical chest of 28 ft in diameter by about 58 ft high. This will allow a side-insert agitator to handle a zone 14 ft high across the full diameter of the chest. This isn't an optimum design for the agitator because of the extreme holdup time in the zone. A reduced bottom chest, even at low consistency throughout would be more appropriate. These two chests would be as shown in Fig. 6-6.

High-density storage of this broke might be a more economical option. A decker could be used to increase the consistency from the broke pulper to 12% for storage with the controlled zone operating at 4%, trimmed to a continuous 3½% at the pump suction. For 36-40 tons of storage, these chests would be as shown in Fig. 6-7.

What are the criteria for these "controlled zone" chests? Tests conducted in "see-through" pilot vessels, determined that the power necessary to create complete motion at a Z/T of 0.7 was not altered as the level increased. The level at which motion ceased simply decreased, finally stopping at a Z/T of 0.5. Regardless of the overall level, this initial horsepower input maintained complete motion at a level equal to 50% of the chest diameter. However, on tests that included continuous feed and withdrawal (at equal rates) it was found that the system was unstable at a total level of up to 150% of the diameter, Z/T = 1.5. The zone would "upchuck"—turnover, with the unagitated zone exchanging places with the agitated zone. Above a Z/T of 1.5, the system became stable. Because of the possibility of changing levels, especially in a broke chest, a safety factor was added to the minimum level, and it was decided that a controlled-zone chest would have a Z/T of at least 2.0 based on the zone diameter in a straight-shell chest or 3.0 based on T1 in a reduced-bottom chest.

Variations from these rules deal with retention time in the zone (next subject), the ratio between T1 and T2 in a reduced-bottom chest (not to normally exceed 1.8:1.0) and the economics of construction. This latter point might be embellished a bit. Tile chests can be built in almost any diameter of integral feet. If a chest was built of stainless steel sheet, we might consider the labor of cutting, number of welds and drop offs. Sheets are usually available in 20- and 40-ft lengths. If you have a π symbol on your calculator, simple division will give you some cost-effective diameters, e.g., 20 ft—6 ft—4 in. i.d.; 40 ft—12 ft—8 in. i.d. Perhaps not a major factor in the overall cost and in the common grades of stainless steels, but it is a viable consideration in a chemical plant and using pure nickel for tank material.

Summary of optimum chest shapes

1. Complete motion
   a. Vertical cylindrical chests
      Z/T = 0.7-0.8
   b. Rectangular chests
      L/W = 1.0-1.5
      Z/W = 1.0
2. Controlled-zone storage chests
   a. Straight-shell vertical cylindrical chests
      Z/T = or 2.0
      Z_m/T = 0.5
   b. Reduced-bottom vertical cylindrical chests
      T2 = T1 × (1.6-1.8)
      Zm = 0.5 × T1
      $Z/T1$ = or > 3.0

N.B.[1] Bottom fillets are shown in all controlled-zone vertical cylindrical chests. These are usually 45-degree fillets intersecting the center line of the bottom of the chest and are normal to the center line of the agitator shaft. Other types of fillets can be used, but this style is the simplest to construct. The purpose is two fold: (1) It ensures no bottom fillet of dead stock at the far wall, and (2) it reduces the energy requirement for zone agitation by a significant amount while reducing the volume only marginally.

[1] N.B. *Nota Bene*, Note Well!

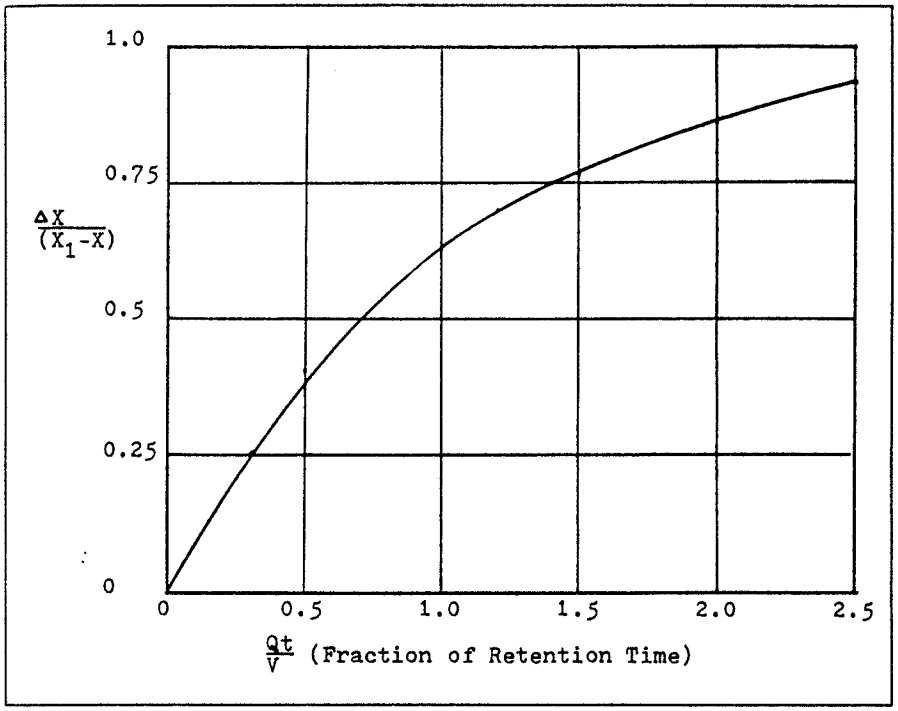


Figure 6-8. Retention - Qt/V vs. X/(X1-X).

Retention Time

Because we can now produce a random pattern of agitation that ensures a blending action on any upsets coming to the chest, it is possible to calculate a minimum retention time to affect control over a known pattern of upsets.

Figure 6-8 can be used in two ways: (1) To determine the change in a variable, "t", time after a change in the feed and (2) to calculate the volume required to control a variable within desired limits from a known cycle of feed variations. In this graph, the abscissa, Qt/V, represents the fraction of retention time where Q = flow rate in gal/min, t = time in minutes and V = agitated volume in gallons. (Other consistent units could be used.) The ordinate, $\Delta X/(X1-X)$, represents the change in the variable after time t, (ΔX), divided by the difference between the value of the variable in the incoming feed, (X1), and the average value of the variable prior to the upset, (X).

Let's examine a particular example: A mill has installed a new machine chest with an agitator designed for complete agitation. The chest contains 60,000 gal. of stock at 4% consistency, and the throughput is 2084 gal/min. (A fairly standard 10-ton capacity machine chest for 500 T/D, retention 29 minutes.) Now we know that the cyclic variations coming to the chest are never as neat and regular as we are going to propose, but to illustrate the use

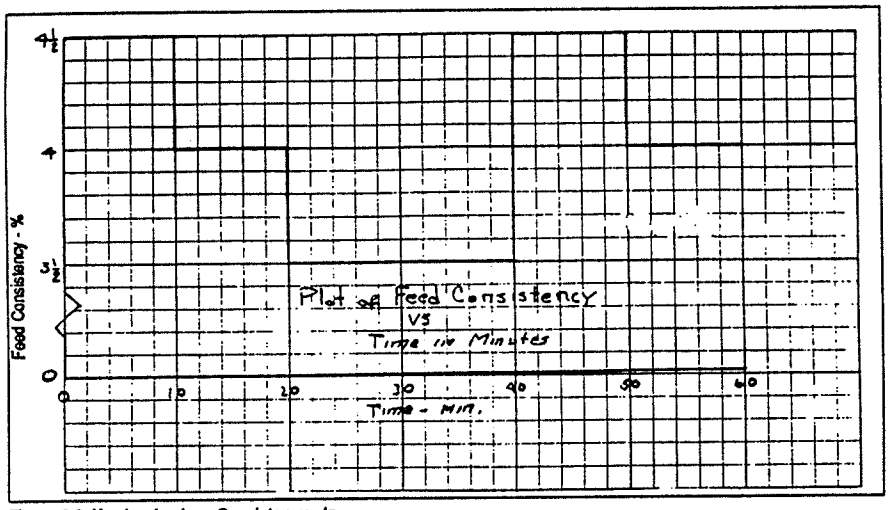


Figure 6-9. Varying feed vs. Consistency - In.

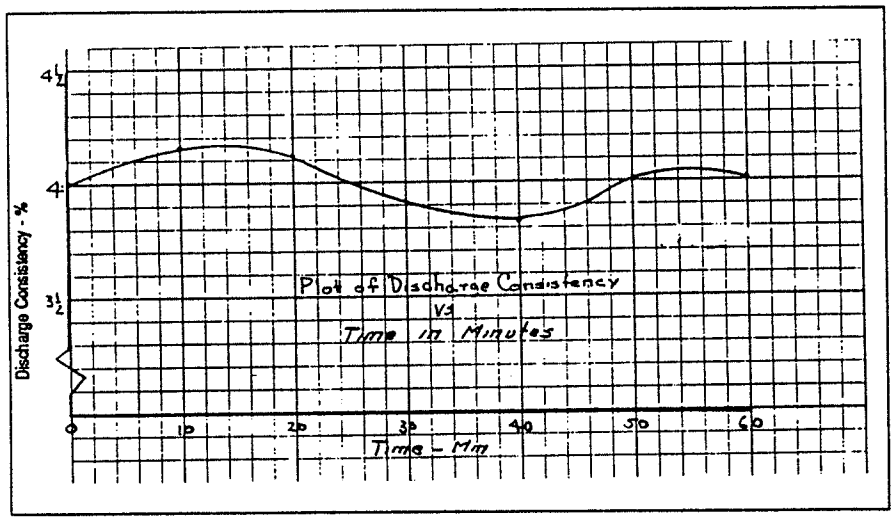


Figure 6-10. Varying feed vs. Consistency - Out.

of the graph, Fig. 6-8, let us assume the following for a typical one-hour period:

Initial average consistency at time zero is 4%. The feed to the chest now begins to change in the following manner:

Time	Feed
0	4.5%
10	4.0%
20	3.5%
30	3.5%
40	4.5%
50	4.0%

Figure 6-9 shows these data plotted,with time on the abscissa. What will the discharge consistency look like when calculated and plotted on the same scale?

1. At time 0 to time 10

$Q = 2084$ gal/min $X_1 = 4.5\%$
$V = 60{,}000$ gal $X = 4.0\%$
$t = 10$ min $X = ?$

$QV = 2084 \times 10/60{,}000 = 0.35$
from Fig. 6-8, $X/(X_1 - X) = 0.29$

$X = 0.145\%$ At time 10, $X = 4.145\%$.

2. At time 10 to time 20

$Q = 2084$ gal/min $X1 = 4.0\%$
$V = 60{,}000$ gal $X = 4.145\%$

ΔX will now $= -0.04\%$
and at time 20, $X = 4.105\%$.

By continuing this calculation and remembering to change the value of X each time by adding or subtracting ΔX, we see that:

3. At time 20 to 40 (same feed for 20 minutes)
   $\Delta X = -0.303\%$ and X at t = 40 will be 3.802%.
4. At time 40 to 50
   $\Delta X = 0.202\%$ and X at t = 50 will be 4.004%.
5. At time 50 to 60
   $\Delta X = -0.001\%$ and X at t = 60 will be 4.003%.

Figure 6-10 shows how abrupt changes in consistency are gradually incorporated into the full volume. At no time does the

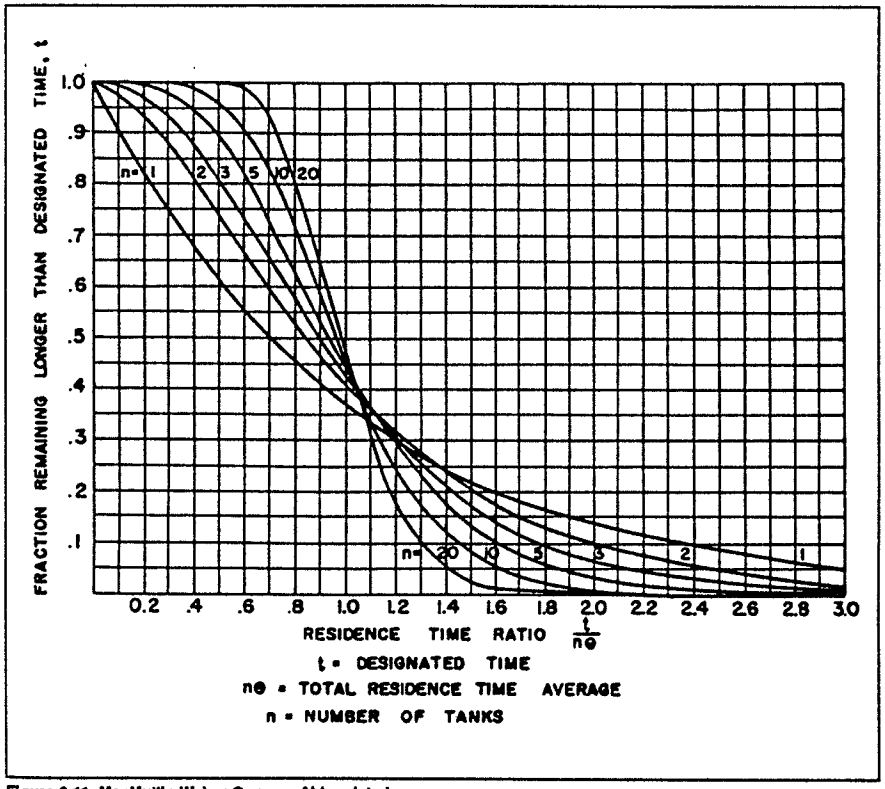


Figure 6-11. MacMullin-Weber Curves - Abbreviated.

deviation from the average 4.0% consistency vary more than 0.2.

This is certainly within the capability of a consistency controller to smoothly operate the trim dilution valve which follows this chest. If the feed variations are much greater or for longer periods of time, we might want to make a reverse calculation, assuming a maximum deviation we can tolerate. This will lead us to a larger chest for more hold-up time or a serious investigation of what is happening upstream.

Another example of the use of these data has to do with freeness variations in a groundwood storage chest. An existing mill *(4)* used these data to determine the size of the bottom zone in a controlled-zone storage chest. The goal was to limit freeness variations to 5 points assuming a maximum change in the feed of ± 10 points for one hour. The volume settled upon was 160,900 gal, and the throughput was 2000 gpm. The average freeness was to be 90. Let's see how close that calculation was:

Q = 2000 gal/min	X = 90
V = 160,900 gal	X1 = 90 ±10
t= 60 min	ΔX = ?

$$QV = 2000 \times 60/160{,}900 = 0.75.$$

From Fig. 6-8, $\Delta X/(X_1 - X) = 0.53$

$$\Delta X = \pm 5.3.$$

But suppose the variation of ± 10 were to normally last about two hours. How large must that volume be to maintain a variation of ± 5?

$$\Delta X/(X_1 - X) = 0.5 \ \& \ QV = 0.70$$
$$V = 2000 \times 120/0.7 = 342{,}900 \text{ gal.}$$

These examples show what retention time does to a specific system. We haven't discussed the effect of horsepower on a fixed system. The minimum amount of horsepower that will produce complete random motion in a particular system, will provide complete mixing in a batch system in some specific time period, t. If we were to increase the horsepower input beyond this level, we would shorten "t" time by some value, Δt, not directly but by some exponential function of horsepower. But we are always concerned with a continuous system, and one must understand that in a single vessel, the residence time of material going into and out of the vessel continuously is statistical. Some portion of that flow will exit immediately and some portion will theoretically stay in the vessel forever. Figure 6-11 is a partial presentation of the classic MacMullin-Weber data *(14)* which allows us to calculate the minimum residence time for a percentage of the flow for a parameter of numbers of vessels. This shows the effect of using multiple vessels in series on the percentage of material staying in contact for the designated time or longer. Since we are concerned with one vessel (stock chest) at a time, we will use n = 1 for these examples.

Using the nomenclature listed under Fig. 6-11, let's imagine we are feeding a crowd and want to heat a big kettle of soup continuously, drawing off a volume of hot soup at the same rate we are adding a cold mixture. We had a trial run before the crowd gathered and discovered it took 10 minutes to heat that volume on a batch basis. Therefore "t" time is equal to 10 minutes. Now suppose we draw off and refill at such a rate as to give an average retention time of 10 minutes; θ = 10 min. Since we only have one soup kettle:

$$n = 1$$
$$1/n\,\theta = 10/1 \times 10 = 1.$$

Now read up the graph from 1.0 on the abscissa to the intersection with the curve for n = 1 and then read the ordinate value of 0.36. This means that only 36% of the cold soup will be heated for the designated time of 10 minutes or longer. Lots

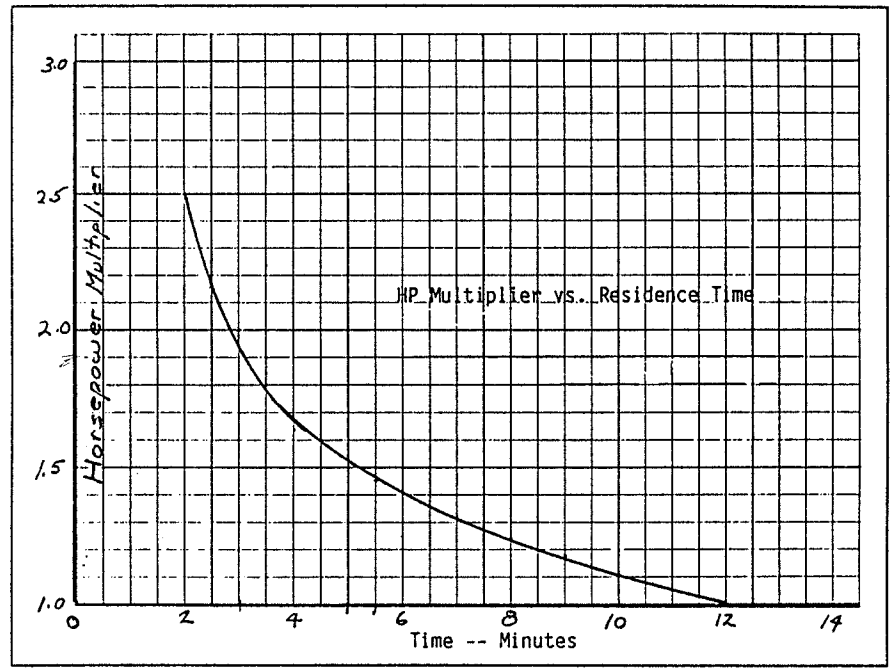


Figure 6-12. hp vs. Retention Time.

of lukewarm soup being ladled out! We must decide on some reasonable percentage of the soup to be heated for 10 minutes or longer. Let's say 80%, that should still be edible. Reading from the ordinate of 0.8 over to n = 1 and then down to the abscissa, we find:

$$1/n\,\theta = 0.22 \text{ or } \theta = 10/1 \times .22$$
$$= 45.5 \text{ min.}$$

This means we either must have a much bigger kettle to give 45.5 minutes average retention time or settle for a much lower flow rate and a lot longer "lunch hour."

We use the same approach in designing the minimum hold-up time in a stock chest, whether for a controlled-zone storage chest or a flash tank for rapid additive blending. The best example is the controlled-zone dilution under a high-density storage chest. We want to have the smallest volume commensurate with accurate dilution to keep the agitator size within reason.

It has been shown that on a batch basis, complete agitation will effect complete blending of some addition in 2-to-4 minutes, depending on consistency. If we assume 3 minutes for the usual range of consistency, 3½ to 4½ %, and we pick 15 minutes for the minimum average retention time allowed in the dilution zone we have:

$$1/n\,\theta = 3/1 \times 15 = 0.2.$$

From Fig. 6-11, 0.2 on the abscissa gives us 0.82 on the ordinate or 82% of the high-density stock, and dilution water will stay in the zone for three minutes or longer. In practice, most suppliers set 12 minutes as the minimum average retention time, which by the same calculation would give us 78% of the feed retained three minutes or longer. This has proved to be quite

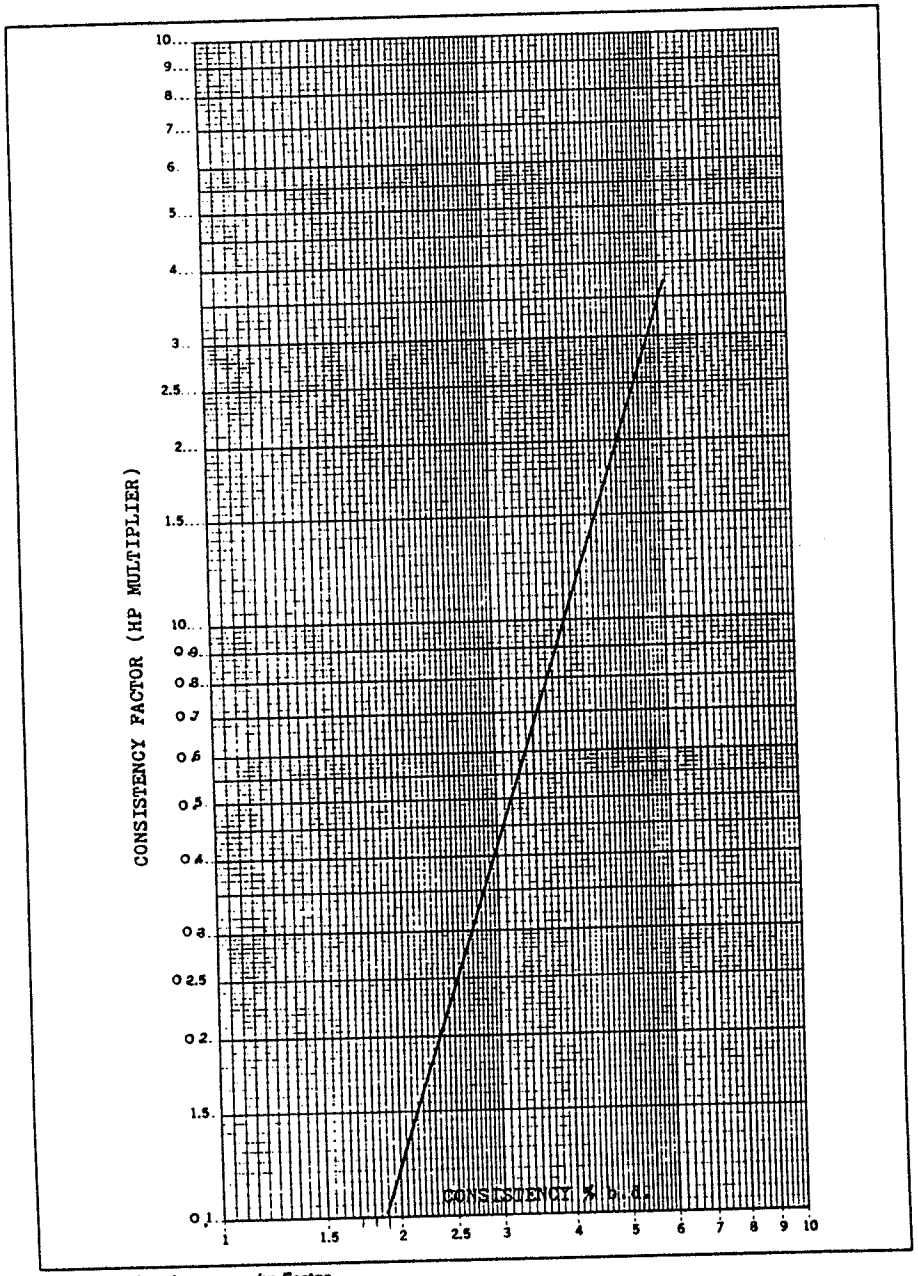


Figure 6-13. Consistency vs. hp Factor.

satisfactory. The action of the pump on the discharge, plus the trim dilution has provided an outfall through the consistency controller that has allowed a smooth control of consistency to the next downstream processing unit. Since, generally, the control system around a high-density storage chest has allowed a percentage of recycle to the zone, this has certainly improved the operation during minimum retention times.

But suppose, due to geometry and economics, we end up with less than 12-minutes retention in the zone. Can additional horsepower correct this shortage? Yes, within limits. Figure 6-12 shows a plot of residence time vs. a horsepower multiplier. We use this with great care, for it "pinch hits" for lesser residence times. It isn't desirable on pulp mill high-density storage if it can be avoided. Classic examples of its required use are on down-flow bleach towers where residence times may be only 2-to-3 minutes in the agitated zone.

