The Design of a Vacuum Pan

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This paper deals specifically with the design of a mechanical forced circulation calandria vacuum pan. The coordinated design of the calandria and the forced circulation impeller resulted in a pan of exceptional performance. An improved type of barometric condenser and functional instrumentation further enhanced this pan's operation.

A complete detailed discussion of vacuum pan design would require considerably more space and time than can be allotted for this paper. Therefore, only one phase of design will be dealt with in detail. viz., the coordinated design of the calandria and the forced circulation impeller.

Our Company has in four of its factories a total of twelve mechanical circulation calandria vacuum pans. Nine of these pans are original equipment which were converted to the forced circulation type by installing Webre circulators and the Webre system of sugar boiling control. Three of these pans are new pans which were designed by the author. One new pan of the latest design is being built this year for next fall's campaign. By next fall we will have three pans of the new design for boiling white sugar and one for boiling intermediate sugar. Of the nine earlier forced circulation pans, three are boiling intermediate sugar and six are boiling raw sugar.

A detailed study of the proportions versus the operating characteristics of the original nine forced circulation pans leads us to draw up the following general specifications:

1. Of the total finished strike volume, not more than 25 percent to 30 percent should be included in the tubes, center well, and in the space below the bottom tube sheet.

2. The proportions of the pan should be such that the depth of the finished strike above the top tube sheet should not be more than half the internal diameter of the pan.

3. In order to maintain an acceptable production rate, and to enable us to use low pressure vapors not over 15 pounds pressure, there should be approximately two square feet of heating surface for every cubic foot of finished strike volume.

4. If we are to meet the three preceding specifications, the tube lengths must be kept as short as possible which of necessity dictates tubes of a smaller diameter than usual in current practice.

5. More circulation than available in the nine original pans was desired and a velocity in the tubes of two feet per second during the initial boiling down of the graining charge was a design goal.

6. The standard Byers type of barometric condenser was generally satisfactory but it was felt that an improved design was necessary to give closer

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[2] Numbers in parentheses refer to literature cited.

temperature control of the pan and to reduce the water consumption of the condenser.

7. A piston-type dry vacuum pump with a displacement in cubic-feet-per minute of not less than the strike volume of the pan was necessary to give fast accurate temperature control of the pan.

8. A coordinated, compact, and functional instrument system was desired which would automatically control the vapor temperature in the pan, the fillmass temperature and/or the fillmass tightness in the pan. and the rate of boiling by controlling the flow of steam to the pan.

9. Last but not least, all other secondary operations by the sugar boiler should be made as convenient and free of exertion on the part of the sugar boiler as possible enabling him to concentrate his efforts toward improving the quality of the finished strike.

Items 4 and 5 of the general specifications, i.e. the design of the calandria and the forced circulation impeller will now be discussed in detail.

The strike volume and the boiling schedule of the new pan were specified by our operating department to be 1,100 cubic feet of white fillmass every two hours. Space limitations limited the diameter of the pan to 12 feet-6 inches. A study was made of four different tube sizes and four calandria sections were designed. Table I lists the features of these four calandria for comparison.

Fluid flow characteristics were calculated for each of the four calandria for the following conditions: Case I, Water at 75° C., 0° Brix, density 60.9#/cu. ft., Viscosity 0.383 Centipoises; Case II, Liquor at 75° C., 82.5° Brix, density 87.0#/cu. ft., Viscosity 300 Centipoises; Case III, Fillmass at 75° C., 90° Brix, density 90.2#/cu. ft., Viscosity 5,000 Centipoises. For each of the three different fluids five liquid velocities in the tubes were assumed, i.e., 0.5, 1.0, 1.5, 2.0, and 2.5 feet per second. Thus, a total of 60 different sets of operating conditions were calculated and plotted.

Fluid viscosities for the liquor and the fillmass were arrived at by a series of approximations. These were later checked against measurements

Table I.—Calandria Proportions.

