Impeller design choice is key to stock

Axial-flow impeller design is both mechanically sound and energy-efficient for all top- and side-entering stock agitation applications

By TOM C. DEVRIES

■ Although each stock agitator supplier has a design with its own unique features, they all have one thing in common—a drive design and an impeller design. The combination of these designs results in an agitator design. To solve a particular stock agitation problem, the supplier selects an impeller design and couples this impeller to a motor through a mechanical design consisting of a shaft, bearings, and a speed reduction device. Since most stock agitators are of the side-entering type, the following discussion will focus on the differences in various side-entering stock agitator designs from a process and mechanical standpoint.

AXIAL VS RADIAL FLOW IMPELLERS. First, it is necessary to understand that a process result, such as stock blending or storage, is a function of the impeller design employed and the horsepower delivered by that impeller. In other words, the key to a successful agitator installation is the proper combination of the impeller's design, its diameter, and its delivered horsepower. To illustrate this point, two extremes of impeller designs—the radial-flow and the axial-flow impeller—will be reviewed. Figures 1 and 2 illustrate these two designs in terms of discharge flow pattern in the top- and side-entering modes.

As can be seen from Figure 1, the radial-flow impeller generates a "butterfly" flow pattern, wherein the impeller draws material from both the top and bottom and discharges radially toward the sides of the chest, as opposed to the axial flow, which draws from the top and discharges toward the bottom. In the top-entering mode, either design is acceptable, and in fact, the radial-flow impeller was used exclusively for years until the advent of the more efficient axial-flow impeller.

Placing a radial-flow impeller in a side-entering mode produces a throttled, inefficient, and unacceptable flow pattern as depicted by Figure 2, which is exactly why radial-flow impellers should not be used on side-entering agitators. The axial flow, on the other hand, discharges along the chest floor, up the back wall for return to the suction side of the impeller. This discussion will show that radial flow is an undesirable component in an impeller design because of its inherent inefficiencies in the side-entry mode.

"QUASI-AXIAL" FLOW IMPELLERS. The next step is to review the axial-flow concept as it relates to impeller design. Building a truly radial-flow impeller is easy, and most vendors of agitation equipment manufacture an essentially identical radial-flow impeller producing the same radial-flow pattern. It is not as easy to manufacture a truly axial-flow impeller. To do so requires accuracy that is only available from a device such as a laser

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FIGURE 1: **Top-entry agitator configuration.**

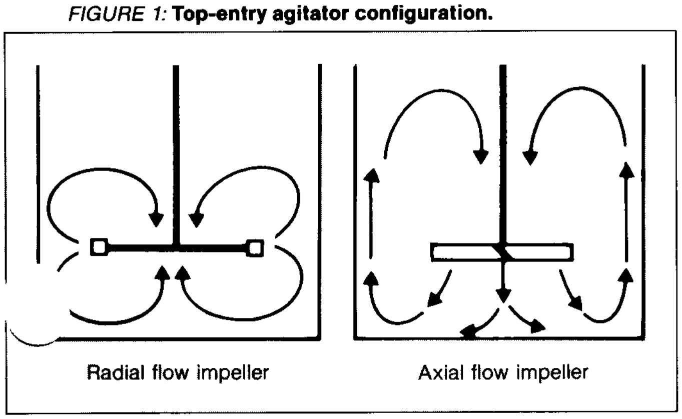
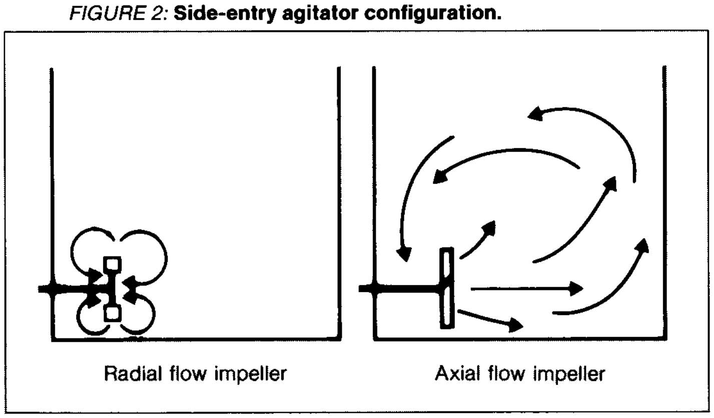