Consistency

We have hinted at the effect of consistency on process power requirement several times in previous chapters. Consistency—pounds of fiber per pound of slurry—is expressed as a percent. The paper industry has given birth to many capricious and whimsical standards, but the concept of "consistency" is perhaps the one most subject to individual interpretation. Many terms are used, often with little regard to an exact value, which the modern designer of agitators is so dependent upon bone dry, b.d; air dry, a.d.; machine dry, m.d.—what do they mean? Bone dry is clearly the only finite value—paper samples weighed in a controlled atmosphere immediately after being dried to a constant weight in a laboratory oven (also called oven dry, o.d). Air dry can only have a finite meaning if one knows the exact relative humidity of the room in which the sample comes to a constant weight or a.d. Machine dry is even more whimsical which is the condition of the sheet off the calender stack going on to the reel. Depending on the grade being made, coated on one or both sides, uncoated, filled, machine-glazed (MG), tissue or whatever, machine dry can be anything from almost 0% moisture to as high as 8%. This moisture content is subject to variation by the humidity of the machine room, the permeability of the wrapper after slitting into "sets" and how soon the rolls are shipped.

An agitation system designer is vitally concerned with the exact consistency for a particular installation because of the exponential change in horsepower required with slight changes in consistency. It's common in the U.S. industry to use a.d as meaning 10% moisture, though there are exceptions. Some mills specifically identify a.d. on their flow sheets as meaning 5% moisture. Our counterparts in Europe and in Canada, use bone dry as a standard on flow sheets. We do have to be careful with our English friends, they have been known to measure distances in barley corns!

Consistency affects the process horsepower requirement by the cube of the change in consistency, $hp \alpha C^3$. Figure 6-13 represents this relationship with the unity factor taken at 4% b.d. The confusion between client and supplier of even that 10% difference in a.d. and moisture-free can make a remarkable change in unit selection. Suppose a supplier is using 5% consistency as the stock condition in a particular chest and makes the assumption, or is led to believe, that 5% is a moisture-free value. Going through all the design criteria, he arrives at a selection of a 100 hp unit, 80 hp at the impeller as you will recall from the loading procedures in Chapter 5. Later, after the award is given to a competitor, he discovers the client really meant 5% a.d. What happened?

5% a.d. x 0.9 = 4.5 moisture-free.
$[4.5/5]^3 x 80 = 58.3$ HP Actual hp required.

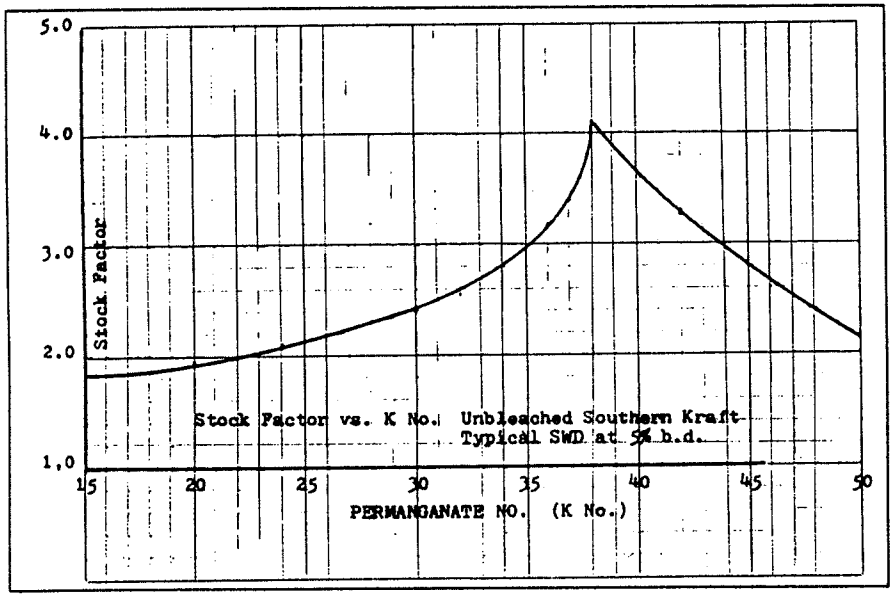


Figure 6-14. K No. vs. Stock Factor.

This allowed the successful bidder, who obviously questioned the client more closely about the consistency, to design to 60-impeller hp and recommend a 75-hp unit which was lower in both energy and capital cost.

Of course it works the other way, too. A 20-hp unit, 16-impeller hp, was installed, based on the assumption that the flow sheet called for 4% a.d. In reality, the flow sheet balanced at 4% b.d. The resultant agitation was unsatisfactory at the design stock level, and only by reducing the level by some 16% was the client able to obtain complete agitation, a level which seriously altered the retention time in the chest. What happened?

$[4/3.6]^3 \times 16 = 21.9$ hp Actual impeller hp required.

The supplier should have recommended a lightly loaded 30-hp unit or at the very least, a heavily loaded 25-hp unit.

We know that many mills find this type of accuracy difficult to determine. The original balanced flow sheet is often at odds with eventual practice. But this leaves the supplier with an enigma. Knowing the disastrous effect of slight differences in consistency, he has some hard choices to make. As the University of Texas football coach once said, "With a forward pass, only three things can happen, and two of them are disasters." He can assume the worst case, b.d. consistency, and lose the order to someone who used the lesser value. He can assume a.d. consistency and take the order but perhaps have a borderline installation reflecting on his credibility. Or, he can take the time to dig into the possibilities with the client and show him the alternatives for either assumption. Operating consistency is important to the optimum selection of agitators for your installation. You should understand the pitfalls that might await you.

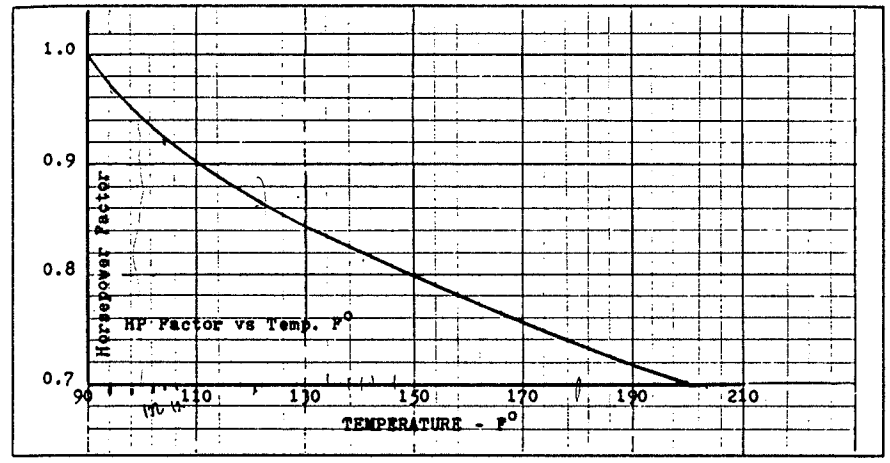


Figure 6-15. Temperature vs. hp Factor.

Stock Type

As we mentioned in Chapter 3, "All cats aren't black," and had the first serious investigation of pulp and paper agitation not taken place in close proximity to "a large photographic company in upstate New York," perhaps the comparison to other types of stock might have been entirely different. If that "curious" company *(19, 20)* had been situated in Northern Louisiana, the standard pulp for comparison might have been southern unbleached kraft cooked to a 38 permanganate number! Nothing could have been tougher to handle, and all other stock factors would have been less than one, compared to this difficult pulp. I'm sure the company wouldn't have been discouraged, but it might have been years before we discovered the more easily agitated varieties such as bleachable kraft, bleached kraft and "good old" low-yield softwood sulfite.

As it was, the first pulp tested was super clean, low-yield bleached softwood sulfite used in the manufacture of fine photographic papers. Perhaps this was a blessing and these neophytes should have had as much encouragement as possible. But, the "hammer" would drop later when we were fully committed to the research program. As other companies began to follow the lead of these first "intrusions" into the sacred realm of midfeathers and vertical circulators, everyone settled on bleached low-yield sulfite as the basic pulp with stock factor 1.0 at all consistencies. The first suggestion that "all cats weren't black" probably came from experiments with groundwood pulp. That's nearly 40 years ago, and my memories are fading a bit. But I know that Canadian mills make a lot of groundwood for newsprint, and the Canadian subsidiary of this company had much to do with keeping our noses to a "grinding wheel." Groundwood pulp acts similar to sulfite until reaching a consistency of about 3%. Then it requires additional process horsepower, referred to sulfite, at an alarming rate as consistency further increases. The difference between these requirements we called "stock factor." As we got further into the industry, we absorbed additional expensive lessons and gradually developed a complete picture of stock factor related to consistency, type of pulp, wood source and a plethora of factors based on the cooking requirements for various unbleached softwood krafts-permanganate numbers. Figure 6-14

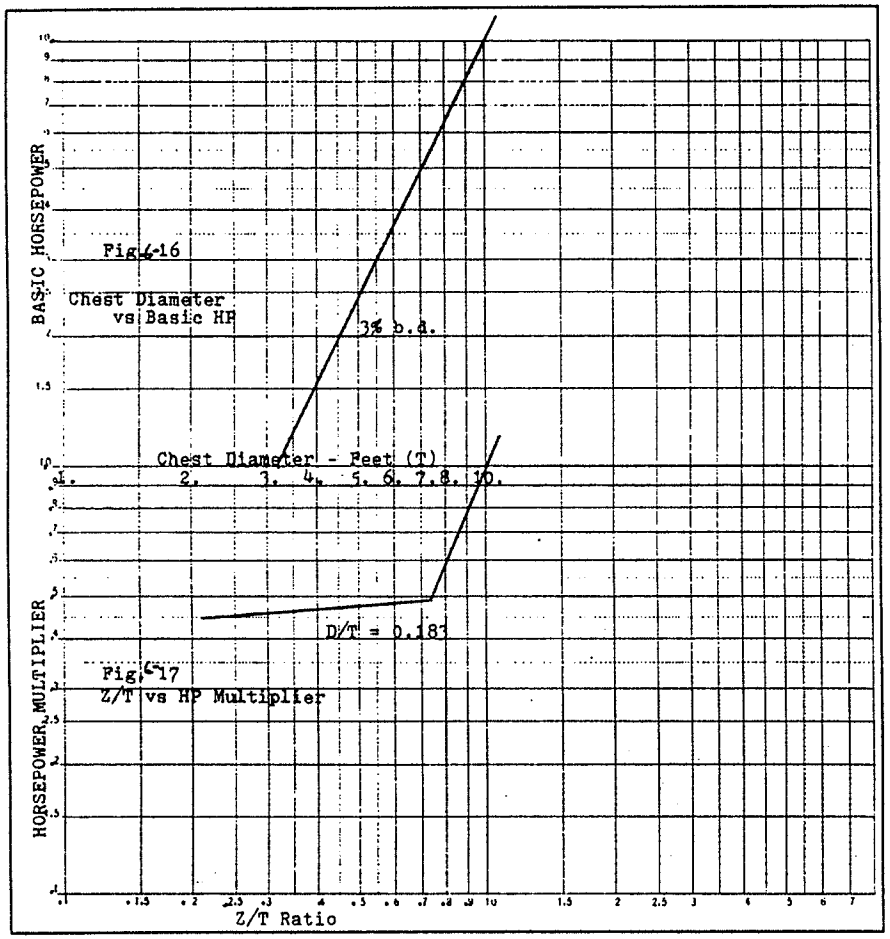


Figure 6-16. Chest Diameter (T) vs. Basic hp @ 3%.
Figure 6-17. Z/T vs. hp Factor @ D/T = .183.

displays an example of stock factor vs. permanganate number for just one level of consistency. We were relieved to discover that hardwood pulps, regardless of cooking process, always exhibited stock factors equal to or less than the standard sulfite pulp. We were dismayed to learn that pulp made from Douglas fir and Slash Pine exhibited extreme stock factors.

All pulps, due to wood properties, cooking procedures, grinding and various methods of fiber treatment, exhibit various freeness and reactions to the shear tester. In all these years, no clear correlation has been found between freeness and stock factor or between stock factor and shear values using the shear tester which has been used to further explain pipe friction values. Rigorous study of the data presented by MacKenzie, Manteufel et al. *(23, 24, 25)* hasn't provided any breakthrough in this area. We know that short fibers, e.g., hwd, contribute to low-stock factors and long fibers, such as Douglas fir, contribute to high-stock factors but beyond that, neither freeness nor shear value are particularly indicative of the degree of divergence from the standard sulfite pulp

A display of stock factors by stock type will be presented in the next chapter.

Temperature

The last of the "first considerations" has to do with the effect of temperature on the agitation of paper pulp. As we all have experienced, the hotter the stock slurry, the easier it flows. This is due to the reduced viscosity or pseudo-viscosity of the slurry. We could make some extremely tedious calculations concerning the effect of temperature on the viscosity of water and the resultant change in flow properties, but since we are dealing with a pseudo-viscosity affected by temperature and by consistency and shear rate, I'd rather leave that to someone's doctoral thesis. This is intended to be a practical approach unburdened by anything you can't see on a graph or do on a pocket calculator. Another consideration we won't cover in these pages is the effect of pH. A pulp with a pH on the caustic side is easier to move than one on the acid side. I use a "seat of the pants" factor or more often expect a little safety margin in dealing with a "basic" pulp. Figure 6-15 presents a horsepower multiplier on the ordinate for an abscissa of increasing temperature. Use this with great care! Sometimes, especially in large chests, it is tempting to look for any factor that will reduce what is obviously going to be a high horsepower requirement. "He said the stock would be hot, maybe almost 140° F." You might look longingly at that 0.82 factor for 140° F but better think about it! Will the stock really be that hot? Was that maybe what the client hopes it will be and will settle for 100° F or 120° F? Or maybe 140° F is the incoming feed stock temperature and the other flow, dilution water or broke, is at ambient making the blend chest a lot less. When taking any horsepower multiplier that reduces the unit selection for a particular chest, be absolutely certain those conditions will be met. It's a lot easier to smile at a little excessive agitation than something less than what is needed!

Now we are ready to look at the earliest methods of sizing a horizontal agitator for the concept of complete random motion. We have observed the effects of chest shape, retention time, consistency, stock type, temperature and D/T; now let's put these factors into some logical sequence in order to select the proper agitator. The classic method, still satisfactory but cumbersome, is to take the dimensions and process definition of some known standard and then "throw on" the multipliers to account for particular conditions. For example:

Figures 6-16 and 6-17 give partial plots of chest diameter vs. basic horsepower for 3% consistency and Z/T vs. horsepower multiplier for a D/T of 0.183 respectively. These are shown specifically to illustrate a typical case.

Assume a 10-ft diameter chest, a stock level of 10 ft (Z), a 22-in diameter propeller and 3% b.d. consistency bleached sulfite stock. The various input values are then:

T = 10 ft, Z = 10 ft, Z/T = 1.0, D = $22/12$, D/T = 0.183, C = 3%
and the stock factor = 1.0.

Using a retention time greater than 12 minutes and 90° F for the stock temperature, Figs. 6-12 and 6-15 both give factors of 1.0.

Basic hp from Fig. 6-16 @ T =
10 ft = 10 hp
hp multiplier from
Fig. 6-17 @ Z/T = 1.0 = 1.0
Impeller hp required then becomes:

Ihp = 10 x 1 x 1 x 1 x 1 = 10 Ihp required.

A parameter of consistency lines could be added to Fig. 6-16 using the relationship shown in Fig. 6-13 (Cons. vs hp factor) and a similar parameter of lines could be drawn in Fig. 6-17 knowing that the hp multiplier is nearly inversely proportional to D/T. For example, if the consistency was 4% instead of 3%, all other factors remaining the same, the required impeller hp would become:

$[4/3]^3 = 23.7$ Ihp.

If in addition to the consistency change, we also increased the propeller diameter to 30 in., $D/T = .25$, the required Ihp would become:

$\text{Ihp} = 10 \times [4/3]^3 \times [0.183/0.25] = 17.35$ Ihp.

The type of stock affects any of these solutions in direct ratio to the stock factor for that pulp at that consistency. Also, notice the abrupt change in the hp multiplier on Fig. 6-17 as the Z/T reaches over 0.7. These data, taken from laboratory and field observations, confirm the ideal shape for vertical cylindrical chests as being a Z/T of 0.7 to 0.8.

Now let's consider a larger chest and discover why this classic "textbook" method is cumbersome and not as direct as the one described in the next chapter.

Chest:	30-ft diam. x 27 ft hi.
Stock level	24 ft $Z/T = 0.8$, vol. = 126,900 gal
Throughput:	400 T/D = 2223 gpm
Retention time	57 min
Furnish:	Northern SWD GWD @ 3% b.d. and 90° F

Rather than complete the parameters on Fig. 6-16 and 6-17 for a method we will shortly discard, let me restate the two relationships we will require to solve this problem:

Basic hp will vary as the square of the chest diameter; $\text{hp} \alpha T^2$.

HP multiplier vs. stock level ratio (Z/T) will vary inversely with D/T; $\text{hp} \text{f} \alpha \frac{1}{D/T}$.

At 3% b.d. consistency, the stock factor for GWD is 1.0.

From the previous example, basic hp at 3% and 10-ft diam. was read as 10.0 (Fig. 6-16).

Since $\text{hp} \alpha T^2$, basic hp at 30-ft diam. will be:

hp = 10 x (30/10)2

hp = 90.

a. Assuming a $D/T = 0.183$ (66-in. diam. propeller) and a $Z/T = 0.8$, read the hp multiplier from Fig. 17 as 0.59. Since S.F. = 1.0 and the temperature factor = 1.0, the process hp required will be:

hp = 90 x 1.0 x 0.59 x 1.0 x 1.0

hp = 53.1.

From Chapter 5 on power response and loading, we recognize that we must use a 75-hp drive for this selection. A 60-hp drive would be too tightly loaded assuming normal efficiency and a 10% safety factor for hydraulic surges.

But is this the ideal selection? To determine this, we must consider at least two other selections. Consider the requirements for a 54-in. propeller $D/T = .15$) and a 72 in. propeller $D/T = .2$). Remember, $\text{hp} \text{f} \alpha \frac{1}{D/T}$.

b. 54 in. hp = 90 x 1.0 x 0.59 x (.183/.15) x 1.0 x 1.0

hp = 64.8.

This isn't very good because; a 75-hp motor would be too tightly loaded and therefore a 100-hp drive would be required.

c. 72 in. HP = 90 x 1.0 x 0.59 x 0.183/0.2 x 1.0 x 1.0

hp = 48.6.

Here we are at almost exactly 80% of a 60-hp motor and a 60-hp drive would be satisfactory.

Now what do we have? Usually these calculations are done on a preprinted form with spaces for three or four selections, but we can tabulate the results as follows:

a. 75-hp drive, 66-in propeller
b. 100-hp drive, 54-in. propeller
c. 60-hp drive, 72-in. propeller.

A choice might be made on energy cost and capital cost with more weight given to one or the other depending on project budget, power costs and payback. The only obvious capital cost item is the 54-in. propeller that would be less costly than a 66-in., or than a 72-in., etc. Be careful in judging unit and drive costs. The 100-hp drive might be the least expensive because of the higher speed (lower torque) of the 54-in. propeller. The 75-hp unit, depending on design, might require a larger shaft and bearing assembly. The 60-hp unit would definitely require a larger shaft and probably a speed reducer rather than a less expensive V-belt drive. All of these factors must be considered to arrive at an optimum selection, and any cost data written here would be out of date before you got this book to your office. Another factor we have not even considered is how would that 72-in. propeller perform if the chest were routinely cycled from normal level to as low as 8 ft. two or three times per day? Someone may well ask, why didn't we look at a 60-in. propeller? Try it! The requirement will come out to be 58.3 hp, still requiring a 75-hp drive, but possibly a viable choice over the 66-in. unit.

This was the method used for many years to solve agitation problems, once the concept of random motion was accepted and understood. Other plots were used to account for the different configurations encountered with square and rectangular chests. Other factors were applied to the basic data for controlled-zone and high-density applications. This was considered a pretty straightforward "cookbook" approach with little understanding by the "calculator" of just what the various trials meant.

It wasn't until the late 60s and early 70s that another method was devised that truly defined the output of the agitator impeller. That method we shall review in Chapter 7.

Chapter 7:

Process Horsepower II

In the late sixties, a small Canadian manufacturer of Mixers & Agitators, Prochem Ltd., became intensely interested in the true discharge profile and capacity of mixing impellers. A paper by Cooper and Wolf *(26)* opened the door for a more complete investigation of the radial-flow turbine impeller and the axial-flow propeller as applied to the agitation of paper pulp slurries. From this study, a new method of selecting agitators was born which directly related process requirement to impeller performance. A superior, proprietary impeller called the "Maxflo" was also developed, but the concept of impeller performance was immediately relatable to all types of axial-flow impellers. We shall follow the line of standard three-bladed marine-form propellers for the presentations that follow.

It has been shown previously that any impeller rotating in a fluid absorbs horsepower by producing some ratio of flow (Q) and head (H), $hp \alpha QH$. For any impeller, operating freely in a vessel unrestricted by stator bars or a draft tube, we understand that the volumetric discharge is at its minimum at the impeller while the fluid velocity is at its maximum at the same point. However, as the fluid stream begins to move away from the impeller, the boundary layer between the fast moving fluid and the stagnant fluid in the vessel begins to entrain additional flow. At the same time, the velocity of the stream begins to decay. As the flow increases, velocity decreases in the same ratio and so these investigators created the concept of the "Conservation of Momentum" and how to use it in describing the capacity of a specific impeller. Earlier work by Fox and Gex and described by Gray *(11)* confirmed this concept in the equation:

$$Mo = \frac{\rho (ND^{2})^{2}}{g_c}$$

For water-like materials, the equation can be reduced to a proportionality:

$$Mo \alpha N^2 D^4$$

which can be derived in another manner by combining the simple proportionalities of flow (Q) and velocity (V):

$$Q \alpha ND^3$$
$$V \alpha ND$$
$$QV \alpha N^2 D^4.$$

These investigators chose to call their momentum number by its two variables, Q and V, or simply "Que Vee." We shall use "momentum number" or Mo.

In order to determine the momentum number for a particular impeller, a constant relating to its efficiency had to be determined. The equation

$$Mo = CN^2 D^4$$

is useless until a value of "C" can be established. The basic equation for "QV" is as follows:

$$QV = \frac{Chp^{.8}}{N_p^{.8} N^{.4} (\frac{\rho}{g})^{.8}}$$

This equation is of interest because it is quickly seen that the efficiency factor is simply:

$$Eff = C/N_p^{.8}$$

This has been shown to be 1.1 for a square-pitch marine-form propeller, which decreases rapidly at higher pitches and increases at a similar rate at lower pitches. It's this increase in efficiency at pitch ratios less than 1.0 that was of great interest during the development of the Prochem "Maxflo" impeller. Although only slightly more efficient in the usual range of pitches possible with a marine-form propeller, 14 to 22 degrees, the proprietary Maxflo impeller is able to operate at extremely low pitch settings, even 0-degree pitch, with efficiencies that greatly exceed any style axial-flow propeller. However, this dissertation isn't to extol the virtues of any particular supplier, and we shall confine our discussion to the marine-form propeller which is available to any supplier or user.

Let us go back to that basic QV equation and derive it from equations we have already discussed:

1. Basic Power Number

$$N_p = \frac{C \times hp \times g}{\rho \times N^3 \times D^5} \quad (1)$$

2. Basic Flow

$$Q = \alpha ND^3 \quad (2)$$

3. Velocity

$$V \alpha ND \quad (3)$$

4. Momentum

$$QV \alpha N^2 D^4 \quad (4)$$

5. Substitute (4) in (1)

Solve for $QV = \frac{Chp}{N_p ND \rho/g}$

$$N_p = \frac{C \times hp \times g}{\rho \times QV \times ND} \quad (5)$$

6. Solve (4) for D

$$D = \frac{QV^{.25}}{N^{.5}} \quad (6)$$

7. Substitute (6) in (5)

Rearrange

$$QV = \frac{C \times hp}{N_p \times N \times \rho/g \times \frac{QV^{.25}}{N^{.5}}} \quad (7)$$

8. Divide by exponent 1.25

$$QV = \frac{C \times hp^{.8}}{N_p^{.8} \times N^{.4} \times (\rho/g)^{.8}} \quad (8)$$

Now let's look at this equation more closely to see what is meant by efficiency. The term ρ/g is common to any impeller and falls out when we correct for all units. The term HP/N^4 is also not a measure of efficiency. This leaves only the term $C/N_p^{.8}$ which is specific to one type of impeller and is a true measure of its efficiency. So that we will have something to work with later, let's look at Table 7-1 which displays all necessary data to calculate the momentum number for any size of marine form propeller at any pitch setting.