Pan Diameter—12′-6″. Well Diameter—4′-9″.				
Total Strike Volume—1,100 cubic feet.				
Heating Surface—2,540 square feet.				
Tube Diameter, Inches	2¼″	2¾″	3″	4″
Tube Diameter, Feet	0.188	0.209	0.230	0.313
Tube Length, Inches	31	33	34	41
Tube Length, Feet	2.57	2.75	2.85	3.40
Number of Tubes	1680	1416	1240	750
End Area of Pan, Ap.	122	122	122	122
Area of Well, Aw.	17.75	17.75	17.75	17.75
Aw./Ap., Percent	14.55	14.55	14.55	14.55
Area Tube Sheet, Ats.	104.25	104.25	104.25	104.25
End Area Tubes, At.	46.3	48.7	51.6	57.7
At./Ats., Percent	44.4	46.7	49.5	55.3
Vol. below Top Sheet	316.3	334.4	349.4	409.0
Vol. above Top Sheet	783.7	765.6	750.6	691.0
Vol. below/Vol. Pan, Percent	28.8	30.3	31.8	37.1
Depth of Strike, Feet	6′-5″	6′-4″	6′-2″	5′-10″

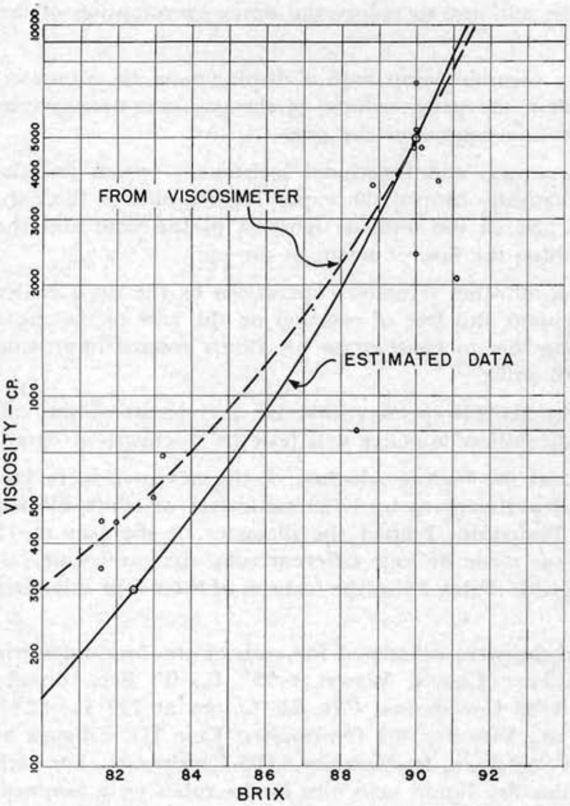

Figure 1.—Apparent Viscosity vs. Brix at 75° C.

taken in the pan with a Bendix Ultrasonic Viscosimeter. The original assumptions of viscosity were surprisingly reliable. Figure 1 shows a curve of apparent viscosity versus brix at 75° C. as measured by the Ultrasonic Viscosimeter, and the curve of assumed viscosities at 75° C.

Reynolds Numbers ($Re = VD\rho/\mu$) for all flow conditions were calculated and Fanning Friction Factors were determined from a chart showing Re vs. f. This is shown in Figure 2. This figure and the method of calculating fluid flow and pressure drops in the tubes are from W. H. McAdams (1)[2], Heat Transmission, Chart V, pages 99 to 134. In Figure 2, I have indicated the regions for the several flow conditions. As will be noted, the flow conditions in the tubes for water are definitely in the turbulent flow region, while the flow conditions in the tubes for the liquor and the fillmass are definitely in the streamline flow region. For the center well the flow conditions for water are very turbulent; for the liquor the flow conditions are just in the turbulent flow region; flow conditions for the fillmass are well into the streamline flow region.