FIGURE 2: **Side-entry agitator configuration.**



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agitator efficiency

velocimeter to accurately measure and record the magnitude and direction of flow leaving the impeller. The laser operates from outside the test tank, and therefore does not interfere with in-tank flow patterns. Without this technology, a truly axial-flow impeller could not be developed. What results, then, is a "quasi-axial" flow impeller consisting of both axial-flow and radial-flow components, as shown in Figure 3.

Because the radial component does exist, a resultant flow is produced that does not parallel the axis of the impeller, and as a result, flow efficiency, expressed as flow per horsepower, is reduced. Not only does the radial component inhibit the impeller's flow efficiency, it also creates additional mechanical loads on the entire agitator system.

Through the use of a laser velocimeter, Lightnin Mixers has developed the truly axial-flow A310/A312 impeller for use with its mixer line. The A312 (the side-entry equivalent of the top-entry A310) has been specifically developed for use on Lightnin's Model VS side-entry stock agitator to withstand the severe service encountered in the paper industry.

Many "energy-efficient" impellers have been developed. When the term "energy-efficient" is used in conjunction with an agitator, it implies that an impeller requires less horsepower to do the process—i.e., the flow per horsepower is higher. There are two ways to render an impeller energy-efficient (lower the horsepower requirement): **(1)** develop a truly efficient axial-flow design that is different in design to other axial-flow impellers and that indeed requires less horsepower at the same diameter, or **(2)** increase the impeller diameter to lower the horsepower required to make the impeller energy-efficient.

Lightnin chose the first approach through the development of the A312. This impeller design generates at least 30% more flow per horsepower when compared with the next most efficient axial-flow impeller of the same diameter.

APPROACH TO ENERGY EFFICIENCY. Because the radial-flow component exists in quasi-axial impellers, its resultant flow is at an angle to the impeller axis and is herefore less flow-efficient. The axial-flow component is the component that does the work. The radial component wastes energy because this component recirculates back to the suction side of the impeller. To bring the efficiency of the alternate design (approach No. 2) impeller "up" to that of the A312 impeller, the diameter of the alternate impeller is increased at the same horsepower so that the axial-flow component has a magnitude of 1.0 (Figure 4). Then both agitators have theoretically equal process capacity at the same horsepower.

The two equal-horsepower agitators in Figure 4 will have equal process capacity only if the impellers are located properly with respect to the chest wall. To draw the proper horsepower and to pump the necessary flow, a side-entering impeller must be located no less than one half of its diameter from the wall on which the agitator is mounted. As an example, an impeller located one third of its diameter from the wall will draw 75% of the power and deliver 60% of the flow of the same impeller located one half its diameter from the wall. This is a significant reduction in performance and is analogous to throttling the flow from a pump by restricting the available volume on the suction side of the pump. To provide for proper off-wall clearance for larger-diameter impellers, a longer shaft is required. This longer shaft combined with a larger-diameter impeller, results in a more mechanically demanding system, as shown in the following discussion.