TABLE 7-1. Pitch - NP - C.

Pitch (Degrees)	Ratio	N_p	$N_p^{.8}$	$C/N_p^{.8}$	C
30	1.50				
22	1.24	.446	.524	0.99	0.519
21	1.18	.425	.504	1.01	0.509
20	1.12	.403	.483	1.04	0.502
19	1.06	.382	.463	1.06	0.491
18	1.00	.360	.442	1.10	0.486
17	0.94	.338	.420	1.12	0.470
16	0.88	.317	.400	1.15	0.460
15	0.82	.295	.377	1.20	0.452
14	0.77	.277	.358	1.23	0.440

Now we can use the equation, $Mo = CN^2 D^4$ to solve for the momentum number of any size propeller at any speed.

Example: A 36-in. propeller at 18-degree pitch running at 260 rpm.

Since the units of Mo are ft^4/s^2:
N = 260/60 rps
D = 3 ft
$Mo = 0.486 \times (260/60)^2 \times 3^4$
Mo = 739.

Let's calculate how much more efficient that propeller would be if we reduced the pitch to 14 degrees and increased the speed to consume the same amount of hp.

At 260 rpm and 18 degrees, the power response in 4% stock would be:

$$hp = \frac{.36 \times N^3 \times D^5}{283.8}$$

hp = 25.1.

The speed for that HP at 14-degree pitch, Np = .277, would be:

$$N = \sqrt[3]{\frac{25.1 \times 283.8}{.277 \times 3^5}}$$

N = 4.73 rps or 284 rpm.

At Np equal to 0.277, C will equal 0.44 (Table 7-1), therefore:

$$Mo = .44 \times 4.73^2 \times 3^4$$
Mo = 797.

An increase in Mo of 7.9% and a decrease in torque required from 6,082 in. lb to 5,568 in. lb or 8 ½%. If we considered another manufacturer's propeller at a high pitch of 22 degrees, with the same propeller at 14-degree pitch, the difference would be even more striking. Any time we can operate at a lower pitch and higher speed for equal or better process results, i.e., higher Mo number, we will have combined efficiency with lower torque and a less costly drive assembly.

The momentum number for a given process result can be a very useful tool. Later, we will see how it can be used for scale-up, velocity calculations and short-cut methods for selecting standard applications such as high-density towers, couch pits and white water chests.

But first, let's see how the basic requirements for any chest configuration and process specification are determined. We will still need a few graphs relating diameter or width, consistency, stock level, and retention time to obtain a final process num-

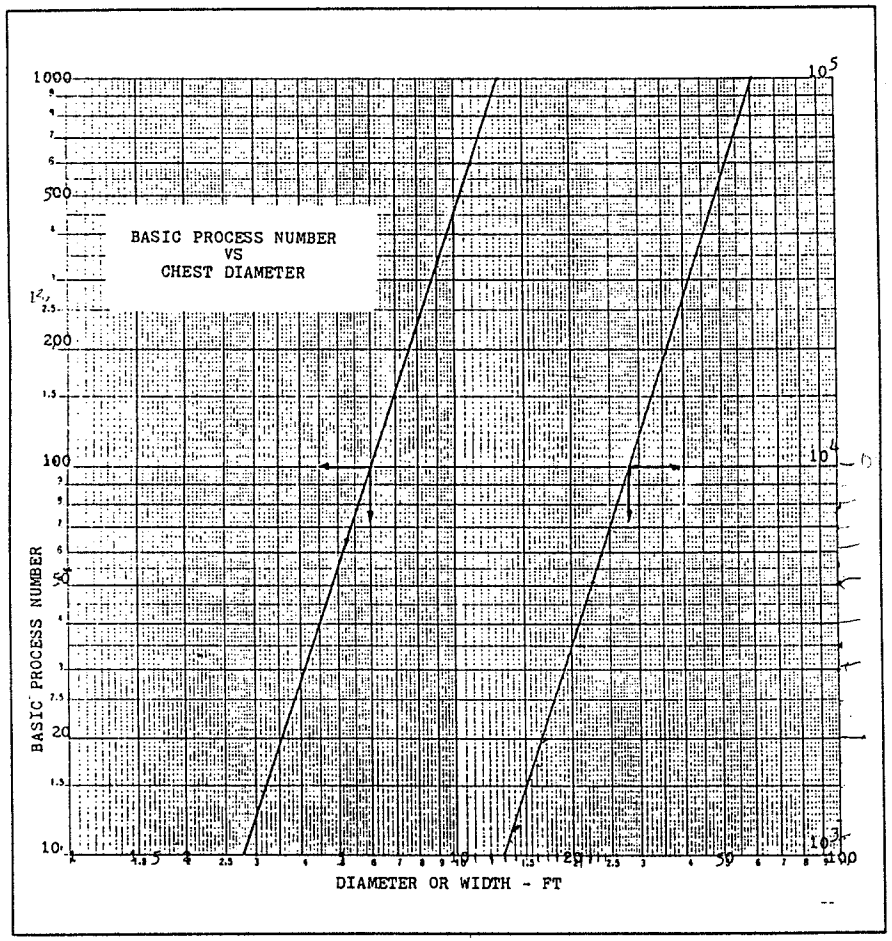


Figure 7-1. Process Number vs. T(W).

ber, and a conversion plot to convert process number to momentum number. However, won't be concerned with D/T and will make one calculation to obtain the process requirement. A table of precalculated momentum numbers for the standard speeds, propeller diameters and horsepowers will allow you to pick the optimum unit size directly.

Well, let's get started! Figures 7-1 through 7-8 will be our working tools. Before doing some specific examples, we need to understand what each of these graphs and the table (Fig. 7-3) accomplishes:

Figure 7-1 is a straight line plot on 2 cycle log log paper relating chest diameter or width on the abscissa to a basic process number on the ordinate. You will notice two lines, one relating diameter up to 13 ft to the left-hand ordinate and the second line continuing from 13 ft to 66 ft relating to the right-hand ordinate. (eliminates a less readable 4 x 2 cycle log plot, if anyone is curious.)

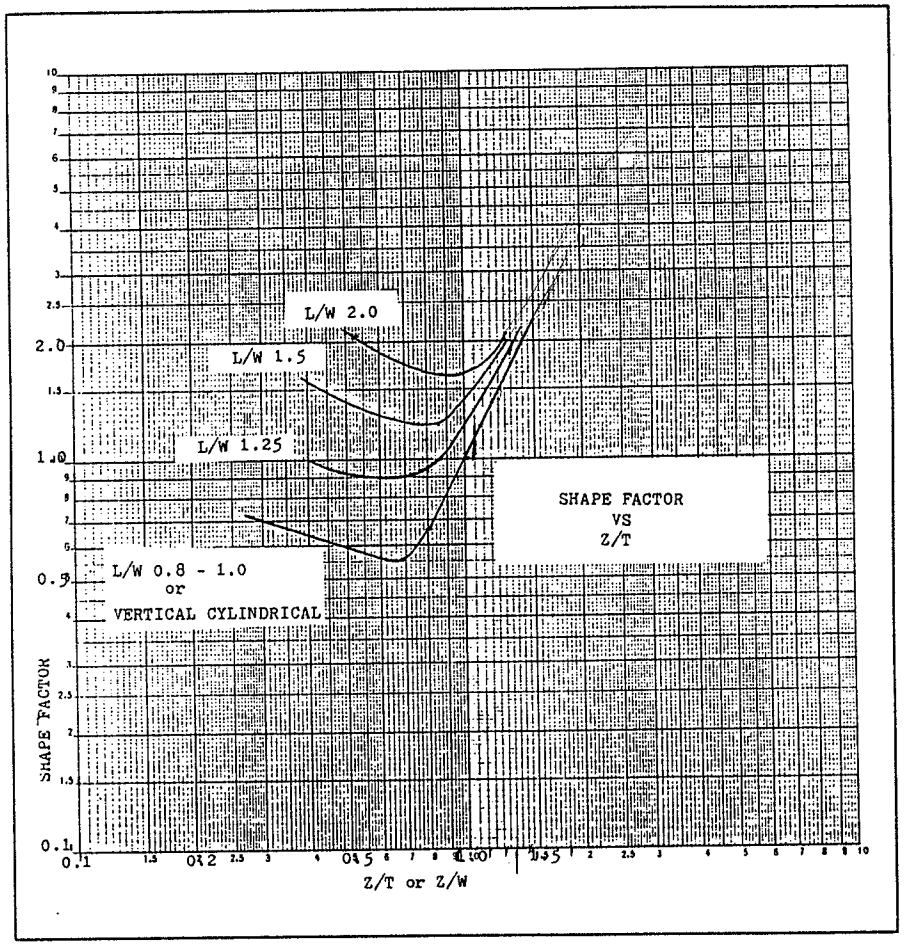


Figure 7-2. Shape Factor vs. Z/T (Z/W).

Entering the plot with any diameter or width, in feet, and reading up to the appropriate intersection and then to the indicated ordinate, gives the basic process number for the diameter, e.g., 15-ft diameter, read 1500 for process number.

Figure 7-2 is a plot relating stock level ratio, Z/T or Z/W, to a multiplier called "shape factor." It doesn't matter whether you are dealing with a cylindrical or rectangular chest, for the parameters for length to width ratio, L/W, are plotted with the lower curve which covers a cylindrical chest or a rectangular chest with L/W of 0.8 to 1.0.

Example: A chest 15-ft diameter with a 12-ft stock level. Enter Fig. 7-2 with a Z/T = 0.8, read up to the first curve and read shape factor = 0.66.

Figure 7-3 is really a table of stock factors relating different types of stock, and treatment, to a range of consistencies.

	CONSISTENCY % B.D.								
	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0
SULPHITE - SWD									
Low Yield - Bleached	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
Low Yield - Unbl. Unref.	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8
Hi-Yield - Bleached, Ref.	1.3	1.5	1.7	2.0	2.5	2.8	3.0	3.2	3.5
Hi-Yield Chips + 90%	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
Hi-Field Chips + 65%	0.35	0.35	0.35	0.35	0.35	0.35	0.35	0.35	0.35
KRAFT - SWD K# = Permanganate No.									
K# 16-24 Unbl. Unref.	1.4	1.6	1.7	1.8	2.0	2.1	2.2	2.3	2.5
K# 24-30 Unbl. Unref.	1.6	1.8	1.9	2.1	2.2	2.3	2.4	2.5	2.6
K# 30 Unbl. Ref. & Semi-Bl. Unref.	1.1	1.3	1.4	1.5	1.7	1.8	1.9	2.0	2.1
K# 32 Unbl. Unref.	1.7	1.8	2.0	2.2	2.3	2.5	2.6	2.7	2.8
K# 38 Unbl. Unref.	1.8	1.9	2.3	2.8	3.2	3.6	4.1	4.5	5.0
Bleached & Semi-Bl. Ref.	1.0	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7
50% Douglas Fir Bl. Unr.	1.6	1.7	1.8	1.9	2.0	2.1	2.2	2.3	2.4
0-5 cc CSF Ref. Condenser	1.33								
GROUNDWOOD - SWD									
To 90°F	1.0	1.0	1.0	1.2	1.5	1.8	1.9	2.0	2.1
Virgin Newsprint to 90°F	1.0	1.0	1.0	1.1	1.3	1.4	1.5	1.6	1.7
Newsprint Broke to 90°F	1.0	1.0	1.0	1.1	1.2	1.3	1.4	1.5	1.6
BAGASSE									
Bl. Unref. 50% Yield	.64	.65	.66	.67	.67	.68	.69	.69	.70
Bl. Unref. 70% Yield	.36	.40	.44	.48	.53	.56	.59	.62	.65
Fully Ref.	1.0								
MISCELLANEOUS STOCKS									
Mixed Waste Paper	1.0	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7
Rag Stock Cooked & Scr.	1.0	1.1	1.2	1.5	1.7	2.0	2.1	2.2	2.3
NSSC - SWD	1.0	1.0	1.1	1.2	1.2	1.4	1.5	1.6	1.7
Masonite - Unref.	1.0	1.0	1.1	1.2	1.3	1.5	2.0	2.1	2.2
Uncooked Chips	0.1								
All HWD Pulps	1.0								

Figure 7-3. Stock Factors - Table.

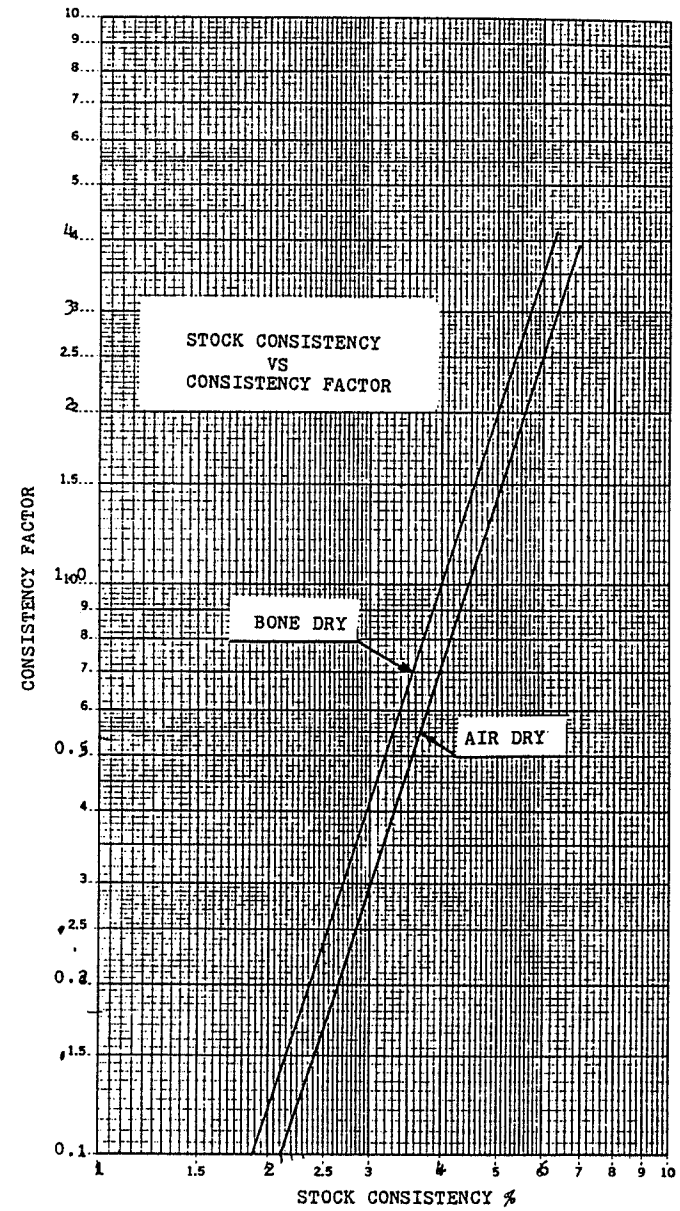


Figure 7-4. Consistency vs. Process Factor.

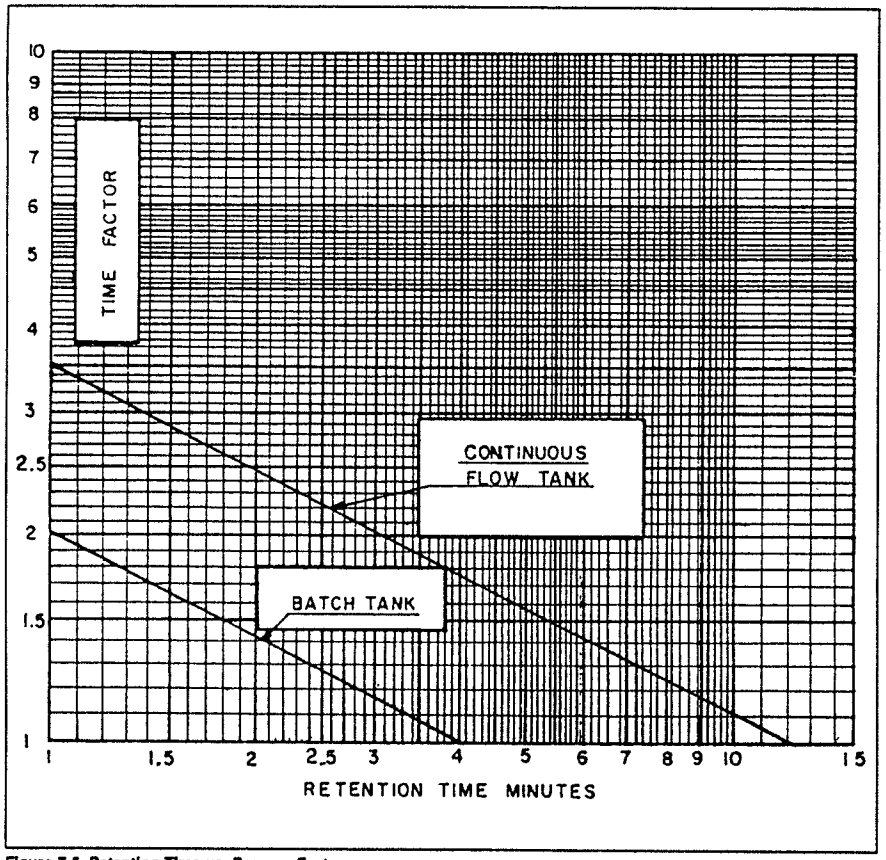


Figure 7-5. Retention Time vs. Process Factor.

Example: Unbleached, unrefined SWD kraft, permanganate no. = 16 - 24 at 4.5% moisture-free consistency, stock factor = 2.1.

Figure 7-4 is a straight line plot on 2 × 1 cycle log log paper relating stock consistency on the abscissa to a consistency factor on ordinate. The line labeled "Bone Dry or (b.d.)" represents the basic relationship. the line labeled "Air Dry" is only for convenience and assumes 10% moisture (a.d.% = 0.9 x b.d.%).

Figure 7-5 is another straight line plot on 1 x 1.5 cycle log log paper relating retention time to a time factor or multiplier. As discussed previously, retention time in a continuous chest is more critical than in a batch chest. The line labeled "Continuous Flow" shows that an average retention time of less than 12 minutes requires a multiplier greater than 1.0, while a batch blend chest can have as low as four minutes retention before requiring correction.

Figure 7-6 plots the conversion of the corrected process number on the abscissa to the required momentum number on the ordinate. To avoid the use of a cumbersome 2 x 4 cycle log log sheet, a second line is drawn using the same ordinate but

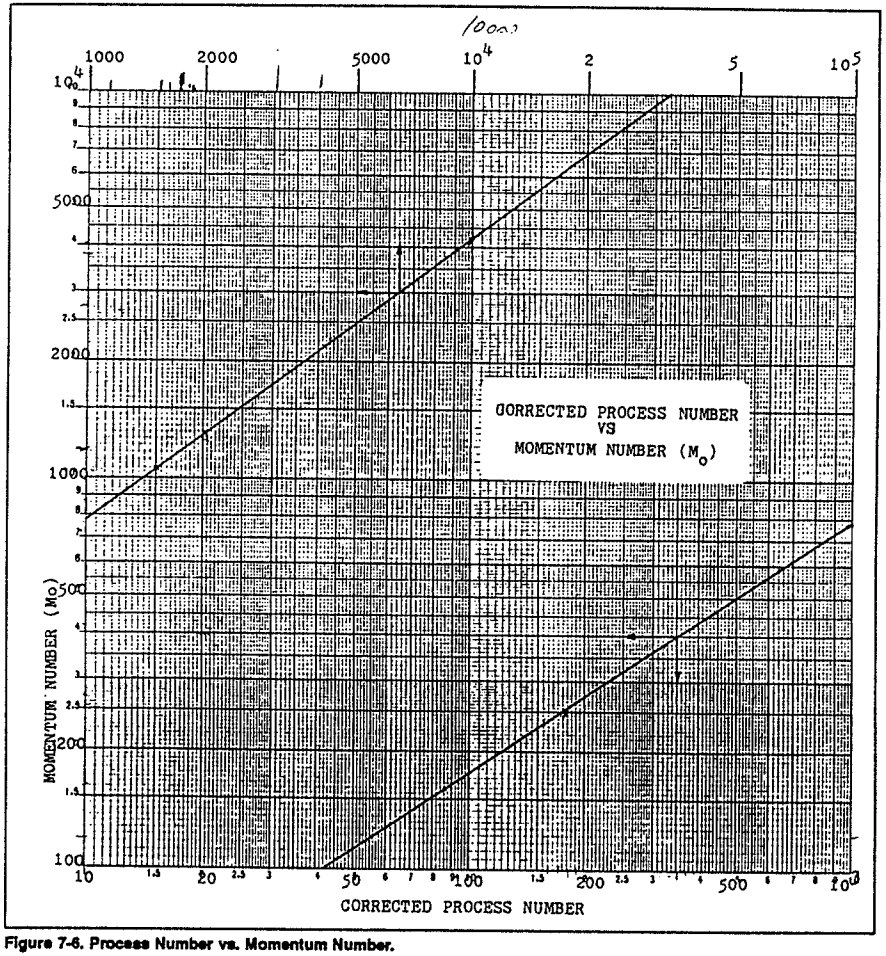


Figure 7-6. Process Number vs. Momentum Number.

continuing the abscissa at the top of the plot.

Example: With a corrected process number (CPN) of 840, enter the plot on the lower abscissa and read a momentum number (Mo) of 700 on the ordinate. Similarly, a CPN of 5000 will yield a Mo of 2500 by using the upper scale and reading to the same ordinate.

Figures 7-7 and 7-8 must be used together and only after a particular unit size has been selected. The table of unit selections (Table 7-2) with their corresponding values of Mo, are all calculated at 4% b.d. consistency and pitch setting for 80% of the indicated motor horsepower. Quite frequently, the particular problem you have solved will involve a consistency greater or less than 4%, and a pitch change will be required to absorb the full 80% of motor horsepower. When the pitch is changed from that shown on the unit selection table, the Mo capacity will also change and the true capacity at this new

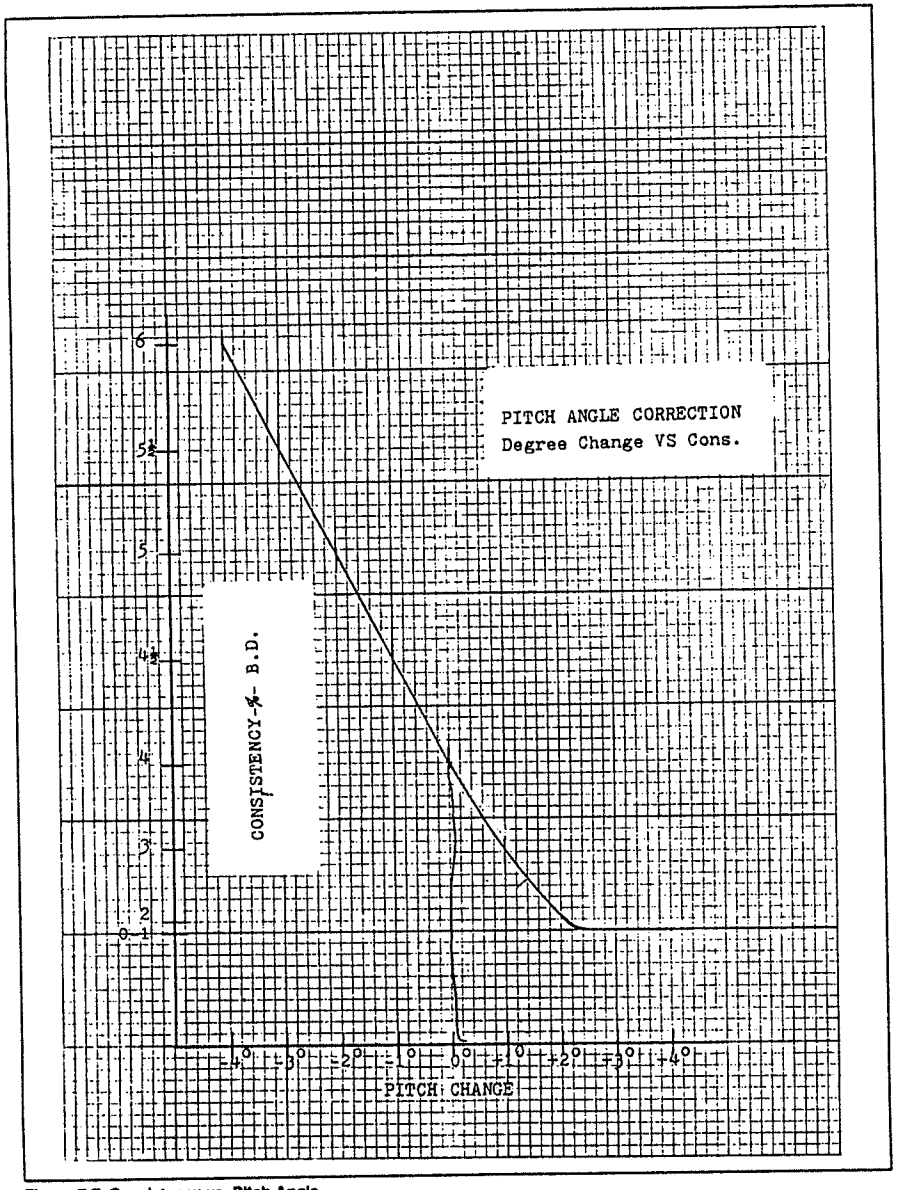


Figure 7-7. Consistency vs. Pitch Angle.

setting must be calculated and then evaluated for the possibility of less than optimum process performance.