Head losses due to fluid friction in the tubes and well were calculated from the typical Fanning Equation (2) $F_v = fV^2 \frac{(2L)}{(gD)}$ where F_v is the head loss in feet of the flowing fluid at given conditions, f is the Fanning

Friction Factor from Figure 2, V is the fluid velocity in the tubes, L is the length of the tube in feet, D is the internal diameter of the tube in feet, and g is the acceleration of gravity, 32.2 feet/sec./sec.

Head losses due to enlargement and contraction of the liquid entering and leaving the tubes and of the liquid entering and leaving the center well were calculated from the following equation (3): $F_e = \frac{(V_1 - V_2)}{2g}$ and $F_c = \frac{KV_2^2}{2g}$, where F_e and F_c are enlargement and contraction losses in feet of the flowing fluid respectively, V_1 is the velocity of the fluid leaving the tube, V_2 is the velocity of the fluid entering the tube, g is the acceleration of gravity, and K is a function of A_2/A_1 as given in Figure 47, page 122 in McAdams (4).

The data from the 60 sets of fluid flow conditions are so voluminous that no attempt will be made to present all these data in this paper. However, the next four figures amply illustrate in graphical form the highlights gleaned from the tabulated data.

Figure 3 illustrates the friction losses F in feet of fluid against the quantity in GPM circulating through the calandria at a velocity of 2-feet per second in the tubes. F_{total} is the sum of the fluid friction, fluid friction in the tubes F_{vt}, fluid friction in the well F_{vw}, enlargement and contraction losses in the tubes F_{eet} and the enlargement and contraction losses in the well F_{eew}.

Note that F_{eet} and F_{eew} are the same for all three different fluid conditions. Closer inspection of the formula for arriving at these losses will show that the losses are dependent solely on the velocities in the well and in the tubes and are independent of the viscosity and the density of the fluid. Note, also, that F_{eew} is of considerable magnitude, and in the case of water it represents the major loss. In the case of the liquor F_{eew} is also of consider-

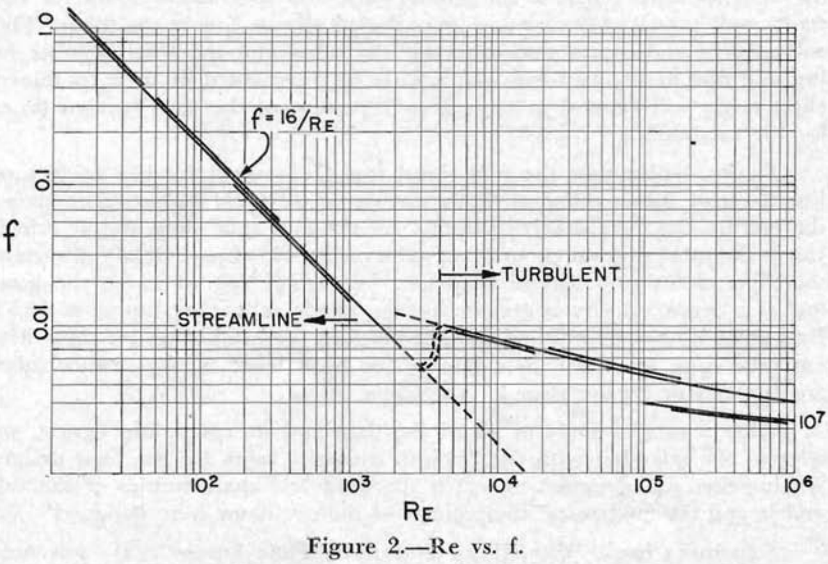

Figure 2.—Re vs. f.