MECHANICAL REVIEW. To illustrate the effect of increasing the diameter of a particular impeller design at the same horsepower to create an energy-efficient design, a review of the mechanical loads on the agitation system follows. First, two basic process relationships must be described:

Horsepower, $Hp = N^3 D^5$
N, Speed
D, Diameter
Process capacity $= Hp \times D$, or
Momentum, $M = N^2 D^4$

FIGURE 3: **"Quasi-axial" flow impeller.**

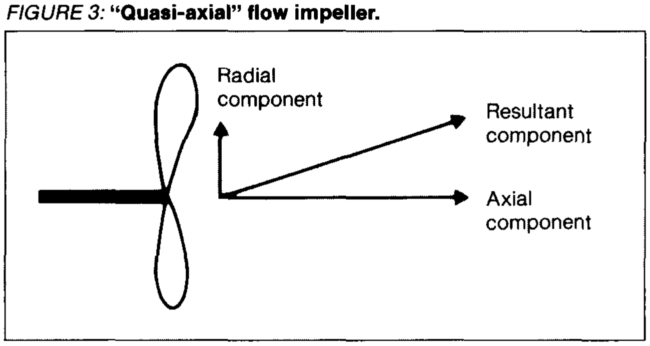
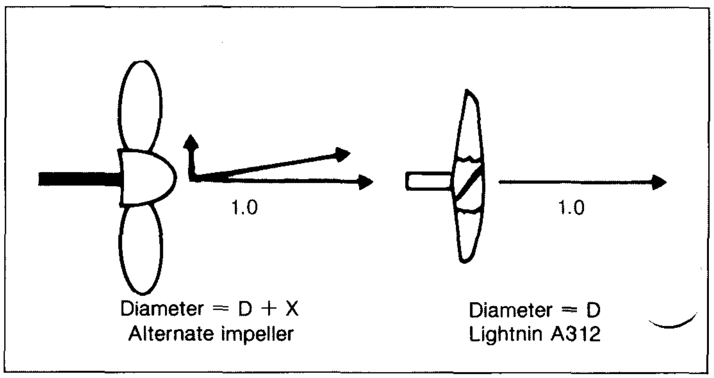


FIGURE 4: **Effect of increasing impeller diameter.**



LIGHTNIN

TABLE IV

LIGHTNIN VS. THE COMPETITIVE
EQUAL PROCESS CAPACITY MACHINES

RELATIVE VALUE	LIGHTNIN VS 75 Hp/50" A312	COMPETITIVE STANDARD SELECTION 100 Hp/50" Impeller	COMPETITIVE ENERGY-EFFICIENT SELECTION 75 Hp/67" Impeller
N	1.63	1.79	1.0
T	.82	1.0	1.34
F	.86	1.0	1.16
W_{prop}	1.0	1.0	1.70
a	1.0	1.0	1.0
X	1.0	1.0	1.0
C	1.0	1.0	1.0
L	1.0	1.0	1.24
W_{shaft}	1.0	1.0	1.24
M_B	.94	1.0	1.73
σ_M	.94	1.0	1.73
σ_T	.82	1.0	1.34
D (at stuffing box)	.94	1.0	1.71

As an example, assume the process capacity required of a side-entry agitator is 5,000. Either a 100-hp, 50-in. impeller or a 75-hp, 67-in. impeller of the same design will satisfy a process capacity of 5,000. Table 1 shows the relative comparison of speed and momentum for these 'wo selections. Table 2 reflects the mechanical relation-hips of torque and fluid force based on the following:

Torque, $T = Hp/N$ (in./lb)
Fluid force, $F = N^a D^b$ (lb)
N, Impeller speed
D, Impeller diameter

Each different impeller design incorporates its own fluid forces—i.e., those forces reacting on the impeller and shaft as a result of displacing stock. The greater the radial component a particular impeller design has, the higher the fluid forces generated by the impeller and the greater the mechanical loads on the agitator.

To mechanically compare the two equal process capacity machines of Table 1 on a relative basis, the following parameters will be used:

- Bending movement
- Bending stress
- Torsional stress
- Deflection.