Figure 7-7 plots b.d. consistency on the ordinate against a pitch change in degrees on the abscissa. Notice that at 4%, the

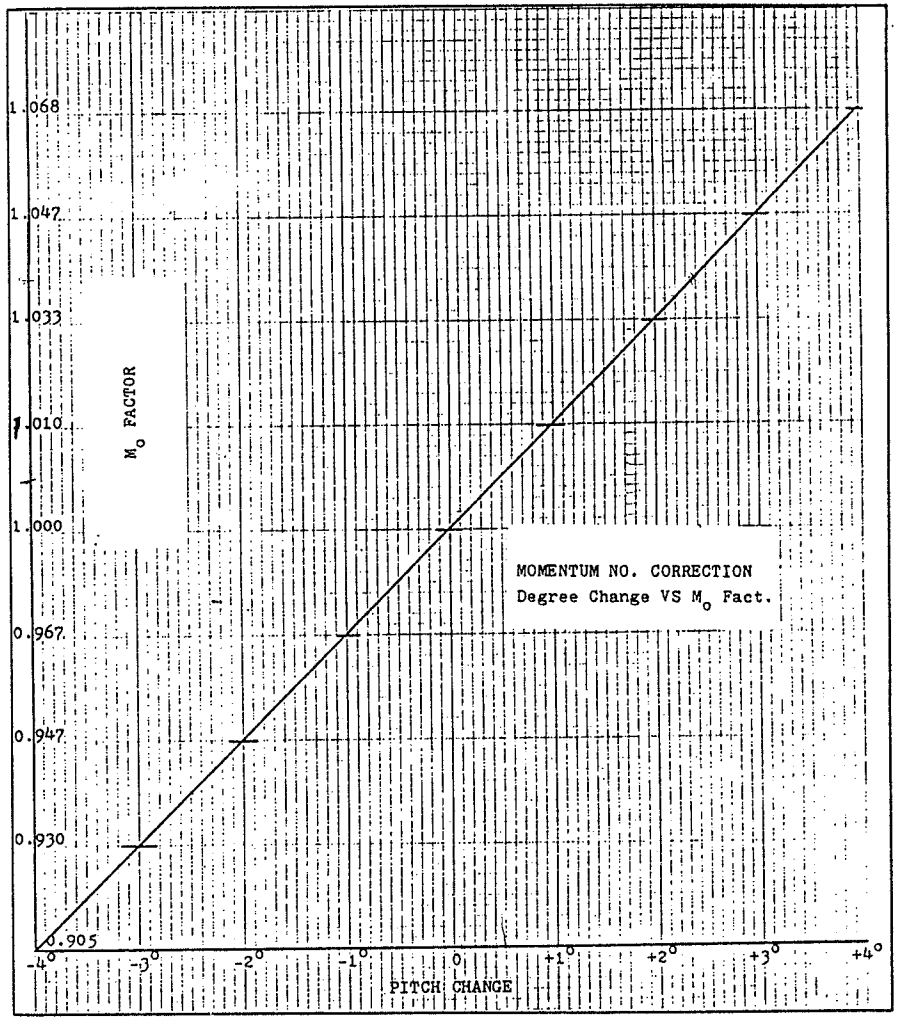


Figure 7-8. Pitch Angle Change vs. Momentum Factor.

pitch change will be 0 degrees, confirming the prearranged calculation. If the consistency in the chest is 4½%, the unit will absorb more horsepower at its standard setting. Using this plot, we see that the pitch angle must be decreased by 1 degree. At 3%, the pitch will require an increase of 1 degree.

Figure 7-8 gives us the change in Mo capacity caused by the change from standard pitch. Notice again that at 0 degrees pitch change, the Mo factor is 1.0, confirming again the standard selection. However, given that -1 degree change for 4½% consistency, the standard Mo will be decreased by the multiplier 0.967. With a

	PROPELLER			MOTOR		
Drive	Dia.	RPM	Pitch	HP	RPM	M_o
3V	18	413	F.P.	5	1775	117
"	24	272	"	5	"	160
"	26	240	FIXED	5	"	171
"	24	304	" PITCH	7½	"	200
"	26	272	"	7½	"	220
"	28	249	" Np	7½	"	248
"	24	333	"	10	"	240
"	26	309	"	10	"	284
"	28	276	" 0.35	10	"	305
"	24	394	"	15	"	335
"	26	342	"	15	"	348
"	28	294	"	15	"	346
"	30	279	"	15	"	410
"	24	424	"	20	"	388
"	26	374	" 0.41	20	"	416
"	30	302	"	20	1170	481
"	36	225	"	20	"	554
"	26	394	"	25	1775	462
"	30	334	"	25	"	588
"	36	249	"	25	"	678
"	26	424	"	30	"	535
"	30	352	"	30	"	653
"	36	260	" 0.36	30	1170	739
"	30	373	"	40	1775	734
"	36	279	"	40	"	851
5V	42	219	18°	40	1170	971
3V	30	422	F.P.	50	1775	939
5V	36	302	F.P.	50	1170	997
"	42	232	19°	50	"	1102
"	36	326	F.P.	60	"	1162
"	42	247	19°	60	"	1249
"	48	209	16°	60	"	1429
"	42	275	17°	75	"	1482
"	48	218	18°	75	"	1598
"	54	185	16°	75	"	1793
"	42	299	18° 0.37	100	1775	1811
5V	48	239	18°	100	1170	1974
"	54	197	17°	100	"	2078
"	42	338	15°	125	"	2152
"	48	253	18°	125	"	2212
"	54	215	17°	125	"	2475
Gear	60	170	20°	125	1775	2519
"	48	280	16°	150	"	2565
"	54	230	16°	150	"	2772
"	60	190	17°	150	"	2946
"	66	155	19°	150	"	2998
"	72	140	17°	150	"	3316
"	54	255	17°	200	"	3481
"	60	210	17°	200	"	3598
"	66	170	19°	200	"	3607
"	72	155	17° 0.36	200	"	4065

Table 7-2. Propeller/hp Selections vs. Momentum Number.

+1 degree change, the Mo will be increased by the factor 1.010.

Example: A final unit selection has been made having an Mo capacity of 1575. The calculated requirement was 1460, and we were quite pleased with what appears to be a comfortable excess. But the consistency for which we calculated the requirement of 1460 was 5 ½%. Are we still safe? Figure 7-7 at 5 ½% tells us we must decrease the standard pitch by 3 degrees. Figure 7-8 at -3 degrees tells us the Mo factor is 0.930. 1575 x 0.930 = 1465! This is still higher than the calculated requirement, and our final selection was correct with the indicated pitch change.

N.B.We can get into a situation where the decrease or increase in pitch is beyond the limits of that particular selection, i.e., a standard selection may already be at 15 degrees, and anything greater than a minus 1 degree change will place the pitch below the practical limit of 14 degrees for a standard propeller. We will cover this event in later examples.

Now we're ready for specific examples and to use Table 7-2 to make the final selection. These data are arranged in ascending order of propeller diameter and motor horsepower. Notice the three or four diameters associated with each motor size giving several different values of momentum number, Mo. All propellers 36-in. diameter and below are standard three-blade fixed-pitch marine-form propellers. Those 42-in. and larger are adjustable pitch with the blade angle tabulated.

N.B. All selections shown are for normal loading in 4% b.d. furnish. For other consistencies, the loading must be checked. Because of the limitation of standard V-belt ratios, many of the fixed-pitch selections are loaded slightly above or below the desired 80% of motor rating. Operating speeds shown are calculated for the most convenient V-belt selections, (3V and 5V) through 125 hp with the single exception of the 60-inch propeller at 125 hp which would require a gear-driven speed reducer. All selections above 125 hp require speed reducers with standard AGMA ratios (some with the standard optional ratios). V-belt driven speeds are based on full load speeds of 1775 rpm for 1800-rpm motors and 1170 rpm for 1200-rpm motors.

Typical examples of the momentum number method:

1. Machine chest

Furnish—100% SWD Unbl. kraft, 30K
Production rate—500 $^{tons}/_{day}$ @ 3½% b.d. and 120° F
Chest—25-ft diam. x 20-ft stock level (Z)

500 T/D @ 3½ % b.d. = 2381 $^{gal}/_{min}$
Vol. @ 20 ft Z = 74,438 gal
Retention time = 31 min
Z/T = 0.8

Fig. 7-1 = 7000
Fig. 7-2 = 0.66
Fig. 7-3 = 1.5
Fig. 7-4 = 0.66
Fig. 7-5 = 1.0
Fig. 6-15 = 0.87 (from Chapter 6)
CPN = 3979 (1 x 2 x 3 x etc.)
Fig. 7-6 = 2100 Mo required.
Fig. 7-7 = +0.5 degrees (pitch angle change for 3 ½%).
Fig. 7-8 = 1.005 (Mo multiplier)

Table 7-2

54 in. @ 100 hp = 2078 x 1.005 = 2088
42 in. @ 125 hp = 2152 x 1.005 = 2163

Choose the 100-hp unit with 54-in. propeller. Final Mo = +99% of requirement and saves 25 hp.

2. Pulper dump chest

Furnish—Bl. SWD kraft bales
Production rate—To accommodate 1 ½ dumps @ 6% a.d. from a 3000-lb furnish

pulper. Dilute to 5% a.d. while holding for 45 min, 90°F

Chest—12 ft diam. x 14 ft stock level (Z)

1 ½ dumps @ 3000 lbs = 4500 lbs @ 5% a.d. (4.5% b.d.)

= 12,000 gal

Z/T = 1.17

Fig. 7-1 =810

Fig. 7-2 =1.4

Fig. 7-3 =1.4

Fig. 7-4 =1.42

Fig. 7-5 =1.0

CPN = 2,254

Fig. 7-6 = 1400 Mo required

Fig. 7-7 = -1 degree (pitch change @ 4.5%)

Fig. 7-8 = 0.967 (change in Mo by pitch change)

Table 7-2 48 in. @ 60 HP = 1429 × .967 = 1382

42 in. @ 75 HP = 1482 × .967 = 1433

This is a cycling chest and low level just before the second dump favors the 42-in. propeller. It would be the mill's choice if the 15-hp savings was critical.

3. Blend chest

Furnish—virgin newsprint

Production rate—250 T/D @ 4% b.d. and 120° F

Chest—10 ft wide, 13 ft long, 10 ft stock level (Z)

250 T/D @ 4% b.d. = 1,042 gpm

Vol @ 10 ft Z = 9,724 gal

Retention time = 9.3 min

L/W = 1.3

Z/W = 1.0

Fig. 7-1 = 455

Fig. 7-2 = 1.25

Fig. 7-3 = 1.3

Fig. 7-4 = 1.0

Fig. 7-5 = 1.15

Fig. 6-15 = 0.87

CPN = 740

Fig. 7-6 = 640 Mo required

Fig. 7-7 = 0 degree change

Fig. 7-8 = 1.0

Table 7-2 36 in. @ 25 hp = 678

30 in. @ 30 hp = 653

Choose the 25-hp unit with the 36-in. propeller.

These three examples are typical of many mill stock chests, though relatively simple. In all but one case, these were contrived to be ideal dimensions for the duty required. You will at once think of other chests for similar duty with much less than ideal dimensions. However, the procedure shown will still result in a finite momentum number required for the application. Many times it will take some imagination for some odd-shaped chests but, as noted in Example No. 2, you must consider all conditions in which the chest will be used before making a final choice.

There's a relationship between the final momentum number required and the volume of the chest which allows some simple scale-up calculations as well as some simple formula for specific applications such as high-density towers, *couch/press* pits and white water chests. So let's go on!

Chapter 8:

Process Horsepower III

We have already discussed the derivation of the momentum number and its meaning in terms of flow and head (velocity), i.e., QV. Now let's look closely at its relationship to the volume of pulp under agitation and how this relationship can make our calculation procedures more direct.

We have previously shown the relationship:

$Mo \propto N^2 D^4$ (1)

Or specifically

$Mo = CN^2 D^4$ (2)

Where the units of momentum are:

$Mo = ft^4/sec^2$ (3)

If we calculate the volume of the chest as:

$V = \pi T^2 \times Z/4$ (4)

$V = ft^3$ (5)

And raise V to the ⅔ power:

$V^{2/3} = ft^2$ (6)

And divide Mo by $V^{2/3}$:

$Mo/V^{2/3} = ft^4/sec^2 \times 1/ft^2$

$= ft^2/sec^2$ (7)

Which we call "level momentum":

$\overline{Mo}$ = level momentum (8)

This becomes a scale-up factor for larger but similar shaped chests.

Example: Let's take a simple case of two chests, each to handle pulp at 4% b.d., at ambient temperature and with a stock factor of 1.0:

Chest #1: 10 ft diam. x 10-ft stock level (Z).

Using the curves from Chapter 7:

Mo = 490

$V = 785.4\ ft^3$

$V^{2/3} = 85.12\ ft^2$

$\overline{Mo}$ = 5.76 level momentum

Chest #2: 20-ft diam. x 20-ft stock level (Z)

Using the same curves:

Mo = 1900

$V = 6283\ ft^3$

$V^{2/3} = 340.5\ ft^2$

But if we multiply $\overline{Mo}$ for Chest #1 by $V^{2/3}$ of Chest #2:

Mo = 5.76 x 340.5

Mo = 1961 momentum required, Chest #2.

This is only 3% over that calculated for Chest #2 using the plots in Chapter 7 and represents a more accurate mathematical answer than using the stepwise calculation.

This technique can also be used when the chest shapes are slightly dissimilar.

Example: In scaling from a small chest with $Z/T = 0.9$ to a larger chest with $Z/T = 1.0$, the Fig. 7-2 factors would be 0.83 and 1.0 respectively. The multiplier on the basic process number would be $1.0/0.83$ or 1.205. To correct the Mo using the smaller chest as reference, we must use this factor, but raised to the 0.8 power as that is the relationship between the correction factors and the level momentum:

$$1.205^{0.8} = 1.16 \times \overline{Mo}$$

N. B. The 0.8 exponent must be used on any geometric or process factor difference when scaling up with Mo as that is the slope of the CPN vs. Mo plot (Fig. 7-6) raised to the 2/3 power. We shall see how this is used in the more direct methods that follow.

Finally, if we go back to the units of level momentum and take the square root of that value, we will have:

$$(ft^2/sec^2)^{0.5} = ft/sec \qquad (9)$$

Which is a theoretic calculation of the average velocity in the chest.

There are a number of chests or agitation applications in which the geometry is fixed by the dimensions of the paper machine or the industry standards for a particular operation. In those cases, it's possible to shortcut most of the modification curves explained in Chapter 7 and go directly to a process requirement using the level momentum concept. We will discuss a few of the most common applications in this chapter, specifically:

(1) High-density storage chests
(2) Couch pits
(3) Press pits
(4) White water chests

(1) High-density storage

You will recall from previous discussions, the evolution of the reduced-bottom tower and the concept of controlled-zone agitation. What was once jokingly referred to as the "upside down milk bottle" has become a standard design in the industry.

Originally the process requirement was laboriously calculated using all the modification factors described in Chapter 7, with additional "crutches" to account for the controlled zone under a head of unagitated stock and the effect of a full 45-degree fillet.

This has all been changed using the concept of level momentum and the fact that the agitated zone is always equivalent to a Z/T of 0.5 and the retention time designed around an optimum value of 12 minutes. Let's go through the derivation of this calculation and then solve a typical problem, beginning with the design of the tower.

A basic $\overline{Mo}$ of 5.41 was established for a controlled zone with a Z/T of 0.5 in a particular stock having a stock factor of 1.0, retention time of 12 minutes and a consistency of 4.0% b.d. We required some formula that would express the volume of the zone in any size tower. Using T1 in a reduced bottom tower, or T for a straight shell tower, as in Fig. 6-7, we can derive:

$$Ft^3 = \pi/4 \times T1^2 \times Zm$$
$$\text{But } Zm = 0.5\ T1$$
$$Ft^3 = \pi/4 \times (T1^3)/2$$
$$Ft^3 = \pi/8 \times T1^3.$$

Then the value of $V^{2/3}$ becomes:

$$V^{2/3} = 0.54 \times T1^2 \qquad (10)$$

If we incorporate the constant, 0.54, into the $\overline{Mo}$, we have:

$$\overline{Mo}' = 2.92 \qquad (11)$$

Now, if we wish to set up a table for general use, it seems logical to group some pulps together and pick limits of stock factors. The first level includes stock factors from 1.0 to 1.3. Modifying Eq. 11 by the 1.3 stock factor raised to the 0.8 power, we have:

$$\overline{Mo}'' = 2.92 \times 1.30^{.8}$$
$$= 3.6 \qquad (12)$$

This is the level momentum used for all stocks at 4% b.d. to maximum stock factor of 1.3, in a standard bottom zone without a back-wall fillet.

Before this concept was used, we attributed a multiplier to the basic process number of 0.59 to account for a full back-wall fillet of 45 degrees. Therefore, to modify $\overline{Mo}''$ further we have:

$$\overline{Mo}''' = 3.6 \times 0.590^{.8}$$
$$= 2.4 \qquad (13)$$

Now we have corrected level momentum for all stocks at 4% b.d. to a maximum stock factor of 1.3 for applications with and without back-wall fillets. Similarly, we can calculate standard values of level momentum for higher stock factors, with and without fillets, until we establish a simple table that covers most applications, we will call these modified values $\overline{Mo}'$:

Table 1

Stock Factor Range	With Fillet	W/O Fillet
1.0-1.3	2.4	3.6
1.4-1.8	3.1	4.6
1.9-2.1	3.5	5.3

Now we can write an equation for the process momentum requirement for all stocks at 4% b.d. and a retention time of 12 minutes or greater as:

$$Mo = \overline{Mo}' \times T1^2 \qquad (14)$$

But of course we won't always be dealing with a consistency of 4%, nor will we always be fortunate enough to be able to design a tower to 12 minutes or greater retention time. We already know the basic process number is proportional to the consistency cubed, and we can see from Fig. 7-5 that the slope of the retention curve is 0.5, $Rt \sim (12/t)^{0.5}$. Therefore, each of these must be corrected by the exponent 0.8 to convert to direct use with level momentum:

$$\overline{Mo}' \sim (\%)^{2.4} \qquad (15)$$

$$Rt \sim (12/t)^{.57} \qquad (16)$$

Now we can write the complete equation for the process requirement for any high-density tower under all conditions:

$$Mo = \overline{Mo} \times T1^2 \times (\%)^{2.4} \times (12/t)^{.57} \qquad (17)$$

You may calculate the consistency and retention factors for any application or you may use the graphs, Fig. 8-1 and 8-2.

N. B. At 12 minutes retention, Rt becomes 1.0. Do not use Rt for any retention time greater than 12 minutes!

Let's design a new high-density tower using the following data: (N. B. refer to Fig. 6-7 for design shape)

Capacity—300 tons at 12% b.d.

Production—750 T/D —4.50% b.d. in agitated zone.

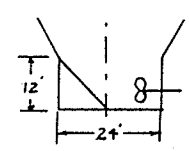
Furnish—Unbl., unref., SWD kraft, K# 24.

750 T/D @ 4.5% = 2278 gpm

At 12 min retention, Vol. = 33,340 gal

A chest with Z = 0.5 x T1 to hold near this volume might be: 24 ft T x 12 ft Z = 40,608 gal

With a 45 degree back-wall fillet, volume would be: 40,608 x 0.78 = 31,674 gal



Retention = 11.4 min

Using a 1.7:1 ratio T2 to T1, T2 = 40.8 ft

Use T2 = 40 ft

The conical section between T1 and T2 is at 60 degrees. The height of the section is then calculated as

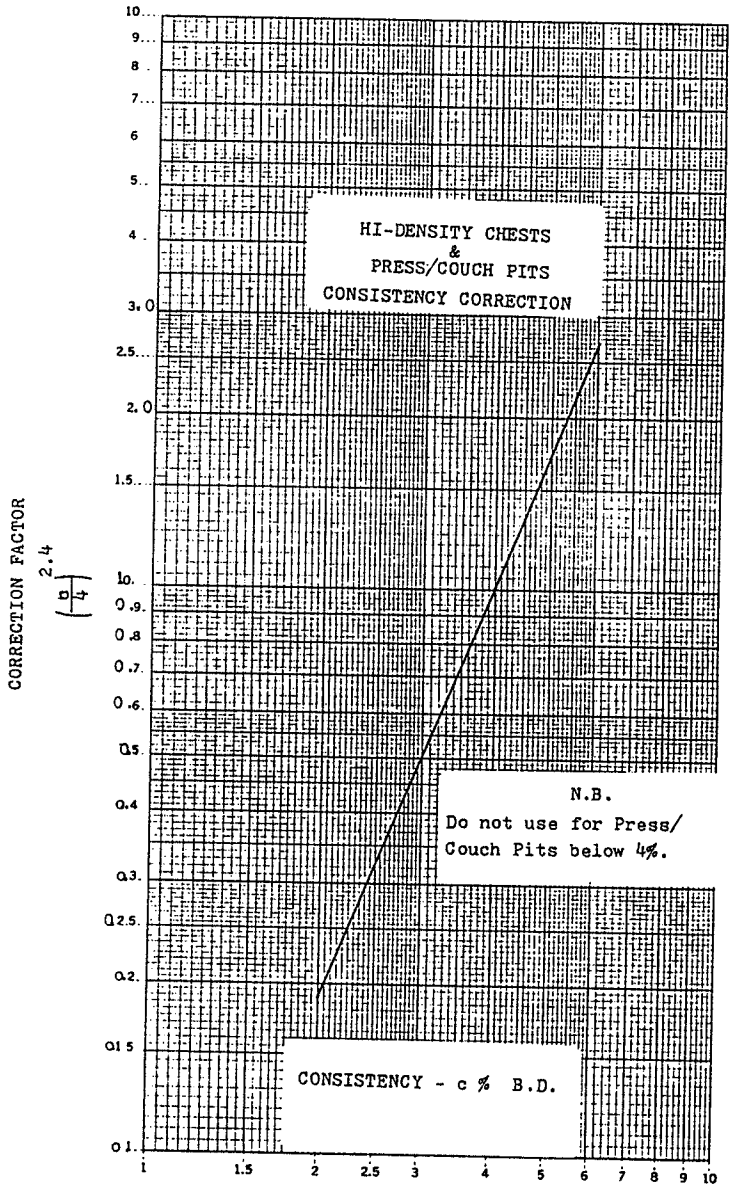
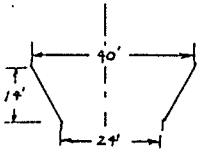
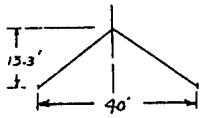


Figure 8-1. Consistency vs. Process Factor (Hi-density & Couch/Press Pits).