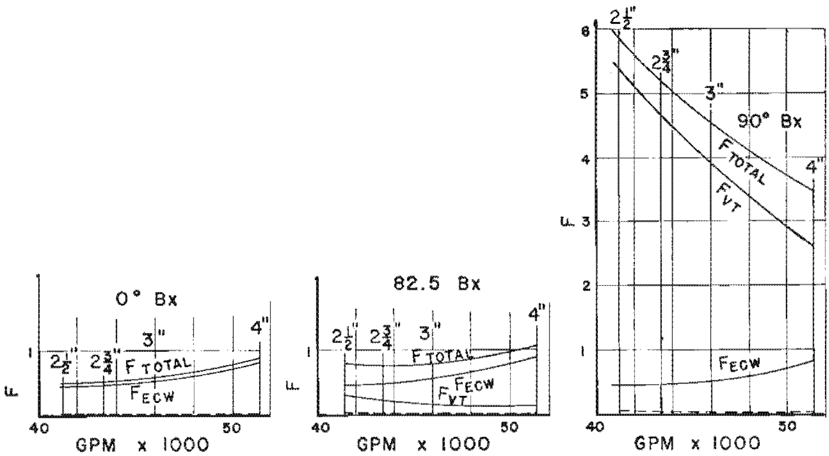

Figure 3.—F, in feet of liquor vs. Q, in gallons per minute. (At tube velocity of 2 feet per second.)

able importance. In the case of the fillmass, F_{ecw} merely modifies the total friction loss. As would be expected, the fluid friction loss in the tubes F_{vt} is small for water, of more importance for the liquor and of major importance for the case of the fillmass. The fluid friction loss in the well as almost so small as to be negligible. Figure 3 illustrates clearly the importance of well size and design as it affects the total head losses encountered in circulating liquor and fillmass in a vacuum pan calandria, especially for forced circulation vacuum pans.

Figure 3 indicates that when one designs a calandria for a vacuum pan, the designer must calculate the enlargement and contraction losses for the center well as well as the friction losses due to viscous flow in the tubes. The enlargement and contraction losses for the tubes and the friction losses in the well due to viscous forces can usually be disregarded as their combined effect rarely will amount to more than 1 percent of the total friction losses for the calandria.

Figure 4 illustrates the total head loss H in the calandria in feet of liquid versus the quantity of liquid circulating in GPM. This figure shows the case for the three different liquids, for the four tube sizes, and at velocities in the tubes of from ½- to 2-feet per second. This figure clearly illustrates the effect of the tube size on head loss; i.e. for the case of water, the head loss at a given velocity is greater for the larger tubes; for liquor at 82.5° Brix there is no appreciable difference between the head losses for the different tube sizes; for a 90° Brix fillmass the head losses in the smaller tubes are appreciably greater than for the larger tubes.

After a careful study of all of the data and the preceding figures, we selected the calandria with the 2¾-inch diameter tubes for the final design. An impeller was designed to match the head loss characteritics of the calandria and the mechanical components of the circulator were designed.

Stepanoff's book, "Centrifugal and Axial Flow Pumps" (5), was used

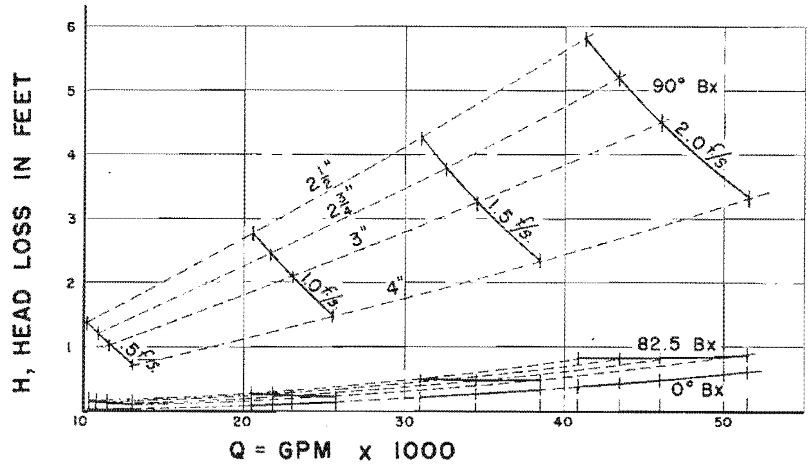

Figure 4.—H vs. Q.

as an aid and a guide in designing the impeller. Briefly the impeller can be described as a four blade, conical hub, axial flow impeller. The initial design conditions are: Fluid, Water; Q = 56,250 g.p.m.; Total dynamic head = 2.06 feet; R.P.M. = 84; Specific Speed = 11,600; Discharge blade angle $22\frac{1}{2}°$; Impelling ratio 1.36.