The definition of these terms is as follows:

Bending moment, $M_B = \left(F + W_{prop} + \frac{W_{shaft}}{2}\right)L$

F = Fluid force (lb)
W_{prop} = Weight of impeller (lb)
W_{shaft} = Weight of shaft from inboard bearing to impeller (lb)
L = Distance from inboard bearing to centerline of impeller

Bending stress, $\sigma_M = \frac{M_B C}{I}$

C = Shaft radius (in.)
I = Moment of inertia at shaft (in.^{4})

Torsional stress, $\sigma_T = \frac{Tc}{QI}$

T = Torque = Hp × 63025/N (in./lb)

Deflection, $\Delta = \frac{Px}{6EI}(2La + 3Lx - x^2)$

$P = F_F + W_{prop} + \frac{W_{shaft}}{2}$

X = Distance from inboard bearing to point in question (in.)
a = Bearing spacing (in.)
E = Modulus of elasticity (psi)

TABLE 1: **Relative comparison of speed and momentum.**

Process capacity	Hp	Dia (in.)	Relative N (impeller speed)	Relative M (momentum)
5,000	100	50	1.79	1.0
5,000	75	67	1.0	1.0

Table 3 reflects the comparison of the two agitator designs, assuming both agitators incorporate equal diameter shafts of the same material and have equal bearing spacing. (Shaft length reflects one-half-dia-off-wall plus 12 in. for wall thickness so that the units remain equal on a process basis.) Table 3 demonstrates that increasing impeller diameter to reduce horsepower results in a significantly more demanding mechanical system. This analysis highlights the fact that equal process capacity machines using the same impeller design are not a legitimate approach unless the mechanical capabilities of the machines are equal. To produce mechanically equal machines, the following review will show the shaft diameter required on the "energy-efficient" selection to yield the same shaft stress as the "standard" selection.

First, the shaft diameter required to provide equal bending stress is as follows:

$$\frac{\delta_{M_E}}{\delta_{M_S}} = 1.0$$

where subscript E means energy-efficient selection
S means standard selection

from above $$\frac{\delta_{M_E}}{\delta_{M_S}} = \frac{\frac{M_{B_E}C_E}{I_E}}{\frac{M_{B_S}C_S}{I_S}} = 1.0$$

because $\frac{M_{B_E}}{M_{B_S}} = 1.73$ and $\frac{I_E}{I_S} = \frac{\frac{\pi C_E{}^4}{4}}{\frac{\pi C_S{}^4}{4}}$

the equation becomes

$$1.73\,\frac{C_E}{C_S}\,\frac{\pi C_S{}^4/4}{\pi C_E{}^4/4} = 1.0$$

then $\left(\frac{C_E}{C_S}\right)^3 = 1.73$

and $C_F = 1.20\, C_S$ for equal bending stress

Similarly, the shaft diameter required to provide equal torsional stress is as follows:

$$\frac{\delta_{T_E}}{\delta_{T_S}} = 1.0$$

from above $$\frac{\delta_{T_E}}{\delta_{T_S}} = \frac{\frac{T_E\, C_E}{I_E}}{\frac{T_S\, C_S}{I_S}} = 1.0$$

because $\frac{T_E}{T_S} = 1.34$ and $\frac{I_E}{I_S} = \frac{\pi C_S{}^4/4}{\pi C_E{}^4/4}$

the equation becomes

$$1.34\,\frac{C_E}{C_S}\,\frac{\pi C_S{}^4/4}{\pi C_E{}^4/4} = 1.0$$

then $\left(\frac{C_E}{C_S}\right)^3 = 1.34$

and $C_E = 1.10\, C_S$ for equal torsional stress

TABLE 2: **Mechanical relationships of torque and fluid force.**

Process capacity	Hp	Dia (in.)	Relative N (impeller speed)	Relative T (torque)	Relative* F (fluid force)
5,000	100	50	1.79	1.0	1.0
5,000	75	67	1.0	1.34	1.16

*Based on empirical exponential values of a = 1.5, b = 3.5 in the fluid force equation.

Because the shaft diameter required for equal bending stress is greater than that required for equal torsional stress, the diameter for equal bending stress controls the selection.