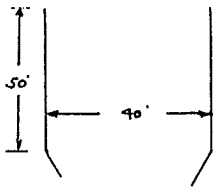
13.9 ft. Use 14 ft for this calculation.



Volume of conical section: Use average diameter at a stock level of 14 ft:
(40 + 24)/2 = 32 ft T x 14 ft Z = 84,224 gal
84,224 gal @ 12% = 42 tons at 12% b.d.



At the top of the tower, assume angle of repose as approximately ⅓ T2 in height.
40 ft T2 x 40/3 Z = 41,778 gal = 21 tons @ 12% b.d.
The remainder of the 300 tons to be contained in the straight shell portion of the tower:



300-(21 + 42) = 237 tons @ 12% = 474,000 gal.
Volume/ft at 40 ft T2 = 9400, 474,000/9400 = 50 ft
Straight shell height required = 50 ft
Total height of tower from bottom of dilution section to top of stored volume:
Z = 12 + 14 + 13.3 + 50 = 89.3 ft. (Call it 90 ft.)
Ratio Z/T1 = 3.75
OAH of tower (H) must include free board, cover and foundation.

This design will be a little too tall for the best economic chest design. Another trial could begin with a larger-diameter bottom, T1, which will allow a significant

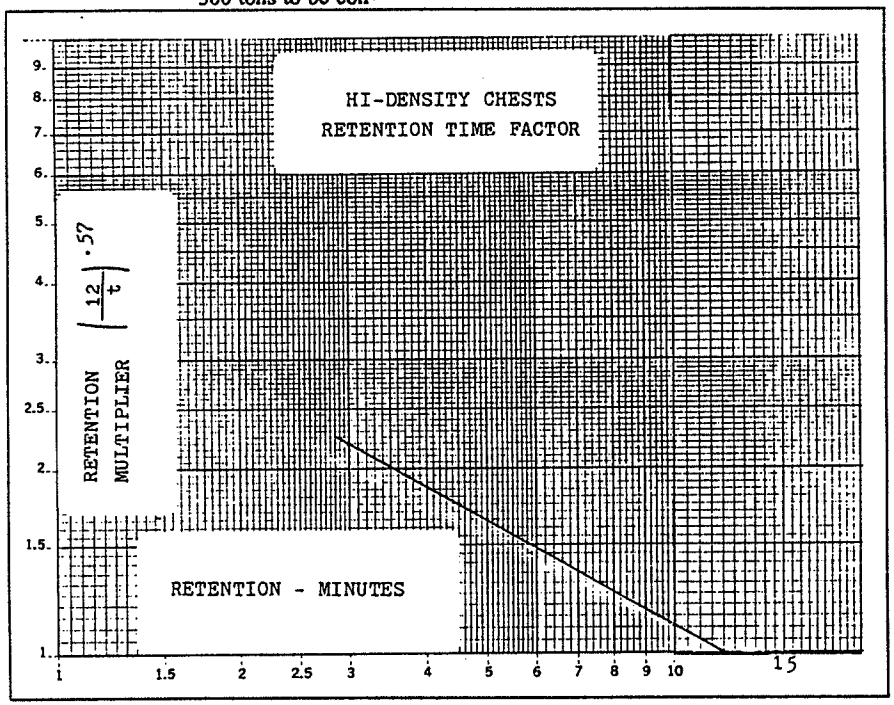


Figure 8-2. Retention Time vs. Rt - Hi-D only.

increase in T2 and thus reduce the OAH. This will increase the retention time in the dilute zone appreciably with a comparable increase in the required process horsepower. Such evaluations will have to be made to compare the total capital costs against the increase in operating costs. Remember from Chapter 6, the desired Z/T_1 should be approximately 3.0. However, for the purpose of this example, these data will be satisfactory.

Agitator design

Stock factor for this pulp, Fig. 7-2 = 2.1
Full back fillet $\overline{Mo}'$ from Table 1 = 3.5
Mo = 3.5 x 24^2 x (4.5/4)^2.4 x (12/11.4)^0.57
Mo = 2754
From Table 7-2 and Figs. 7-7 and 7-8
54 in. @ 150 hp = 2772 x 0.967 = 2681
60 in. @ 150 hp = 2946 x 0.967 = 2849

These answers pose an interesting choice. Both final Mo's are within 3% of the theoretical case. There is no difference in operating costs (150 hp each), but the higher speed for the 54-in. propeller may mean a significant difference in capital cost (reducer size). A further investigation will be necessary but before going into that, let's use our imagination for a moment.

Remember, I said an active imagination must also be applied to these selections. Regardless of which of these two units is selected, we will still have a single agitator in a tower holding 300 tons of stock. Suppose that unit should fail for whatever reason! How would you dump this chest? If we had two units, side by side, parallel to each other, at 1.5 propeller diameters spacing to provide a single flow pattern with the pump suction between them, the second unit will provide enough agitation near the pump suction to allow you to evacuate the tower. Consistency control will be depreciated, but at least you can "dump" 300 tons of stock without drastically changing your stock flow. How do we do that?

The requirement is Mo = 2754.
With two units we need 1377 each!
From table 7-2:
48 in. @ 60 hp = 1429 x 0.967 = 1382
42 in. @ 75 hp = 1482 x 0.967 = 1433
1433 x 2 = 2866-more than enough!

A further bonus, both units would be V-belt driven, less expensive than the gear-driven units, and less maintenance.

Therefore, the most reliable recommendation for this tower would be:

2-75 hp units with 42-in. propellers (16-degree pitch)

Units to be side by side, straddling the pump suction and installed 63-in. ℄ to ℄.

(2) Couch pits

The couch pit under a paper machine has always been of great concern to the operating people. It's also a concern to the agitator supplier who may have spent many hours under the infuriating glare of the machine superintendent as the pit consistently plugged with high-consistency pulp and overflowed the machine sills. There were a few nights in a southeastern newsprint mill when I fervently prayed I had never heard of a couch pit! However, out of such catastrophes comes understanding and one learns to appreciate the experience some years after the embarrassment and verbal blasting have been forgotten (almost). There are different methods of designing a couch pit, different ways of operating the pit and, of course, different expectations by the mill personnel.

Three different methods of operation, requiring different agitation selections *(13)* are as follows:

1. Couch pit always at low consistency, 1-2%, pumping out to a thickener with thick stock going to a broke chest and underflow to a saveall.
2. Couch pit liberally showered to 1% or less at all times and pump out only to a saveall.

3. Couch pit maintained at a consistency compatible to saveall feed except during full machine break. Pump out is to a broke chest during a break, or to a saveall during normal machine running time.

The first two types of operations require a large pit to obtain a longer retention time and minimize upsets to downstream process equipment. On some paper machines, press broke may be accommodated in this type of pit. The third method of operation, sometimes called a "swing couch," is the one we will consider for the agitation selection because it is critical for smooth operation of the paper machine and associated pieces of process equipment. In a "swing couch," we must keep the retention time within narrow limits, ideally 3-5 minutes.

This is so that the moment a break occurs and the full sheet hits the pit, the consistency will rapidly increase to a level compatible with the broke chest. When the break ends, the pit must be able to return quickly to a consistency compatible with the saveall. The drastic alternatives to this are a dilute and overflowing broke chest followed by a plugged saveall; neither of which improves the disposition of the machine superintendent.

It isn't always possible to stay within these narrow limits of retention time. The minimum size of the couch pit is dictated by the cross-machine direction of the machine and a proportion of that dimension in the machine direction for optimum agitation results. On larger machines making a lightweight sheet, e.g., newsprint, machines are often quite wide for moderate tonnage rates compared to heavy liner board grades at high tonnage. In the case of extreme retention times with slower increase in consistency during a break, it would be important to consider a delay in switching to the broke chest at the start of the break and another delay in switching back at the end of the break to allow time for the consistency to rise and then fall back. However, this would be a ticklish procedure because the level in the pit would be rising as we hold back flow to the broke chest for a period of time. There are other ways to reduce retention time

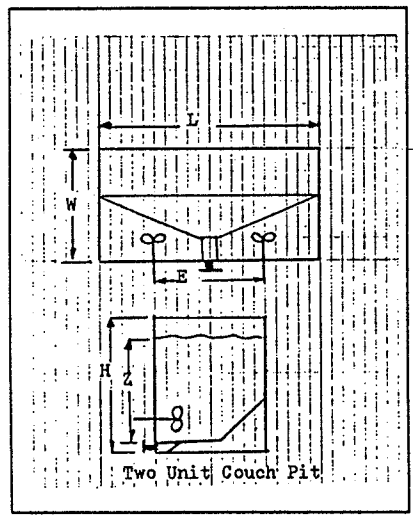


Figure 8-3. Two Unit Couch Pit.

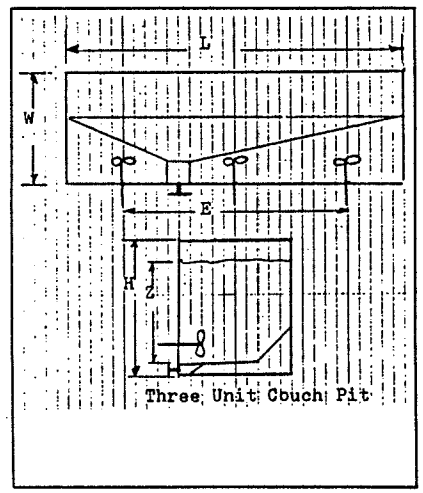


Figure 8-4. Three Unit Couch Pit.

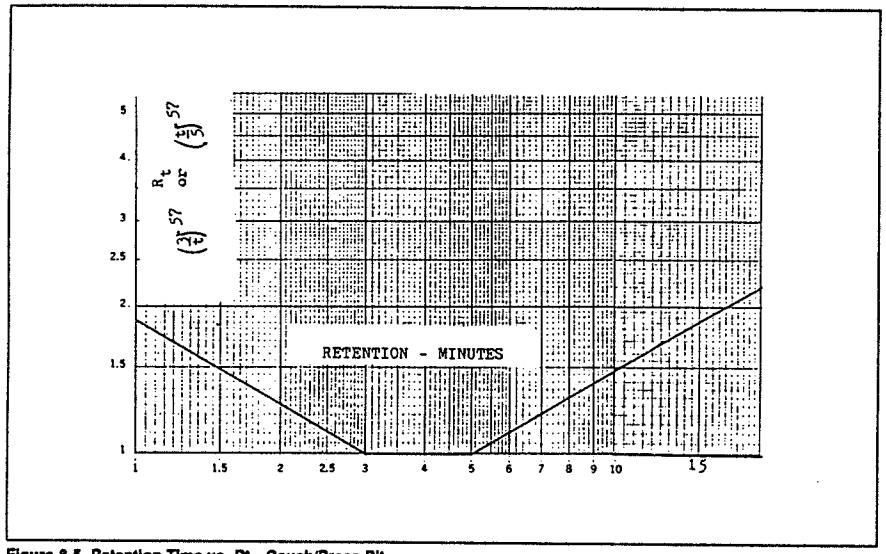


Figure 8-5. Retention Time vs. Rt - Couch/Press Pit only.

which we will discuss later. Retention times shorter than three minutes can also occur, though not as often, and those cases abet the quick change in consistency but make the selection of the agitator(s) more critical.

The design of the couch pit for agitation actually involves the design of one or more rectangular chests. Ideally we want to have one or more "cubes" under the paper machine, i.e., a chest, or chests, with width (W), length (L) and stock level (Z), all equal. A machine that measured 20 ft in the cross-machine direction might then be considered as having two chests, each 10 ft wide, 10 ft long (machine direction) and with a 10-ft stock level. Two agitators would be required set at ¼ points on the cross-machine wall. Likewise, for a large machine, say 30 ft across, we would consider three units, imagining three chests each 10 ft by 10 ft and with a 10-ft stock level. Figures 8-3 and 8-4 illustrate these designs. If with this initial rough design, we find the retention time excessive, we still might change it by considering a shorter machine direction and using an additional agitator or by introducing large back-wall fillets to reduce the volume.

Example: A couch pit measuring 20 ft cross machine by 10 ft machine direction with a normal stock level of 10 ft gives a capacity of 2000 ft^3 and a retention of 10 minutes at full machine tonnage. The pit had been designed for two agitators. By decreasing the machine direction and normal level to 7 ft, we can reduce the capacity to 980 ft^3 and reduce the retention time at the same tonnage to just under five minutes. Three units would be required but the total installed horsepower would be less, still satisfying all requirements.

A direct method of determining the momentum number required for a couch pit application makes use of a number we call momentum days per ton, (Mo $^{D}/_{T}$), similar to the horsepower days per ton, (hp $^{D}/_{T}$), we use in dry-end pulping applications. When this factor is multiplied by the machine rate in T/D and modified by a factor for (a) paper grade sheet factor, (b) consistency above 4% and (c) residence time factor if less than three minutes or longer than five minutes, a final momentum number is obtained. The equation looks like this:

$$Mo = Mo\ ^{D}/_{T} \times T/D \times SF \times (^{C}/_{4})^{2.4} \times Rt \quad (18)$$

Where: Mo $^{D}/_{T}$	= 2.4
T/D	= production rate
SF	= sheet factor
Newsprint	= 1.2
Linerboard	= 1.3
Kraft bag	= 1.3
Corrugating	= 1.3
Book, Print	= 1.4
Bl. kraft	= 1.4

Rt = <3min. $(^{3}/_{t})^{.57}$; >5 min $(^{t}/_{5})^{.57}$

N. B. Figure 8-1 may be used for consistency correction.
Figure 8-5 may be used for Rt correction.

$$Mo = 2.4 \times T/D \times SF \times (^{C}/_{4})2.4 \times Rt \quad (19)$$

The momentum number calculated is the total required for the pit. To find the individual unit momentum, divide by the number of units required.

Restrictions

1. The equation assumes a pit design for one or more "cube" shaped chests as defined earlier. If the chest configuration involves an $^{L}/_{W}$ or $^{Z}/_{W}$ greater than 1.0, a shape factor correction raised to the 0.8 power, from Chapter 7 must also be used.

2. This is correct for consistencies above 4% only. DO NOT make a correction for less than 4%. If tissue grades or other situations in which the consistency during a break is less than 2%, these data will give excessive agitation. With caution, the Mo $^{D}/_{T}$ could be lowered to 1.5 in these extreme cases.

Let's try one or two examples:

1. A linerboard machine: 32 ft across machine. 1400 T/D, 42# liner, 4% in pit during break (5835 gpm).

a. If we assume a pit 32 ft across, 16 ft machine direction and 16 ft stock level, we will calculate a volume of 61,276 gal and a retention time of 10.5 min.

b. If we assume a pit 32 ft across, 11 feet machine direction and 11 ft stock level, we will calculate a volume of 28,963 gal and a retention time of 5 min.

Using the pit from (a):

$Mo = 2.4 \times 1400 \times 1.3 \times 1.0 \times (^{10.5}/_{5})^{0.57}$
Mo = 6667

At two units each at Mo=3334, we would require 2-150 hp, 72" propellers.

Using the pit from (b):
$Mo = 2.4 \times 1400 \times 1.3 \times 1.0 \times 1.0$
= 4368

At three units each at Mo = 1456, we would require 3-60 hp, 48-in propellers, or 3-75 hp, 42-in. propellers.

Obviously the pit designed in (b) is preferable with either unit selection.

2. A newsprint machine: 28 ft inside machine walls. 500 T/D, 30# news, 3.5% in pit during break (2381 gpm).

a. Assume a pit 28 ft x 14 ft x 14-ft stock level. Calculate a volume of 41,050 gal, retention time of 17.2 min.

b. Assume a pit 28 feet x 9 feet x 9-foot stock level. Calculate a volume of 16,965 gal, a retention time of 7.1 min.

Using the pit from (a):
$Mo = 2.4 \times 500 \times 1.2 \times 1.0 \times ((^{17.5}/_{5})^{.57}$
Mo = 2940
Using two units at 1470 each we require:
2-60 hp 48-in. propellers
Using the pit from (b):
$Mo = 2.4 \times 500 \times 1.2 \times 1.0 \times (^{7.1}/_{5})^{.57}$
Mo = 1759
Using three units at 586 each we require:
3-25 hp 30-in. propellers.

Again, the pit designed in (b) is preferable. The seven-minute retention time might be further reduced by the inclusion of large back-wall fillets. You might want to work a few more examples, perhaps

from your own experience or in your own mill.

(3) Press pits

The press pit, if you have one under your machine, is selected in a similar manner to the couch pit. However, there are some operational differences to understand. Its use conforms more to the size press or dry-end pulper; it shouldn't vary in consistency beyond the desired 3.5 to 4.0%, and pump out will always be to the broke chest or some intermediate chest of similar consistency. There are some installations where the press pit is "slaved" to the dry end pulper—not a desirable situation, but we won't go into that at this time.

The press pit receives isolated broke from the second- or third-press section and, on machines that are difficult to thread from the couch to the first dryer, it is the favorite spot to drop the sheet when there is trouble on down the machine. The incoming broke, unlike the couch which is seldom greater than 20%, can be from 35 to 45% consistency depending upon the grade being made. Special cases of press broke will be discussed in the next chapter. However, at 40% or less incoming broke consistency, the press pit is designed and sized for propeller agitation just as we did for the couch pit with one exception. It's more difficult to repulp the 40% sheet with a propeller than the "raggy" 20% sheet off the couch roll. Therefore, the Mo D/T must be increased to a value of 3.6, making the equation:

$$Mo = 3.6 \times T/D \times SF \times (c/4)^{.57} \times Rt \qquad (20)$$

All of the previously mentioned data and graphs will apply.

(4) White water chests

Agitation in a white water chest isn't only uncommon, but is undesirable in many cases. In non-filled grades such as liner, corrugated and even newsprint, the high flow rates and minimum fiber content readily flow through the chest without sludge or slime buildup. Chests with smooth walls and generous fillets aid in the cleanliness of such installations. However, some chests in older mills may have less than optimum shapes and fillets, leading to stagnant areas in which fibers may collect and buildup. More important, highly filled sheets can leave increasing solid buildups, even in ideal shapes, unless relieved by an agitator.

If vendors who market mixers and agitators know anything about agitation, they should find the agitation of water (and white water is little more than that) the simplest of applications. If we were to retreat to the "hoary" method of selection, horsepower per unit volume, we would say an input of 0.5 hp per 1000 gallons would be ample for rapid turnover in most reasonably shaped chests. But with the knowledge we have gained about momentum in the last two chapters, we know that 0.5 hp per 1000 gal. can be achieved in many different ways. If we convert that crude selection procedure into a level momentum number and assume a chest, either vertical cylindrical or ideal rectangular, with a shape factor of 1.0, we discover that a level momentum of 1.23 multiplied by the chest volume in cubic feet raised to the ⅔ power would give the momentum required:

$$Mo = 1.23 \times (ft^3)^{2/3} \qquad (21)$$

But in those cases requiring agitation in a white water chest, it isn't just the circulation of water that concerns us. We may have a very poorly shaped chest from an agitation standpoint, dictated by the geometry of the site. More importantly we may have a heavy concentration of filler, clay, TiO_2, etc., that must be kept uniform lest we release slugs of settled material sporadically to the system.

From an analysis of a multitude of installations, a higher value of level momentum has been extracted, modified by the shape factor that may apply, Figure 7-2, again raised to the 0.8 power as we are dealing with $\overline{Mo}$, to arrive at the correct momentum number for the installation. The equation now becomes:

$$Mo = 1.6 \times ft^3 \wedge 2/3 \times S.F.^{.8} \qquad (22)$$

Notice the retention time plays no part in this process design. It's usually quite short and increased throughput only helps to maintain uniform suspension.

Let's try just one example:

Chest	12 ft W x 18 ft L x 14 ft Z
Volume	3024 ft^3 (22,620 gal)
Vol. 2/3	210 ft2
$L/W = 1.5$;	$Z/W = 1.17$
S.F. (Fig. 7-2) = 1.8	

$Mo = 1.6 \times 210 \times 1.8.^8$
$Mo = 538$

Unit selection (Table 7-2):
20 hp, 36-in. propeller
25 hp, 30-in. propeller

The 20-hp unit would be satisfactory, but notice that if we revert to the hp/unit volume basis, this represents an installed value of 0.88 hp/1000 gal, and our old rule of 0.5 hp/1000 gal would have left us short and possibly in trouble.

Even in this simple application, we must be aware of basic requirements and how these are affected by adverse chest design. This isn't a well-proportioned chest for this service. It might be perfect for a niche in the basement, using the floor-to-floor dimension to give additional height with reasonable free board, but for optimum process results and an economic installation, the chest is too long compared to its width and the level is too high to permit a minimum selection. But we seldom can control those dimensions, and thus this method of selection is preferred to achieve optimum results.

As you become more familiar with the momentum number concept, you will see other standard applications that can be reduced to a level momentum number.

Now let's discuss some special types of agitation equipment and where these are used.

Chapter 9:

Special Types of Agitators

General

Thus far, we have only discussed four types of impellers used in the agitation of paper pulp:

1. The flat paddle
2. The propeller as a circulator
3. The spiral backswept turbine
4. The propeller as an agitator
5. The "Maxflo" impeller (proprietary device of Prochem Ltd.).

These have been presented by historical chronology within our industry. At the present stage of evolution, the "modern agitator" is most often a three-bladed propeller, though we are now moving toward a hybrid three- or four-bladed axial-flow impeller at very low pitches. The greatest percentage of all pulp agitation devices in service, or going into service today, employ some type of axial-flow impeller and use the side-insert configuration for the most optimum process performance to energy-consumed ratio. Manufacturers are continually seeking ways to improve the efficiency of their particular impeller, and I imagine that the search for the "magic impeller" will continue as long as we continue to make paper by the "wet process." However, the "eurekas" died down some time ago and the phrase "special types of agitators" mostly refers to adaptations of the axial-flow unit to meet special circumstances rather than any breakthrough to a new design.

Special Modifications

A. Wiped extraction

The greatest number of agitators are installed in stock chests handling completely pulped stock, serving only to keep the chest or a portion of it in uniformity while the discharge pump draws from an open suction. The sump or insert leading to the pump is placed near the agitator location only to ensure that it is the most active area of chest, even during pump down.

However, for those few applications in which the agitator is "part of the act" in

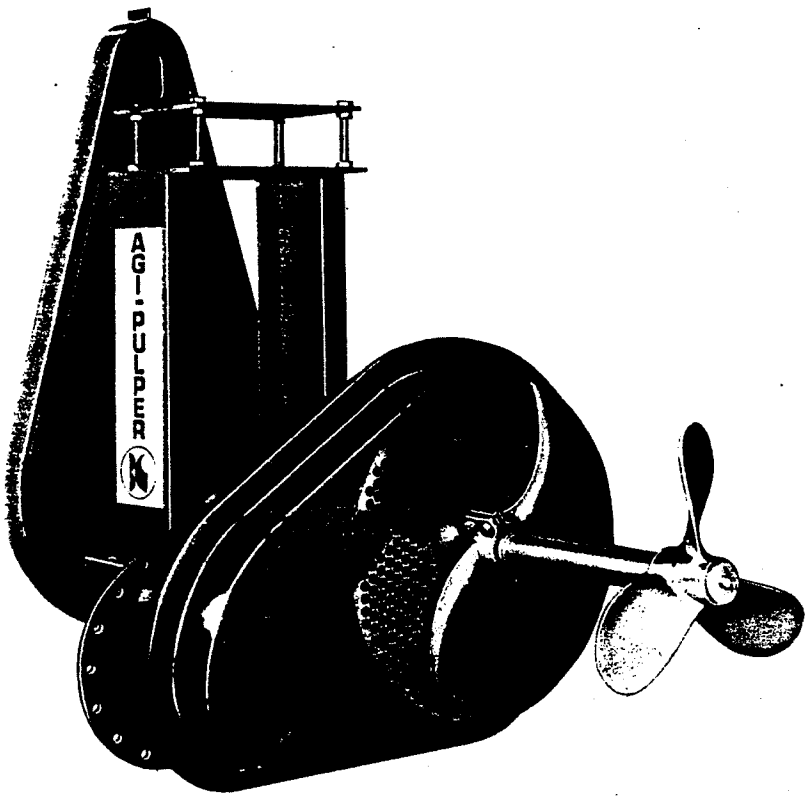


Figure 9-1. Side Insert with Wiped Extraction (Black Clawson).

producing a pumpable slurry, there may be some concern over how an open suction installation might affect the pump or next piece of process equipment downstream from the pump.