Therefore, the alternative "energy-efficient" selection must have a shaft diameter that is 20% larger than the shaft diameter of the standard selection in order to make the two selections equal for both the process and mechanical designs. Anything less will result in a premature shaft failure as compared with the "standard" selection.

To summarize the essence of this analysis, two facts are evident: **(1)** within the same impeller design (same blade shape, angle, etc.) the horsepower required to properly agitate a stock chest can be reduced by increasing the diameter of that impeller design, but **(2)** the mechanical demands imposed by the larger-diameter impeller necessitate a larger shaft to maintain the same degree of mechanical integrity as that of a higher-horsepower/smaller-impeller design.

Remembering that the above analysis was done solely to explain the process and mechanical considerations involved in the alternative approach of increasing impeller diameter to lower horsepower, it is now necessary to understand the basis for the design of Lightnin's A312 impeller and the VS agitator.

AXIAL-FLOW IMPELLER. Large-diameter impellers are often used to decrease connected horsepower requirements. This approach is taken solely for the purpose of increasing the axial-flow component (the workhorse of a side-entry impeller). Lightnin's laser-developed A312 has the advantage of being a purely axial-flow impeller.

Figure 5 shows a comparison of alternate impellers at diameter D and diameter D + X to an A312 of diameter D to show the proportionality of the axial-flow component (Q_A).

Because the A312 generates pure axial flow, the impeller horsepower can be reduced as well as the impeller diameter. In essence, the Lightnin VS agitator can, based on laboratory analyses and comparisons, offer the power savings equal to or greater than other "energy-efficient" selections, in addition to offering a less demanding mechanical system.

For the alternate energy-efficient selection to be of mechanical integrity equal to the Lightnin VS machine, the alternate shaft must be 23% greater in diameter per the same analysis done above. Anything less will result in premature shaft failure due to excessive bending stress.

AGITATOR PROCESS EVALUATION. When evaluating stock agitator proposals based on pumping capacity (the value of Q in Figure 5), caution must be taken to be sure that the basis for the pumping rate is understood.

First, only the primary pumping capacity should be considered—that is, the flow that is actually leaving the impeller, not the total pumping capacity, which includes the estimated and immeasureable secondary or induced flow. The primary Q that is ordinarily reported is that flow that is associated with the resultant vector Q in Figure 5.

However, because only the axial flow contributes to stock motion, the proper value of Q to be reported should be the axial portion (Q_A) of Q. In the case of the A312, these are the same. However, for the alternate impeller, Q_A is certainly less than Q because of the radial component. The approximate ratio of Q_A/Q for the alternative impeller is 0.9, which deduces the fact that the ratio of the radial component, Q_R, to the primary flow, Q, is 0.44.

This means that more than 40% of the total flow is in the radial direction—an undesirable condition from a mechanical standpoint, as previouly noted. To properly evaluate any agitator's reported pumping capacity, documentation should be presented that substantiates the axial-flow component.

The use of hp × D, or momentum, for evaluation is only valid when evaluating impellers of the same design. As stated earlier, each impeller design incorporates its own characteristics in terms of flow efficiency. Therefore, to evaluate the process capacity of a stock agitator, documentation must be presented that substantiates any claims of flow efficiency. ■

TABLE 3: **Comparison of equal-process-capacity agitators having alternate impellers of the same design.**

Relative values	"Standard" selection 100 Hp/50-in. impeller	"Energy-efficient" selection 75 Hp/67-in. impeller
N	1.79	1.0
T	1.0	1.34
F	1.0	1.16
W_{prop}	1.0	1.70
a	1.0	1.0
x	1.0	1.0
C	1.0	1.0
L	1.0	1.24
W_{shaft}	1.0	1.24
M_B	1.0	1.73
C_M	1.0	1.73
δ_T	1.0	1.34
△ (at stuffing box)	1.0	1.71

FIGURE 5: **Axial-flow components of various impeller designs.**