An agitator installed in a press pit would be a good example. The "job description" at this location involves the continuous random circulation of pulp at 3.5-4.0% consistency plus a violent repulping action on the incoming sheet during a break on the machine. The sheet at the press location can be between 35-45% consistency which means it has a certain amount of strength and resists being repulped into individual fibers. If we were to pump this repulped slurry directly from an open 8- or 10-in. outlet, the danger of pulling out large clumps of unpulped sheet would be great. At worst, we might blind over the pump suction or plug the pump. A lesser, but still undesirable result, would be to deliver an excessive amount of large flakes to the broke chest or to damage a consistency probe or control valve. The answer has been to provide an extraction

chamber, similar to those used on dry-end pulpers and to pipe the pump suction to this chamber. Making this integral with the agitator allows the manufacturer to install a single- or double-bladed wiper on the shaft, which rotates at shaft speed and in close proximity to the extraction grate, wiping the grate clean. Such a design is shown in Fig. 9-1. Hole size is usually ¾ to 1 in., just small enough to stop large pieces of unpulped sheet. Horsepower isn't affected as the wiper consumes a minuscule amount of energy. The velocity across the grate and through the elbow off the chamber, must be closely checked. It isn't desirable to exceed three ft. per second at any point on the suction side of the pump, but especially so across the extraction grate. In a press pit requiring two or more agitators, it's usually necessary to use this type of unit for all units supplied since it would be unlikely to be able to extract the total tonnage through one unit.

The couch pit doesn't require wiped extraction. The incoming sheet during a break seldom has greater than 20% consistency. It also has very little strength and "rags out" pretty well just on contact with the water. The violent action of the agitator(s) does a nearly complete job of breaking down the sheet for each pump out through an open suction.

B. Cross shaft propeller agitator

One almost has to apologize for this relic of the "Dinosaur Age," but it has retained its usefulness—indispensability—through all the other advances in agitator design. It still finds a place for itself where even the best designed "modern agitator" has yet to perform successfully. It does its work "the old-fashioned way" by brute force. Figure 9-2 shows a typical design. It's not pretty to look at, and it's not cheap. It looks a little like the vertical "Christmas tree" circulator laid on its side. Generally, it consists of a large-diameter pipe shaft, 8- to 18-in. diameter, spanning the paper machine up to 30+ ft. across, if

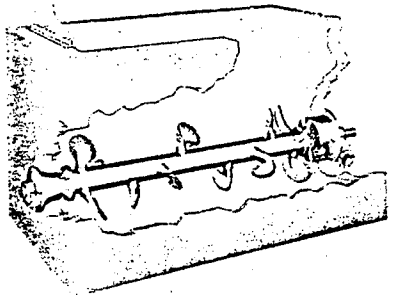


Figure 9-2. Cross Shaft Propeller Agitator (Brinkley).

necessary. It always has one wiped extraction chamber, often two, one at each end which makes for a "not so neat" piping system. The pipe shaft carries a number of single-propeller blades arranged in a spiral pattern the length of the shaft, for which we drop the term "propeller diameter" and refer to "swing diameter." Where do we hide this "iron worker's delight?" Most often in a press pit on a paper machine when the sheet off the third press is above 40% in consistency. We've already talked about the selection of side-insert propeller agitators for couch and press pits (Chapter 8), but we have found from experience that a 40% sheet is about the maximum weight that this type of unit can pull down, repulp and circulate to uniformity. A heavier sheet tends to kill the velocity of the flow pattern, and areas between the two (or three) single-propeller units become stagnant and the repulping action "dies." The cross-shaft unit with its multiple blades, sometimes 10 to 15 on a wide machine, distributes the total horsepower more evenly across the pit. This produces flow in the cross-machine direction and breaks up the sheet by brute force! This style of unit has also been used as a dry-end pulper on some newsprint machines. Its main disadvantage in that position is that it requires a much longer retention time, 10-15 minutes, which requires a

"tub" sometimes extending from the last dryer all the way back to the reel. Such a "municipal swimming pool" often requires two or three of these units. However, off-

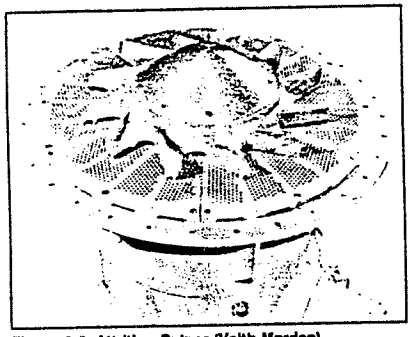


Figure 9-3. Attrition Pulper (Voith-Morden).

setting this added expense in those mills that have used them successfully, has been the elimination of one or two conveyors that might have been required to bring dry broke from several broke holes to the narrow tub used with a normal attrition pulper.

The process selection of this machine is rather crude. At the wet end (press pit), the total installed horsepower is based on a fraction of the hp $^{D}/_{T}$ normally used at the dry end. For example, on a 500-T/D newsprint machine, we would likely install 150 hp on a single cross-shaft unit. At the dry end, we would consider 0.9 to 1.0 hp $^{D}/_{T}$ divided among as many units as were required in a pit equivalent to 10-15 minutes retention time. You don't need a computer for this selection procedure!

Loading the machine to 80% of motor horsepower is also partly science and partly empirical. We would use the power number relationship we described in Chapter 5, use the swing diameter as the propeller diameter and divide the total number

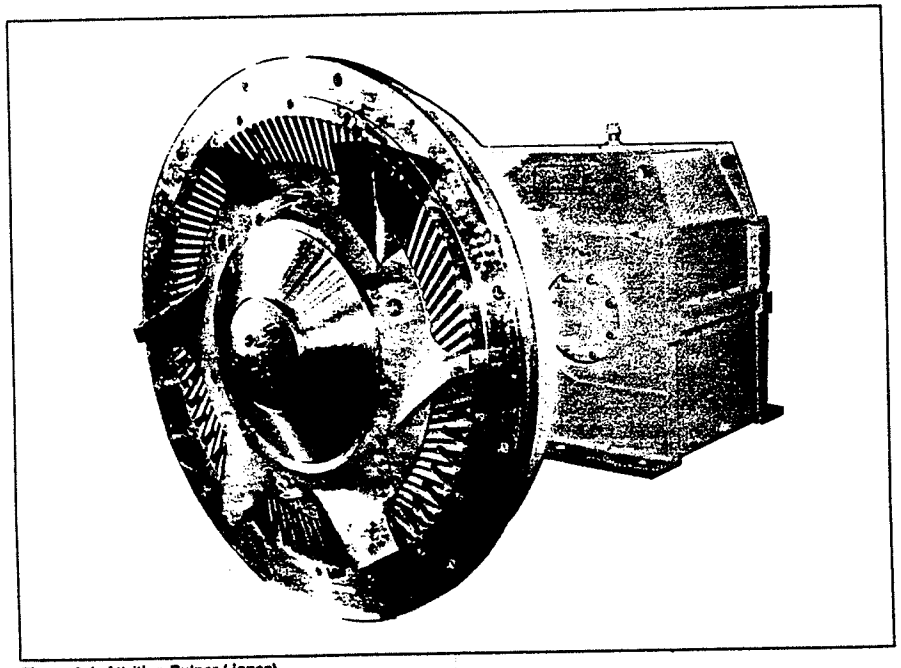


Figure 9-4. Attrition Pulper (Jones).

of single blades by three to arrive at a fictitious number of three-bladed propellers.

Example: A cross-shaft agitator has 12 single blades with a swing diameter of 50 ins.. We wish to properly load a 200-hp drive in 4% stock.

Theoretical number of propellers at 50-in. diam.:

$12/3 = 4$

Power number from Chapter 5:

$Np = 0.36$

$hp = [[(Np \times N^3 \times D^5)/ 283.8] \times 4]$

$N = [(283.8 \times 160) / (.36 \times (50/12)^5 \times 4)]^{1/3}$

$N = 2.93$ rps or 175.5 rpm

We like to use a propeller pitch equal to or greater than 18 degrees for this class of service:

$hp^1 = 160 \times (170^{*}/175.7)^3$ * where 170 = std. optional ratio in rpm

$hp^1 = 145$ and $160/145 = 1.10$

Pitch factor at 20 degrees—1.12

$hp = hp^1 \times 1.12$ or 162 hp—satisfactory

The cross-shaft agitator has been in use for many years and, in spite of my rude comments about "her," "she's a good old girl," and will likely be with us for many years to come. There are just some areas where "she" is the only device to do the job!

C. The attrition pulper

Maybe you aren't familiar with this term. I've used it a number of times in this "history" because I've tried to be fair with all vendors. You are probably more familiar with it as, "the Shark pulper," "the Slushmaker," "the Brute," "the Hydrapulper" or just plain "dry-end pulper." It's the necessary "energy eater" at the dry end of your machine, usually between the last dryer and the stack with occasionally an extra one at the size press. Figures 9-3, 9-4 and 9-5 describe some typical units. These are built in side-insert and bottom-entry configurations to suit the geometry or restrictions of individual paper machines. Now why do I include this as a special type of agitator? Well, mostly because *it is an agitator!* and a very special type of agitator. In mixing terminology, it's a "single-suction, radial-discharge turbine." It's a "lousy" circulator (and we'll discuss that shortly) because its main function is to "eat" broke. That's where the "attrition" name comes from; it

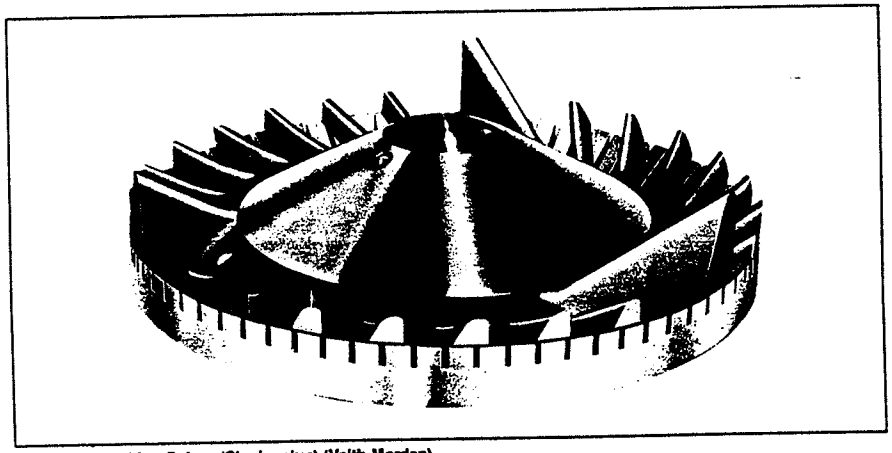


Figure 9-5. Attrition Pulper (Slushmaker) (Voith-Morden).

is designed primarily to submerge a dry sheet and tear it into little pieces by shearing action. So, if you remember that application spectrum from way back in the beginning, (chapter 4) its *Q/H* ratio is very heavy on the "H" and light on the "Q." Still, after it has done its job of making "little ones out of big ones," it must have enough pumping capacity left to distribute the repulped slurry in the tub. Because it's such a poor circulator, it needs help—help in the form of very rigid ratios of length to width and height of the tub. Special deflectors are needed for side-insert units to contain the discharge from the impeller and convert that energy into submergence.

I bring up this special "agitator" to emphasize how a specific requirement dictates the versatility or lack of it to a single piece of equipment. The dry-end pulper selection usually requires a power input of 1-1.6 horsepower for every ton per day across the machine, depending on grade, 1-1.6 HPD/T. There are other criteria that modify that value such as cubic feet per horsepower, (ft^3/hp), an indication of circulating capacity; and retention time relating to defibering capacity when combined with cubic feet per horsepower into a value of (ft^3/hp– min). We then discussed how a propeller agitator can repulp a wet sheet in the couch pit and with some limits in the press pit at relatively low levels of horsepower. We then talked about how a cross-shaft agitator can handle a tougher job in the press pit and even be used at the dry end on some grades with moderate horsepower levels. ***But, it doesn't work the other way!*** I have, many times, tried to dissuade a client from wanting to put an attrition pulper in a press pit, when the mill "wanted to do a 'super' job," because of the excessive horsepower required. A press pit that could easily get by with 150 hp on a cross-shaft agitator would still require 400-500 hp with an attrition pulper (nearly the same as at the dry end) because, although the repulping requirement would have been met many times over, the poor circulating qualities of the pulper would still require that much horsepower to circulate the tub.

D. Others

There are a number of other special types of agitators such as pipe line mixers, chlorine mixers, high-consistency mixers in a.d. systems, etc. But most of these aren't encountered in the everyday application in the mill. Remember, way back, I said "Let's follow through on just making a sheet of paper from a tree, nothing fancy." My concern has been to show the special modifications used for agitators under the paper machine and the relationship in applications as we moved from the couch roll to the reel.

Couch pit—Side-insert agitators

Press pit—Side-insert agitators to 40% incoming broke and with wiped extraction. Cross-shaft agitators above 40%.

Size press—Attrition pulpers or cross-shaft agitators on light grades.

Dry end—Attrition pulpers or cross-shaft agitators on light grades.

Now let's move on to mechanical considerations in the design of agitation equipment.

Chapter 10:

Mechanical Design

General

In this chapter, we will cover some of the common elements of good design and a few of the pitfalls. This won't be a doctoral thesis on mechanics; the vendor you favor for the purchase of equipment should have a solid reputation for building safe and reliable machinery. We want to emphasize features and choices of design and suggest what we believe to be most desirable for your application.

You should require that all proposals provide sufficient mechanical specifications to allow an independent confirmation of critical areas whether on one unit or on a group of units for a complete project. Of course you want to be assured that the equipment will perform to your process specifications, but you should also know the operating speed, shaft size, impeller size and weight, shaft overhung distance from the first bearing and bearing spacing. Most suppliers have specific limits on critical speed, but if you want to know just how safe your unit is, you may have to make your own calculations. The same reasoning is applied to shaft stress; to be absolutely sure of the value, you need sufficient data to make the calculations. Many suppliers don't like to give out this information, not because they are trying to hide something from you, but because it takes more time at the proposal stage. Some product descriptions are already in a computer program and special additions defeat the "automatic proposal" sequence, and some vendors just don't think you are that aware! If you have to wait until you've issued a purchase order to obtain critical data, it may be too late or too costly to make a change.

I'm not trying to be so "picky" to reduce an evaluation to a comparison of "pounds of iron" or a difference of a half in. in shaft size where the bigger shaft is "window dressing." But if you asked for a critical speed ratio of less than 0.2 or a maximum combined stress of less than

5000 psi or a B10 bearing life of 100,000 hours, you should have sufficient information in the proposal to prove those values. I'm certain some suppliers won't agree with this approach, but when I make an evaluation for a client, I insist on it or that vendor doesn't receive consideration.

Another area to be specific about before the "marriage" is spare parts. There will be enough proprietary items that you will have to buy from the vendor, and you will pay dearly for those no matter whose "parlor" you end up in. However, many items which will be listed as "recommended spares" may already be in your stockroom or readily available locally. The most common item will be bearings, and there are few, if any, agitators which require a "we are the only source" bearing. However, most suppliers will identify their bearings by their own stock number (because they often stock the same bearing by several manufacturers). You should ask for the specific bearing style, number and perhaps fit, so that you can find a reliable, and less expensive, local source for these important items. Other items will come quickly to your mind; some suppliers create a mystique over certain hardware at "gold rush" prices—after you cut through the "code" you find that ⅜ in. x 4 in. cap screws in 17-4 PH are available at your local hardware store. Packing can be another area, although precut sets with separators in individual boxes make servicing a stuffing box cleaner and more efficient than an inexperienced mechanic cutting rope packing on the top of the bearing member of the unit. You must be the judge of where to draw the line on supply of mechanical spares. The convenience of one purchase order may outweigh the cost savings you might otherwise make, at least on the initial purchase, but do you anticipate 20 or more years of service from this machine?—*caveat emptor!*

A. Speed reducers

It's obvious that all of the agitation equipment we are considering requires some type of speed reduction device between the prime mover and the agitator shaft. In my experience, I have seen agitators driven by gasoline engines, air motors, steam turbines, hydraulic drives and electric motors. Probably 99+% of all agitators utilize an electric motor as the prime mover. Since the cost of these motors increases rapidly with the number of poles in the frame, we should select a reduction device that allows us to stay with a 4-pole (1800-rpm) or, at worst, a 6-pole (1200-rpm) motor. Occasionally, you will receive a proposal requiring an 8-pole (900-rpm) motor, but unless you have that size in your stock, these are to be avoided, because of cost and because of the weight of the motor on an integral agitator frame. Such a motor in any size may weigh 50% more than a comparable 4-pole motor.

1. Geared speed reducers

(I personally refrain from the term "gear reducer." Hammer mills, jaw crushers and rod mills reduce gears; I trust that your gearbox only reduces speed!) A single reduction gear box is available in ratios that will give several different output speeds from 420 rpm to 230 rpm in AGMA steps from an 1800-rpm motor. In large agitators where the drive speed must be less than 230 rpm, a 1200-rpm motor with a single-reduction gearbox can be used. On separately driven (not integral) agitators, a parallel-shaft double-reduction gearbox can be used with the lower-cost 1800-rpm motor. Properly-maintained, gearboxes are reliable, provide long life and deliver constant speed—good solid reasons for considering this method of speed reduction. Even so, I don't recommend geared speed reducers below 150 horsepower or until the agitator size and output speed make the selection of a belt drive impractical.

The first objection to a gear drive has to be the availability of output speeds. The standard AGMA ratios, 190, 230, 280, etc., leave large gaps of speed choices that can hardly be compensated for by pitch changes. Based on the cubed relationship of speed to horsepower, the power differential between one standard ratio and the next averages 75-80%. The range of power adjustment from an adjustable pitch propeller is only ± 25-30%. Even with the optional ratios available from some of the gearbox manufacturers, the power change is still 30-40%. We "luck out" occasionally in finding just the right conditions for loading at a standard speed, but using geared speed reducers often requires particular care in matching impeller requirements to both process requirement and power response.

The second objection I would suggest is cost and availability of spares. Initially the gearbox is more costly than a belt drive. If it isn't an integral drive but a separate parallel shaft reducer, there are two additional couplings (flexible couplings—never use a gear coupling on an agitator) in the drive train. Spare parts are cost intensive if you invest in a spare set of gears; if you don't, you may invite prolonged downtime waiting for a new set to be delivered. You also have to be concerned about an additional set of bearings.

Finally, someone years ago, commenting on the basement of a paper mill, paraphrased the *Rhyme of the Ancient Mariner* by saying "Water, water everywhere, and most of it in the speed reducer!" It's true; we do splash a lot of water in the vicinity of a lot of critical equipment. Some of it is accidental, some of it just by inattention during a routine washdown. Water leaking into the breather of a gear box is deadly! I remember a particularly disastrous situation in a Midwestern mill in which, by some combination of events, most of the non-integral speed reducers selected exceeded the thermal rating of the gearbox. The maintenance group piped cold water lines to each location and allowed water to pour over each box to keep them cool. You can imagine the bearing life which was experienced.

Nevertheless, there are applications for which a geared-speed reducer is the only practical approach. Treated as a precise piece of machinery, it will reward you with long and satisfying service. The minimum service factor to be used for agitator service is 1.5, based on motor horsepower. When used for under-machine service such as in the couch or press pit, you may want to increase that to 1.75 or 2.0 because of the possibility of shock loading.

2. Belt drive-speed reducers

The belt-driven agitator is the most commonly accepted design offered by most major agitator manufacturers. Prior to the introduction of the 3V, 5V, and 8V section belts and sheaves, and now the even more efficient toothed timing-belt style, belt drives weren't as easily adapted to this service. Flat belts were inadequate or at best cumbersome for all but the lowest horsepower installations. These were sometimes forcibly integrated with geared-speed reducers in order to reach particular ratios. The A-, B-, C-section belts gave a welcome boost to the use of simple single-reduction drives, but still the capacities were low, requiring 10- and 12-belt combinations which were difficult to tension properly and were often limited to low-speed motors. Advances in the capacity of belting and the design of these newer drives quickly covered the reduction ratios and horsepower requirements of all but the largest of agitators. When properly applied and maintained, the modern belt drive solves most of the problems associated with the geared speed reducer.

The ratios available with 1800- and 1200-rpm motors allow the selection of nearly exact speeds well within the limits of pitch change for an adjustable pitch propeller and close enough for most fixed-

pitch installations. Drives can be designed so major changes in speed and horsepower can be easily made by the simple substitution of a different size driver sheave and a minor adjustment of the center distance. Spare sheaves and belts (matched sets) are readily available from local stocks and are inexpensive enough to warrant stocking mill spares.

V-belt drives require attention. Periodically, the tension of each belt in a set should be checked. Most manufacturers can provide a tension tester that makes this operation relatively simple. The old A-, B-, etc. section belts are so far behind us now that few experienced millwrights remember the old method of pushing a belt with their thumb and saying "That's pretty good." The proper tension of today's belts is an exact amount of deflection for an exact force in pounds. When properly set, these don't slip or squeal and carry their portion of the load. If inadvertently sprayed with water they are affected, but the heat of transmission will quickly dry them back to normal operation. (This doesn't mean *carte blanche* to routinely shower V-belt drives.) Care must be taken in selecting the driver sheave size in relation to motor speed and horsepower. Drive suppliers list nominal minimum sizes for standard motors, but in borderline situations, it's best to check with the motor supplier for exact limits of overhung load.

The minimum service factor for agitator service should be 1.5 based on motor horsepower, but again, if shock loads are anticipated beyond normal hydraulic surges, you may want to use 1.75 or 2.0.

3. Other speed reduction devices

There are other methods of reducing output speeds, but none, other than these two just mentioned, are found in normal agitator applications. There are other types of belts, some of which use a single strand of extreme capacity and require exceptional tension. The simplicity and effectiveness of today's standard belts relegate the others to infrequent use. It would be possible to design a chain and sprocket drive for an agitator, but the surging and occasional shock loading inherent in agitator service would make these drives a very troublesome installation. The "high-tech" DC and synchronous drives, and also hydraulic drives, are just too expensive to be considered for routine agitator service. They are best left to applications which control running speeds on the paper machine.

Accept V-belts or a geared-speed reducer within the parameters discussed, but don't be satisfied with a one-line description such as "... complete with our specially selected QXY Superdrive." Insist on full specifications of any drive offered so you may plan ahead for proper maintenance.

B. Propeller design

General

Most of the impellers used with agitators in paper stock for the past few decades have been some form of the three-bladed marine-form propeller. The most usual identifying characteristics are the three blades, a fixed-pitch angle of 18 degrees and a developed area ratio (DAR) of 0.45 to 0.50. The adjustable-pitch design uses the same area ratio but allows pitch changes from a low of 14 degrees to a high of 22 degrees. Because of the lead angle of the marine-form propeller, pitch angles of less than 14 degrees aren't recommended for service. Propellers of this design will exhibit the power number (Np) of 0.36± discussed in earlier chapters, at the square pitch of 18 degrees. Some manufacturers, especially some of the earlier designs used for midfeathers and the early mining nozzle equipped high-density towers, promoted wide-blade designs and extreme-pitch angles. A study of the momentum number data clearly shows that the efficiency of the propeller decreases rapidly with increase in pitch and power number. High-pitch angles do increase pumping capacity but at the expense of high torque and lower Mo.

1. Adjustable pitch propellers

The principal advantage of the adjustable-pitch propeller lies with the supplier because of the ability to alter loading after the fact without resorting to a speed change (of course, you have to empty the chest to do it). The disadvantages all relate to disasters that can occur because of poor design and, though the supplier may supply a new propeller at no charge, the downtime and lost production are undeniable yours!

The strength of the propeller blade is an important factor in the design of an homologous series of propellers. Each size must be designed to withstand the forces imposed upon it at any pitch angle and at any speed and horsepower combination. You cannot simply increase the speed to absorb more horsepower to meet a higher process requirement without knowing the limits of the blade strength. At the time we were investigating the strength of propeller blades in 1968, we were surprised to discover that there was very little in the literature until we talked with the marine engineering faculty at Stevens Institute, Hoboken, N.J. Most of the in-depth studies had been done on ship propellers. With their help, we were able to adapt the work of J. E. Conolly, "Strength of Propellers" *(27)* and D. W. Taylor, "The Speed and Power of Ships" *(28)*. The agitator propeller is a very special case, as it represents a "vessel with zero wake"—as we were told, akin to a tugboat which must develop extremely high power at virtually zero forward speed. I cannot present a full exposition of those design equations in these few pages, but the supplier of your choice should be familiar with these methods.

The method of fixing the propeller blade in one position is also important. Over the years, the use of set screws providing a force normal to the twisting moment of the blade at the shank has been inadequate and, depending on the profile of the blade, often disastrous. One manufacturer finally settled on a locking taper for the blade shank with set screws providing a force opposed to the centrifugal force of rotation which has proved quite satisfactory.

The blade profile should also be considered in the design of the blade to prevent disastrous failure. The old style concentric "paddle blades," in the event of movement due to a locking failure, invariably twisted to a full 90-degree pitch which immediately meant extreme overload—locked rotor—shut down! That was the "good news," if all three blades went together. If only one or two moved, severe vibration would set in leading to a bent shaft and, sometimes, rupture of the chest wall. "What do you do with five ft. of groundwood pulp in the visitors' parking lot?" A blade design using a retreating trailing edge, if the locking method fails, forces the blade to 0 degrees pitch setting. Certainly the efficiency of the agitation falls to nothing, but the evidence is shown on an ammeter or a consistency chart—not by a hole in the chest wall!

2. The Prochem "Maxflo"

This is the proprietary impeller designed and manufactured by Prochem Ltd. of Brampton, Ont. It has been irreverently referred to by scoffers and competitors as "the Mickey Mouse." However, since its first appearance, ca. 1967, it has made believers out of the scoffers and has been copied by many competitors who finally realized the advantages and efficiency of the low-pitch, airfoil-style impeller.

It's a unique design and, to set the record straight, it isn't a propeller. It may look like one and it is an axial-flow impeller, but a propeller doesn't "propel" at zero pitch or even close to zero pitch—the "Maxflo" does. (It must be said that since

this book was begun, several newer designs of the "Maxflo" have emerged.) The three blades (four in special situations) are designed similar to the airfoil of an aircraft wing. Motion of liquid or, in our case, stock slurry is produced by the difference in pressure between the back and front of the blade, in the same manner that an airplane wing provides lift. At zero pitch (zero angle of incidence), differences in pressure are still present, just as an airplane can fly "straight and level," and produce axial pumping action. As we have noted previously for any impeller, the lower the pitch, the lower the power number, the higher the efficiency and the higher the QV or momentum produced. The "Maxflo" impeller is usually operated at very low-pitch angles for high efficiency at relatively high speeds (low torque). It's almost always furnished as a fixed-pitch impeller (adjustable-pitch is available but seldom used) with the blades welded to the reinforced nose cone. Any change in process conditions requiring a change in power response will necessitate a change in operating speed, usually by the substitution of a different diameter driver sheave. I don't have the stress and design calculations available for this impeller but, from my experience, it has performed quite well in service.

C. Shaft design

Agitator shafts can present some rather complex design problems. Unless you have specifically requested all of the dimensional information discussed at the beginning of this chapter, you will have to rely on the integrity of the supplier for the safe design of your agitator shaft. Even with the usual data of diameter, length and bearing spacing, many of the unique stresses in an overhung agitator shaft will still be unsolvable without an understanding of all the forces applied. Some shafts become more complex when stepped shafts are used, which is common with agitators driven by direct connected gearboxes and utilizing oversized (sometimes undersized) reducer shafts. The design of the shaft isn't a simple matter of a safe critical speed ratio and an allowable torque. There is a fluid force which is normal to the centerline of the shaft which creates a bending moment. The weight of the impeller adds to the bending moment and a thrust occurs parallel to the shaft, creating a column effect—all in addition to the twisting torque and the consideration of the approach to the natural frequency of the system.

For this discussion, we shall limit ourselves to a straight-through solid shaft, equal diameter end-to-end and a shaft material of steel or a steel alloy, $E = 30 \times 10^6$. Figure 10-1 illustrates the dimensional data required and the location of forces to be used or determined.

Where:

La = Shaft length 1st bearing to C/L of impeller—ins.
d = Shaft diameter—ins.
L = Impeller clearance from wall—ins.
a = Bearing spacing—ins.
D = Impeller diameter—ins.
C = Distance C/L 2nd bearing to C/L driven sheave—ins.
F = Fluid force—lbs
We = Impeller weight—lbs
Fa = Thrust—lbs
B = Belt tension—lbs
Ws = Weight of drive sheave and hub—lbs
St = Torsional stress—psi
Ts = Torque—in. lbs
Nc = Critical speed—rpm

1. Thrust Fa

The thrust of the propeller is calculated from the empirical formula:

$$Fa = hp \times 33{,}000 \times 0.65/\text{pitch} \times rpm \quad (1)$$

Where: 0.65 = Efficiency factor for propeller
Pitch = $\pi \tan \theta \times (D-1)/12$
θ = Pitch angle
D = Diameter—ins.

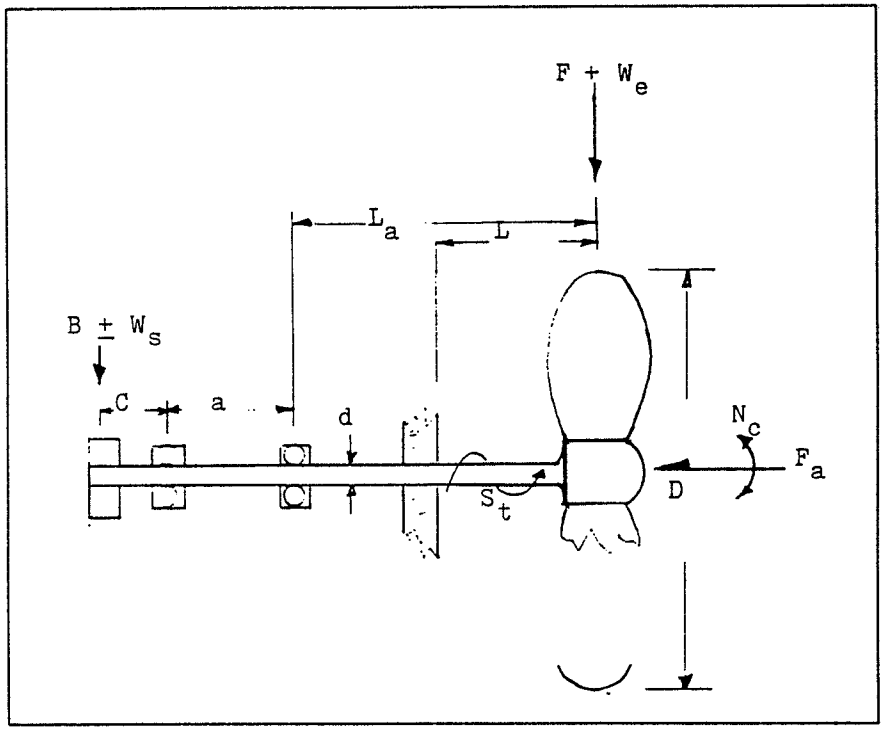


Figure 10-1. Shaft Design Dimensions.

An alternate method for calculating thrust, which is satisfactory for all cases since thrust has a small effect on combined stress, may be used from the formula:

$$Fa = 0.97 \times Mo \quad (2)$$

Where: Mo = Momentum number for the particular speed, diameter, hp relationship.

2. Fluid force F

Hydraulic forces in the agitated volume act upon the impeller, constantly changing directions. To calculate the maximum bending moment to which the shaft will be subjected, we must assume a point value of that force acting normal to the centerline of the shaft at the impeller. Normal practice is to add the weight of the impeller to that force. There are various conditions which will drastically change the fluid force on the impeller. If the stock level is constant and the impeller is always sufficiently covered to prevent swirling, the force will remain at a minimum level. If some vortexing is allowed as during drawoff, the force will become significant and produce a major increase in the combined shaft stress. If the level were maintained for any length of time at the centerline of the shaft so that violent surging and vortexing occurs, the force would be extreme—a factor of perhaps 2.5 times that exhibited during mild vortexing. For the usual calculation of combined stress, we assume the intermediate case and prohibit operation for any period of time at

the air/slurry interface. For this condition, we calculate the fluid force by the formula:

$$F = 1.20 \times 10^{-7} \times Np \times N^{1.67} \times D^{3.53} \quad (3)$$

Where: Np = Power number at pitch angle used

N = Rpm of agitator shaft

D = Impeller diameter—ins..

3. Maximum bending moment M

The force creating the maximum bending moment which the shaft will be subjected to is that fluid force just calculated, assumed to be acting vertically downward normal to the shaft centerline at the impeller tip, plus the weight of the impeller. The moment arm will be the overhung length of the shaft from the impeller centerline to the centerline of the first bearing, La. You will notice a distance "L" noted on Fig. 10-1, "impeller centerline to the inside wall of the chest." This dimension doesn't affect the bending moment calculation, but is a critical dimension relating to the process performance of the unit. It has been determined that a particular minimum area between the impeller (thrusting outward) and the chest wall is necessary in order to develop the full horsepower and required flow pattern. The distance in water-like materials was established as ⅓ the diameter of the impeller. For paper pulp slurries and most materials, a distance of ½ the impeller diameter has been recommended. Clearances greater than this don't improve process performance and only increase the bending moment to which the shaft is subjected. The maximum bending moment is calculated from:

$$M = La \times (F + We) \quad (4)$$

4. Critical speed—Nc

The critical speed (first natural frequency) of any shaft and impeller combination is related to the overhung length of the shaft from the first bearing (La), the diameter of the shaft (d), the bearing spacing (a) and the weight of the impeller (We). Assuming, as we did in the beginning, a Young's Modulus of 30×10^6 psi for all shaft materials used, we won't make a further correction for strength. Also, we assume a rigid mounting and don't make allowance for the contribution of a flexible support, such as a vertical unit mounted on beams across the chest.

The allowable approach to the actual critical speed is one of judgment, type of unit and location. As the operating speed approaches the first natural frequency, the tendency for the shaft to deflect becomes increasingly greater, as you may have observed in the literature describing the "magnification factor." In chemical mixing applications using vertical mixers, centrally mounted in fully baffled tanks, it's possible to approach to within 80% of the calculated critical speed, No/Nc = 0.8. In special applications such as off-center locations, high gas rates or extremely high power inputs, the limiting value is proportionately reduced. For side-insert applications, especially in paper pulp slurries, we reduce this limiting ratio even further. We have gravity working against us in the weight of the impeller and want to limit the possible deflection at the packing box. We also have the possibility of severe surging and even shock loading. We shouldn't allow the critical speed ratio to exceed 0.25 and should keep it lower whenever possible. (Note: This is another reason to maintain the "L" dimension at ½ impeller diameter.)

There are as many different ways to calculate the value of the first natural frequency as there are suppliers. To observe the differences between them, I made a rigorous analysis of all the equations within my knowledge, including the basic one found in *Uhl & Gray*, Volume II *(31)*, and was able to reduce each one to the simplified version published by Chemineer in *Chemical Engineering*, 1965 *(30)*. Using nomenclature from Fig. 1-10 and eliminating the Modulus Factor for the reasons given earlier, that equation becomes:

$$N_c = \frac{1.681 \times d \times 10^6 \times ((L)/(L_a + a))^{.5}}{L_a^2 \times \sqrt{1 + (19 \times We)/(La \times d^2)}} \quad (5)$$

5. Torsional stress—St

This is straightforward from any engineering handbook and is incorporated in our final equation for combined stress. However, you may want to check the torsional stress at the reduced diameter of the shaft at the drive end

$$St = (321{,}000 \times hp)/(d^3 \times N) \quad (6)$$

Where: hp = The value of the installed motor horsepower

6. Torque—Ts

This is a straightforward calculation found in any handbook. We require this value for the final combined stress calculations:

$$Ts = (63{,}025 \times hp)/N \quad (7)$$

Where: hp = Installed motor horsepower

7. Combined stress—Ss, St

We need to look at two values to determine the safety of the shaft design *(29)*.

Ss = Maximum combined stress in shear—psi

St = Maximum combined tensile stress—psi

Different manufacturers may apply different limits for these values in their designs. Some even allow different limits depending upon application. For the rugged continuous duty we experience in the paper industry, I recommend a maximum of 5000 psi for Ss and 10,000 psi for St with the greater emphasis on the value of Ss. The calculations are made as follows:

$$S_s = \frac{16}{\pi d^3}\sqrt{\left(M + \frac{Fad}{8}\right)^2 + T_s^2} \quad (8)$$

$$S_t = \frac{16}{\pi d^3}\left(M + \frac{Fad}{8} + \sqrt{\left(M + \frac{Fad}{8}\right)^2 + T_s^2}\right) \quad (9)$$

If you're familiar with these equations from Ref. 29 or any other source, you may recognize that I have neglected the factor "α" which is a multiplier on the term (Fad)/8. This factor relates the maximum intensity of stress from the axial load to the average axial stress and is calculated by:

$$\alpha = 1/((1 - ((0.0044 \times a)/k))) \quad (10)$$

Where: a = Bearing spacing—ins.

k = Radius of gyration of the shaft—ins.

The effect of thrust on the combined stress is so minimal that it was felt this added factor only complicated the formula for our use.

8. Drive end

We won't go into an analysis of the drive end for the following example, but it's obvious a similar selective analysis may (and should) be made. There's no thrust component, but the belt tension becomes the "fluid force" now identified as "B" and, in those cases of an integral agitator with the motor mounted on the top of the unit, the weight of the driven sheave may be deducted from "B." In all other cases, Ws would be a vectored force to be incorporated with "B."

9. Bearings

Bearing loads and bearing life may be calculated in the usual manner from the forces derived from these formulae and the usual handbook references.

Now let us go through one complete example using these data:

An agitator has been recommended having the following specifications:

Installed horsepower—100
Operating speed, N—197
Impeller diameter, D—54 in. @ 17°
Shaft diameter, d—4½ in.
Overhung length, La—41⅜ in.
Distance off wall, L—27 in.
Bearing spacing, a—29⅛ in.
Drive end overhand, C—10½ in.
Impeller weight, We—497 lbs
Power number @ 17° Np—0.338

1. Thrust—using Eq. 1

$Fa = 100 \times 33{,}000 \times 0.65/\pi \times \text{Tan } 17° \times 53/12 \times 197$

$Fa = 2567$ lbs

2. Fluid force—using Eq. 3

$F = 1.2 \times 10\text{-}7 \times 0.338 \times 197^{1.67} \times 54^{3.53}$

$F = 359$ lbs

3. Maximum bending moment—using Eq. 4

$M = 41.375 \times (497 + 359)$

$M = 35{,}417$ in lbs

4. Critical speed—using Eq. 5

$$N_c = \frac{1.681 \times 4.5 \times 10^6 \sqrt{\dfrac{41.375}{41.375+29.125}}}{41.375^2 \sqrt{1+\left(\dfrac{19 \times 497}{41.375 \times 4.5^2}\right)}}$$

$N_c = 966$ rpm

$No/Nc = 0.20$

5. Torsional stress—using Eq. 6

$$St = 321{,}000 \times \frac{100}{4.5^3 \times 197}$$

$St = 1788$ psi

6. Torque—using Eq.7

$Ts = 63{,}025 \times 100/197$

$Ts = 31{,}992$ in. lbs.

7. Combined stresses—using Eqs. 8 and 9

$$S_s = \frac{16}{\pi \times 4.5^3} \sqrt{\left(35{,}417 + \frac{2567 \times 4.5}{8}\right)^2 + 31{,}992^2}$$

$$S_t = \frac{16}{\pi \times 4.5^3} \left(35{,}417 + \frac{2567 \times 4.5}{8} + \sqrt{\left(35{,}417 + \frac{2567 \times 4.5}{8}\right)^2 + 31{,}992^2}\right)$$

$Ss = 2728$ psi(N.B. if the "α" factor, Eq. 10 had been included, $Ss = 2736$ psi a negligible increase)

$St = 4788$ psi

In this example, all calculations are well within the limits discussed. If you had used Eq. 2 for thrust, Fa would have equalled 2016 lbs or 79% of Eq. 1. Why? Because Mo is based on the true impeller horsepower 80% of installed motor horsepower, therefore the agreement is nearly exact!

D. Shaft closures and shutoffs

1. Shaft seals

We spoke briefly about packing boxes earlier in this text. One of the "advantages" of the vertical unit was because a packing box wasn't required and "Everybody knows that packing boxes leak and are a maintenance headache." This was before the superior process advantages of the side-insert agitator were fully exploited. Also, those who "beat the drum" for vertical units without shaft seals conveniently neglected the more aggravating problem of the necessary steady bearing and the physical damage it could produce when a bushing seizes. (I know that some suppliers now recommend vertical units without steady bearings in selected operations. I strongly feel that the possibility of mechanical disaster isn't worth the risk, where a vertical unit is necessary for some physical reason.)

But a shaft seal can be designed and maintained to produce near zero leakage; it's simply a matter of the manufacturer's design and the mill's maintenance philosophy.

Shaft seals, as a generic term, can be divided into two categories. What we generally refer to as a "mechanical seal" is properly called a "rotary seal" in which a rotating face affixed to the shaft or sleeve runs against a fixed face in the seal chamber. This type of seal is commonly used on pure liquids in the "single-seal" configuration and lubricated by the process fluid. In the "double-seal" configuration, a separate fluid is maintained between the two rotating faces and is under sufficient pressure to keep process fluid from the chamber. The second category, generally referred to as a "packing box" is properly called a "packed seal" and consists of a similar chamber into which we have "packed" a number of rings of a pressed fibrous material (asbestos, Teflon–, etc.), held in place by a "follower" to maintain pressure on the rings. In liquid systems, lubrication is provided through a "lantern ring" usually located in the center of the chamber. The lubrication may be grease or some circulating fluid compatible with the process fluid.

It is important to go back to the basics and remember that, regardless of the type of sealing mechanism, neither one will work without a fluid film between the stationary and rotating elements. In the rotary seal, even though the two faces are lapped to light bands of flatness, they don't in themselves affect the seal. The lubrication fluid in the seal chamber (or process fluid in a single seal) must exist as a thin film between the faces to affect the closure and maintain the pressure in the system. For the packed seal, the same mechanism is present. The packing rings don't affect the seal. If these are run dry and simply squeezed harder in an attempt to prevent leakage, they will slowly act as a parting tool and groove the shaft or physically cut the shaft sleeve down to bare shaft metal! The packing is really just a sponge which holds excess lubricant, always providing a liquid film between the rotating shaft and the stationary rings. Excessive pressure on the packing follower, rather than decreasing leakage, will actually break the film, allowing the packing to burn and thus increase leakage.

For paper pulp agitator applications, the rotary or mechanical seal has never become very popular. Fibrous materials are hard to seal against with this type of seal. A double-seal with a flushing gland ahead of the inboard seal could work nicely, but the expense is prohibitive. Some are being used today, but they aren't universally accepted. There have been many modifications to the historical "lantern ring" style of packing box as applied to pulp slurries. Most designs today substitute a throttle bushing for the lantern ring and place that bushing in the front of the box, followed by four or five rings of packing. Lubrication, usually clean water, is tapped to the bushing, which is grooved on the inside diameter. Because of a very close clearance lip on the inboard end of the bushing, most of the lube water flows back through the box along the shaft, maintaining a film between the packing and the shaft sleeve. When properly installed and maintained, water flow at about 5 *gal/h* and pressure about 10 psi above the head in the chest, should result in a minimum drip of clear water at the gland follower. Excessive pressure, higher volumes of sealing fluid, or both, will simply "chew out" the packing, resulting in excessive leakage and fiber loss.

2. Shutoffs

Eventually, even with the best of care, a packing box will require repacking. How do you accomplish this with a chest full of stock? Contrary to the simplistic advertising claims of all my friends in "the stirring business," the safest and surest way to repack a packing box is to DRAIN THE CHEST FIRST! The box can be completely cleaned, the shaft and impeller can be inspected for damage (which might have caused an early failure of the seal) and new packing can be run in and checked without the urgency if you were trying to immediately get back on line with a full chest.

This idealistic, but safe, recommendation cannot always be followed. Where can you put 500 tons of high-density pulp while you service a packing box? There are certainly other large storage chests and

other critical installations that cannot arbitrarily be shut down and emptied just to service a badly leaking shaft seal. Of course, preventative maintenance might minimize or eliminate all but the most unexpected failures. Experience and maintenance records can predict the frequency of necessary service and critical units can be repacked during routine "downs." But there are still those situations, too often to be ignored, that require something other than convenience and good maintenance practice. A way of "shutting off" the packing box against a full head of stock while it is opened and repacked is unavoidable.

Every major supplier "touts" his own design for this undeniably desirable feature. There are really only two types of shutoffs, though variations on the theme are numerous. They are (1) a compression seal affected by moving the shaft outboard to bring a fixed collar with an "O" ring, or flat seal, flush against the inboard face of the packing box flange and (2) a fixed chamber at the flange face containing an inflatable diaphragm which can be inflated by air pressure to seal off the box.

Take your choice, because either one will effectively shut off the packing box—IF routinely inspected, installed properly, operated properly and, most importantly, returned to running position properly after the repacking procedure has been completed. This is a little like having a penicillin shot to get rid of pneumonia. I know it's good for me. I know it's going to sting. I'm sure I'll have a violent reaction and—how did I get into this mess in the first place? Regardless of the manufacturer or the type of shutoff you choose, don't kid yourself; these are dangerous to your "health" in the hands of an amateur. Let's look at the things that can happen, or to be positive, look at the pitfalls good practice can avoid:

The compression-type seal

Most of these seals require the loosening of a separate housing on the fixed bearing so some method of jacking the shaft backwards will bring the "O" ring or flat seal flush with the face of the flange. (The pressure head in the chest will probably do this without much help from you. "Jacking" is required to put it back in position.) The travel is so slight that the radial bearing just moves within its normal "float." If the collar or sealing ring has become damaged by some foreign object in the chest since the last inspection, you won't know it until you've got the gland follower on the floor followed by a part of the chest contents in your lap. Some manufacturers include a test tap to check for leakage before pulling the gland. This is a good idea. After repacking and fixing the gland follower finger tight and cracking the water valve to wet the packing, someone in a hurry forgets to return the shaft to the run position or forgets to tighten the fixed bearing housing. Zap! The seal is gone and maybe worse.

(2) The inflatable seal

This is the simplest of the two seals—perhaps the most effective but also equally susceptible to error. The seal is always ready to be activated; the shaft or bearings don't have to be moved. We just bring an air line—a bicycle pump is safer—to the designated fitting and inflate the seal. Again, unless that test tap is available, you won't know if the seal is working until it's too late. Assume so far, so good. Repack the box, wet the packing and—forget to deflate the seal! Zap again! The seal is gone, but at least the unit is running and maybe you'll have a long "down" before it needs replacing again. Another danger with this type of shutoff is the proper identification of the air tap. The first one I ever had to actuate was on a high-density tower and, when I got to the unit, I found a ⅜ in. copper line permanently connected to the fitting. Tracing that back to its source, I found 40-psi water had been going through the seal since startup. We didn't repack the stuffing box that day!

E. Materials of construction

Most suppliers will ask, "What do you want?" In the chemical industry, this is certainly a viable question. The supplier isn't a metallurgist and any recommendation he might make will certainly open him up to charges, if not a law suit. The user is more familiar with the environment of his system and should be able to specify the materials of construction acceptable for the wetted parts of the equipment. In the paper industry, we generally do not have the exotic chemical mixes our brothers in the process industries must contend with. Aside from the bleach plant, our biggest concern is with the corrosive effects of water and lightly *acidic/basic* systems of perhaps pH values of 2 to 11. If we are only manufacturing unbleached grades, carbon steel or ductile iron will serve our needs without much of a problem. However, critical areas such as shaft sleeves, packing boxes and impellers are subject to oxidation (rust) and we must make a decision about what we will accept. Impellers of ductile iron and shafts of carbon steel are sometimes acceptable in this service, although a T316 SS sleeve at the packing box and a T304 SS faced flange and gland housing makes for longer system life.

At one time, there was a 10% difference between T304 and T316 SS castings and shafting, which led many suppliers and their foundries to standardize on T316. This probably is still good insurance against extreme values of pH and temperature, but the margin between these two alloys is now so great it leads to a reevaluation of what is required. In bleached grades and other fine papers, stainless steel wetted parts are a necessity. The individual mill must decided on whether the present premium for T316 is warranted. There have been some experiments with coated impellers, such as epoxy or sprayed plastic coating, and even the complete casting of an impeller in some type of high-strength plastic. In my experience, coated impellers lose their coating down to the base metal with disastrous results. Solid-plastic impellers and even shafts are expensive (and I will leave those up to your own judgment).

In the bleach plant, we have an entirely different environment. I have offered, at the mill's request, T317 SS in preference to T316 and wondered at the exorbitant difference in price for the infinitesimal difference in metallurgy. (I'm just an old chemical engineer reading a metals handbook.) I have also offered high-nickel alloys for this service and felt the extreme expense was justified. (Perhaps the latest trend away from the chlorination stage may change this emphasis.) Again, the supplier isn't a metallurgist—you are calling the shots for materials, but don't be surprised at the price of anything beyond the usual grades of 300 series stainless steels.

I believe hardened sleeves for shafts at the packing box area are a fetish. We don't have pressures in an atmospheric chest that warrant the expense nor do we want to tolerate the risk of fracture that accompanies the shrinking on of a T440 SS or stellite sleeve. Pressurized refiners are a "different animal" and generally use sleeves of some hardened alloy, but we're talking about agitators! Even when we consider pulpers, whether these are batch, continuous or of the under machine variety, the industry seldom supplies anything more exotic than T316. A typical metallurgy for a dry-end pulper on a linerboard or newsprint machine would be T304 SS for the elbow and flange facing, T410 SS for the rotor and impeller ring and perhaps T316 SS for the extraction grate. The T410 is used primarily for its hardness during abrasive service. (And sometimes it's more trouble than it's worth since a 300 series, though softer, is more easily repaired.) T410 SS will show rust spots if exposed to air for any length of time, but because it is generally submerged and in

continuous rotation—the pulping action keeps it clean.

The "on-the-shelf" agitator metallurgy is also pretty plain. It involves shafts of T304 SS (some suppliers have used T303 for its machinability and lead content which precludes welding), propellers of T304 or T316, sleeves of T316 and generally T304 trim, flange facing and packing boxes. This will vary slightly from house to house. Some suppliers having a captive foundry still prefer to pour all their castings in T316.

In summary, the client usually knows what he needs, and after some 30 or 40 years of experience, that need is pretty well matched by the supplier's standard design. There are some special materials to be considered and most suppliers are ready or willing to supply them, at some extra charge. Hardware includes 17-4PH, pulping impellers of 17-4PH; facing, shafts and propellers in T317 (be prepared for an extended delivery) and, of course, carbon steel and ductile iron where acceptable. There are some exotic packing materials, but Teflon-impregnated asbestos is still a very popular and available material. Paying a little more for precut sets, including Teflon separators, is a wise investment as mentioned at the beginning of this chapter.

Chapter 11: Major Suppliers

The only fair way to present a list of suppliers of agitation equipment for pulp and paper slurries is to list them alphabetically. The list is somewhat abbreviated, as I am including only those I am familiar with on the North American continent. There are agitators made in Europe, the Scandinavian countries, and perhaps in the Soviet Union; but I doubt your exposure to those designs represents a significant part of the North American market. There are also several other manufacturers of mixing equipment in the United States who penetrated the paper industry by way of coating kitchens, clay storage, pulp mill recovery systems, etc. But they have not marketed a pulp agitator or have only done so on a very marginal basis.

A. Names

The major suppliers of paper pulp agitation equipment are:

1. **Beloit Corporation, Jones Division**
   401 South Street
   Dalton, Massachusetts 01226

2. **The Black Clawson Company**
   Shartle-Pandia Division
   Box 160
   Middletown, Ohio 45042-0160

3. **James Brinkley Company, Inc.**
   1001 South Weller Street
   Seattle, Washington 98104

4. **HYMAC Ltd.**
   P.O. Box 434
   Laval, Quebec, Canada H7S1V9

5. **Ingersoll-Rand, Impco Division**
   150 Burke Street
   Nashua, New Hampshire 03061

6. **Mixing Equipment Company, Inc.**
   "Lightnin Mixers"
   P.O. Box 1370
   Rochester, New York 14603

7. **Prochem Ltd.**
   35-190 Hwy. 7W
   Brampton, Ontario, Canada L7A1A2

8. Voith-Morden, Inc. (now Voith, Inc.)
2003 N. Meade Street
Appleton, Wisconsin 54911

In all cases, I wanted to name the contact person most closely related to the sales and application of pulp and agitation equipment. However, given writing dates and publication dates, you all just move too fast! Let's now comment briefly on the background of each company. Everyone has had excellent experience with a particular supplier and, likewise, some disasters. Strangely enough, these roles may be reversed in someone else's mill. There are those of us who will swear by brand "X," and others will just swear! Perhaps a little background will help you with future choices.

B. Background

1. Jones Division, Beloit Corporation

Certainly an old line company. Completely integrated stock preparation equipment company; formerly E. D. Jones & Sons prior to acquisition by Beloit in 1959. A complete redesign of agitators, mechanical and process application in the years 1963 to 1973, during the writer's tenure.

2. The Black Clawson Company, Shartle—Pandia Division

Another old-line company with a completely integrated stock preparation division. Made the name "Hydrapulper" a household word, much to the chagrin of many who found their pulpers relabeled by the mill as "Hydrapulper #1," etc. Agitation line recently revised mechanically. Application data may still be lagging behind small, aggressive competitors.

3. James Brinkley Company

A company that's been around a long time and pretty well known for its "Hell for stout" construction. It's not very pretty, but it gets the job done and keeps running. Products are mostly agitation equipment, materials handling, guillotines, and some specials. Application data and much design standardization done in years between 1974 and 1976 during the writer's tenure at Voith-Morden (see Item 8).

4. HYMAC Ltd.

A respected Canadian company, manufacturer of stock prep machinery. Agitation equipment sold widely in Canada, fair penetration in U.S. market. I confess to little information on their application expertise.

5. Impco Division, Ingersoll-Rand

Who doesn't remember the "Commander" Arthur Whiteside? One of the early "giants of agitation" in the heyday of midfeathers and vertical circulators. Impco has always been a powerful force in stock preparation, especially in pulp mills and bleach plants. Agitator line now offers a standard side-insert unit and updated application data. Still offers some of the "golden oldies" if required.

6. Mixing Equipment Company

Where the revolution started. First successful controlled-zone agitation. Initiated the reduced-bottom, high-density tower design. First to recognize, quantitatively, individual stock factors for all types and grades of pulp. This company was the beginning of this writer's career. The leading manufacturer of mixing equipment for all the process industries. Manufacturer of all types of mixers and agitators for the paper industry, but no allied stock preparation equipment.

7. Prochem Ltd.

Relatively new company, ca. 1967. Very aggressive company, developer of the "Maxflo" impeller and pioneer of the momentum theory of application. A Canadian company marketing through sales representatives in the U.S., expansion begun during the writer's tenure, 1971-72. Manufacturer of a complete line of agitators and mixers for all process industries. No other stock preparation equipment.

8. Voith-Morden, Inc. (now Voith, Inc.)

Nee Morden Machines Company of Portland, Oregon. Merged with Voith GmbH in 1974. Complete integrated stock preparation machinery company. Famous for the "Slush maker," a high-intensity pulper, still available as an alternate to the "Brute," agitators are built by James Brinkley Company and marketed throughout the United States with the exception of Oregon and Washington State. (See Item 3 for agitator comments.)

C. Evaluation

You now have five or six proposals in front of you, and you need to make a decision on which of these offers the best and safest buy for your company. If you have read everything that has gone before this page, use the data provided and find out how many of the bidders "know what they are doing." Throw the rest away after a short letter of condolence. Make your selection on the best balance of capital cost, energy consumption, and your own experience. If you've just opened this volume to this page, you've got a problem. Well, perhaps we can simplify it a bit.

Sometime back, we introduced the concept of momentum, and we said the momentum number, Mo, was equivalent to:

$$Mo = CN^2D^4$$

Where: N = Operating speed—rps
D = Impeller diameter—ft
C = An efficiency factor for the impeller.

Let's first calculate N^2D^4 for all the bids, neglecting the value of "C," *(12)*. For standard marine-form propellers at 18-degree pitch, the value of "C" will essentially be a constant (0.48-0.50). For adjustable-pitch propellers, between 14 and 22 degrees, the value of "C" can only change ± 9%. So those bids that give similar values of N^2D^4 are reasonably close to your process requirement. If the supplier has told you the pitch angle, which you should have asked for, so much the better; you can alter his N^2D^4 value roughly in a straight line, 22 degrees being 9% higher and 14 degrees 9% lower. (Wish you'd read the whole thing?) If one of the bids includes the Prochem "Maxflo" impeller, you can still make a rough guess. The "Maxflo" will most likely be at a much lower pitch than 14 degrees, so if we were to use C - 0.486 for 18 degrees on a fixed-pitch propeller, an average value of C = 0.44 will suffice for the "Maxflo."

Now, with rough estimates for momentum number for all the bids, you can still lump those closest to an average and make a decision. Now you've narrowed the field to three or four bidders. My next concern would be, how close is the nearest representative and how qualified is he? A field representative has often been humorously described as a man with a telephone credit card in his pocket, $50 cash and an airline ticket. You deserve better than that! Does the local representative recognize your particular problems? Does he visit you often, even when you're not buying something? Does he open the conversation by asking, "What's that big thing out there with all the rolls that makes so much noise?" Oh, I know I'm becoming factitious, but the point is, a good local "rep" can make up for a lot of deficiencies in equipment. Choose your supplier on the quality of his equipment and service. Mill experience is certainly an important consideration. You wouldn't buy a paper machine from a guy who sold insurance on the side!?

I encourage you to go back and read this whole treatise. Size your own agitators and see how close the proposals come to your calculations. You will have to live with this decision for a long time.

Chapter 12: Agitation Reprise

During the last 40 years or so, the terms "agitation" and "mixing" have become inexorably entwined in my life. "Chief agitator" has been a "title" conferred on me numerous times, mostly in jest, sometimes with a trace of sarcasm, but the realization of the potential of better results achieved from proper mixing (agitation) has been engrained in my subconscious. Always reaching for a "whisk" when I beat up an omelette (for its high shear qualities), using an up-and-down as well as rotational motion when I use a spoon to dissolve chocolate syrup in a glass of milk (to produce uniformity rather than swirl) are simple examples of that awareness of "mixing science" always close to my consciousness.

It's a *learned* science from the early days of observing solid particles swirling about the bottom of an improperly baffled tank to unlocking the mysteries of impeller performance and the relationships of diameter and speed to power response.

Probably the most gratifying investigation was the prolonged, sometimes discouraging, study of those little devils we call paper pulp fibers. How could the suspension of such infinitesimal percentages of solids be so frustrating. Knowing we were but a link in a long chain of investigators, stretching back into the early 19th century, still mystified by the obscure relationships of consistency and type of fiber was a thrilling sensation. The fact that a preponderance of studies had concluded that circulating stock slurries around a wall erected in the center of a bath tub built for a sperm whale was the ultimate answer to the agitation of paper stock only spurred us to more exciting experiments.

Had paper machine builders been content with the existing cylinder machines and the relatively slow fourdrinier machines, perhaps we would still be living in an industry filled with midfeather chests and vertical "Christmas trees." Fortunately, apathy wasn't a part of the vocabulary of many of the machine builders. The

appetite of American industries and the American people for paper of all kinds overtook the capacity of the early fourdrinier machines. Newsprint machines stretched out to 300 ins. and raced toward the dry end at 2000, 2500, 3000 and more ft/min. A linerboard machine of 500 or 750 tons/day is now essentially a special purpose machine with its bigger brothers, 1500 and 2000 tons, looking on it as a "toy." A "mile a minute" machine for tissue was a dream and was surpassed with hardly a celebration. The days of the circulation stock chests were numbered and here was a new technology being born to meet the need. Exciting? You bet!

Well, what about the future? Will a paper machine ever run faster than 6000 ft? Probably so as I write this conclusion! Will we find something better than today's "modern agitator?" Remember the man who wanted to close the Patent Office back in the early 1800s? Oh, and say, didn't Fulton's steamboat use a propeller? I don't know what a paper mill in the year 2000 will look like; if I did, I'd buy stock in whatever the new technology will be. Will we ever retreat from our present technique of dilution and concentration and dilution, to lay a thin slurry on a moving wire? We are already marketing pulpers that operate at 12 and 18% consistency, unthinkable just a few years ago. Well, what about dry forming? A pilot phenomena or tomorrow's paper mill?

I am optimistic the giants of our industry won't die and pass away, but will be reborn in the curiosity of today's graduates, tomorrow's designers, researchers and mill engineers. I hope one day to hear, "Well done, thou good and faithful agitator, you were pretty close with your N^3D^5, it's really _____.

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TIS 0418-03
ISSUED — 1986

Recommended nomenclature for agitators, mixers, and pulpers

Stock chests with horizontal propeller agitators

Shown are a vertical cylindrical chest (Fig. 1), a rectangular stock chest (Figs. 2 and 3), and a high density storage tower (Fig. 4).

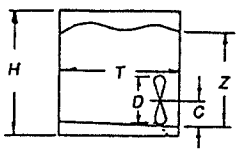

Fig. 1. Vertical cylindrical chest: T = chest diameter; H = chest height; C = off-bottom distance for impeller measured from lowest point in chest to center line of agitator shaft; D = impeller diameter; Z = stock level

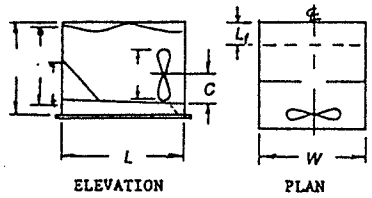

Fig. 2. Rectangular stock chest (L/W = 1.0 to 1.5): W = chest width; L = chest length; H = chest height; Z = stock level; Z_f = fillet height; L_f = fillet length; D = impeller diameter; C = impeller off bottom distance measured from lowest point in chest to center line of impeller shaft.

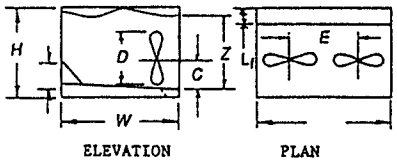

Fig. 3. Rectangular stock chest (L/W > 1.5): W = chest width; L = chest length; H = chest height; Z = stock level; Z_f = fillet height; L_f = fillet length; D = impeller diameter; E = spacing of units; C = impeller off bottom distance measured from lowest point in chest to center line of impeller shaft.

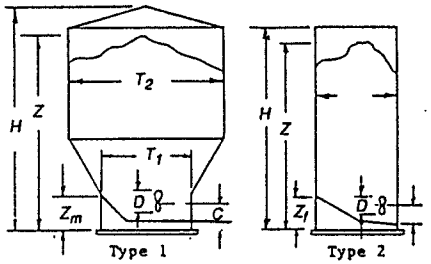

Fig. 4. High density storage tower: T = chest diameter (type 2); T_1 = reduced section diameter (Type 1); T_2 = storage zone diameter (Type 1); Z_m = height of active zone; Z = total stock level; H = chest height; C = off-bottom distance for impeller measured from lowest point in chest to center line of impeller shaft; Z_f = fillet height; D = impeller diameter.

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Stock chests with vertical propeller agitators

Shown are a vertical cylindrical chest (Fig. 5) and rectangular chests (Figs. 6 and 7).

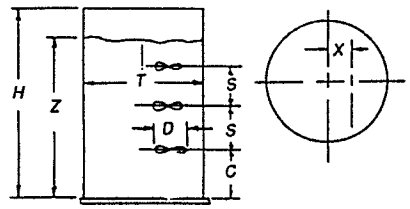

Fig. 5. Vertical cylindrical chests: T = chest diameter; H = chest height; Z = stock level; C = impeller off bottom distance; S = impeller spacing; D = impeller diameter; X = vertical off center distance measured from center line of chest.

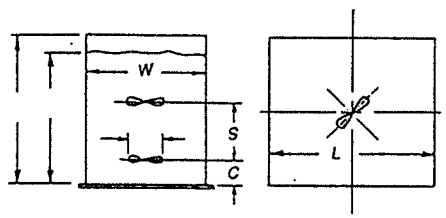

Fig. 6. Rectangular chests (L/W 1.0–1.5): W = chest width; L = chest length; H = chest height; Z = stock level; D = impeller diameter; C = impeller off bottom distance; S = impeller spacing.

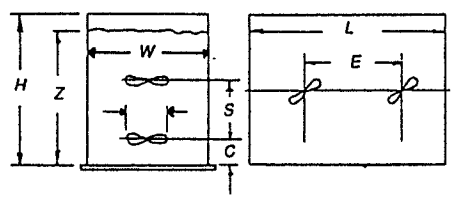

Fig. 7. Rectangular chest (L/W > 1.5): W = chest width; L = chest length; H = chest height; Z = stock level; D = impeller distance; C = impeller off bottom distance; S = impeller spacing; E = spacing of units.

Mid-feather stock chests

Figure 8 shows a mid-feather stock chest.

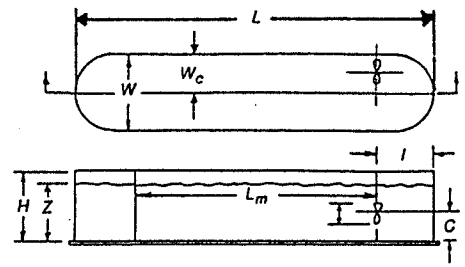

Fig. 8. Mid-feather stock chests: W = chest width; L = chest length; H = chest height; Z = stock level; W_c = channel width; D = impeller diameter; C = impeller off bottom distance; L_m = length of mid-feather wall; l = impeller insertion.

Chemical mixing tanks

Shown in Fig. 9 is a vertical cylindrical tank with vertical mixers, while Fig. 10 shows a vertical cylindrical tank with side-entry mixers.

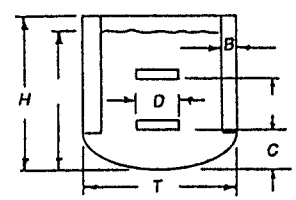

Fig. 9. Vertical cylindrical tank with vertical mixers: T = tank diameter; H = tank height; Z = liquid level; B = baffle width; L = impeller diameter; C = impeller off bottom distance; S = impeller spacing.

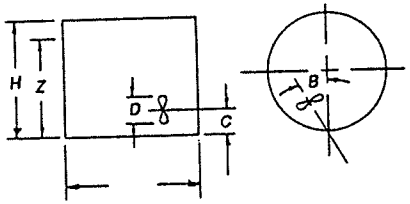

Fig. 10. Vertical cylindrical tank with side entry mixers. T = tank diameter; H = tank height; Z = liquid level; D = impeller diameter; C = impeller off bottom distance; θ = angle of mixer from tank center line.

Pulper tanks

Shown are a vertical bottom rotor pulper (Fig. 11), a horizontal single rotor pulper (Fig. 12), and a horizontal double rotor pulper (Fig. 13).

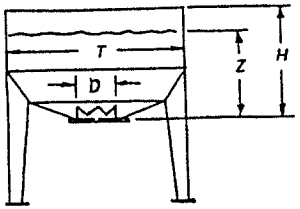

Fig. 11. Vertical bottom rotor pulper: T = tub diameter; H tub height; Z = stock level; D = rotor diameter.

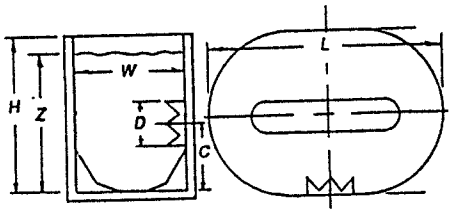

Fig. 12. Horizontal single rotor pulper: W = tub width; L = tub length; H = tub height; Z = stock level; D = rotor diameter; C = rotor off bottom distance.

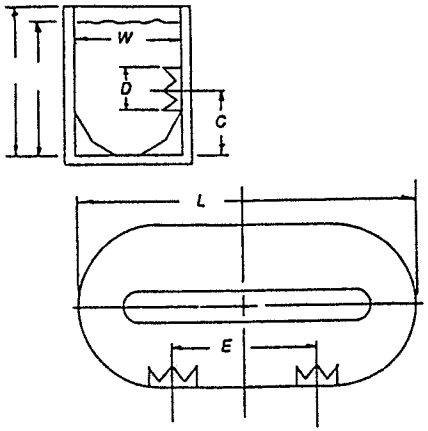

Fig. 13. Horizontal double rotor pulper: W = tub width; L = tub length; H = tub height; Z = stock level; D = rotor diameter; C = rotor off bottom distance; E = rotor spacing.

Miscellaneous nomenclature

Process symbols

c	=	stock consistency
c_a	=	stock consistency, % a.d.
c_b	=	stock consistency, % b.d.
Q	=	impeller pumping capacity
t	=	time
ϵ	=	specific gravity or density
μ	=	viscosity
P	=	horsepower (hp)
N_p	=	power number
N_{Re}	=	Reynolds number
h	=	film coefficient
C_p	=	specific heat

Mechanical symbols

N	=	speed (rpm or rps)
N_c	=	critical speed
l_s	=	shaft length – ℄ impeller to first bearing
a	=	bearing spacing
w_o	=	weight of impeller
w_e	=	equivalent weight
w	=	weight per unit length of shaft

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Celebrating its 75th anniversary in 1990, TAPPI is the world's largest professional society of executives, operating managers, engineers, scientists, and technologists serving the paper and related industries. Total membership is approximately 30,000 with some 80% residing in the United States. The remainder live in one of 76 other countries.

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- We serve as a world-wide forum to exchange technical information, promote research, and recognize individual achievement.
- We create success by the quality, timeliness, and innovativeness of our products and services.

Integrity and fellowship characterize our association.

SI Conversion Units

Quantity or Test	Value in Trade or Customary Unit	x	Conversion Factor	=	Value in SI Unit	Symbol
Area	square inches		6.45		square centimeters	cm^2
	square feet		0.0929		square meters	
	square yards		0.836		square meters	
	acres		0.405		hectares	ha
Basis Weight* or substance (500-sheet ream) or Grammage* when expressed in g/m^2	17 x 22		3.760		grams per square meter	g/m^2
	24 x 36		1.627		" " " "	"
	25 x 38		1.480		" " " "	"
	25 x 40		1.406		" " " "	"
	pounds per 1000 square feet (Paperboard)		4.882		" " " "	"
Breaking Length	meters		0.001		kilometers	km
Burst Index	$\frac{g/m^2}{g/m^2}$		0.0981		$\frac{g/m^2}{g/m^2}$	
Bursting Strength	pounds per square inch		6.89		kilopascals	kPa
Caliper	mils		0.0254		millimeters	mm
Concora Crush	pounds		4.45		newtons	N
Edge Crush	pounds per inch		0.175		kilonewtons per meter	kN/m
Energy	British thermal units (Btu)		1005		joules	J
Flat Crush	pounds per square inch		6.89		kilopascals	kPa
Force	kilograms		9.81		newtons	N
	pounds		4.45		newtons	N
Length	angstroms		0.1		nanometers	nm
	microns		1		micrometers	um
	mils		0.0254		millimeters	mm
	feet		0.305		meters	m
Mass	tons (2000 lb)		0.907		metric tons	t
	pounds		0.454		kilograms	kg
	ounces (avd p)		28.3		grams	g
Mass per Unit Volume	ounces per gallon		7.49		kilograms per cubic meter	kg/m^3
	pounds per cubic foot		1.60		" " " "	
Puncture Resistance	foot pounds		1.36		joules	J
Ring Crush	pounds (for a 6-in. length)		0.0292		kilonewtons per meter	kN
Stiffness (Taber)	gram centimeters (Taber Units)		0.0981		millinewton meters	mN.m
Tear Strength	grams		9.81		millinewtons	mN
Tensile Breaking Load	pounds per inch		0.175		kilonewtons per meter	kN/m
	kilograms per 15 millimeters		0.654		" " "	"
Volume, Fluid	ounces (US fluid)		29.6		milliliters	mL
	gallons		3.79		liters	L
Volume, Solid	cubic inches		16.4		cubic centimeters	cm^3
	cubic feet		0.0283		cubic meters	m^3
	cubic yards		0.765		cubic meters	m^3

* See TAPPI Technical Information Sheet 0800-01.