The impeller as finally designed had a maximum blade tip diameter of 63 inches, an average or nominal blade tip diameter of 60 inches, and a minimum blade tip diameter of 57 inches. The center well is 57 inches in diameter, and there is a conical guide ring surrounding the blade tips. Maximum hub diameter is 30 inches and is conical in shape reducing to 12 inches in diameter at the small end of the hub.

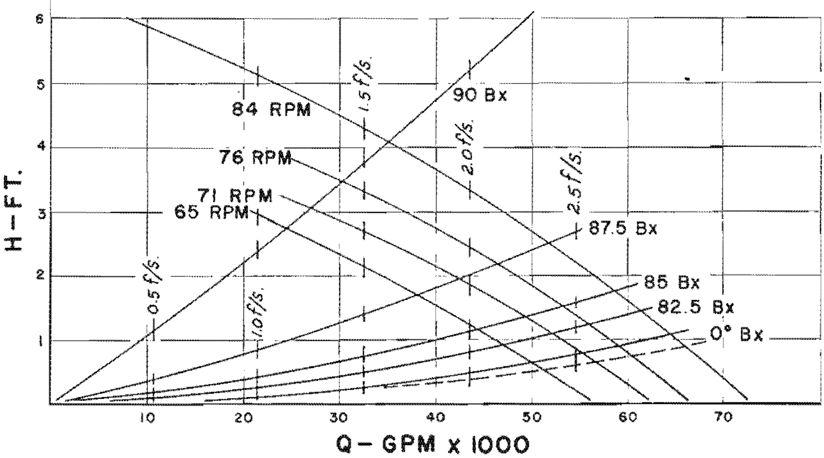

Figure 5.—H vs. Q for $2\frac{3}{4}$-inch tubes and H vs. Q for impeller.

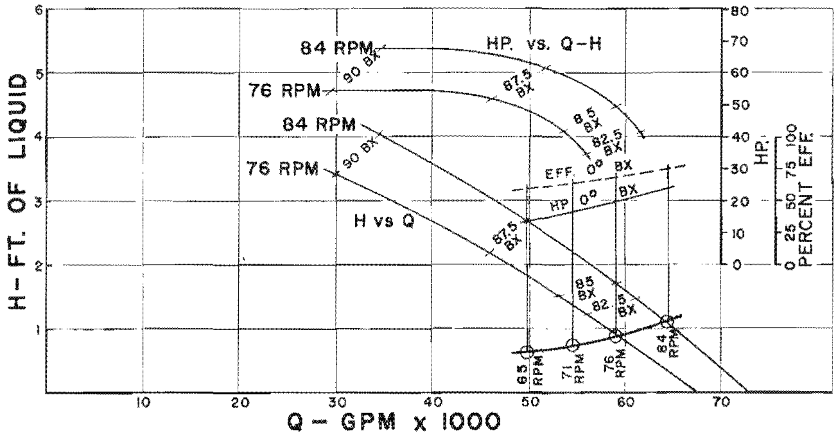

Figure 6.—Impeller characteristics.

Figure 5 shows a plot of the fluid flow characteristics of the 2¾-inch tube calandria versus the Q-H characteristics of the impeller. This figure is a special form of plot known to pumping engineers as a System Head Curve. For this figure I have calculated and plotted additional head loss curves for the calandria for the cases of fillmass at 85° Brix and for fillmass at 87.5° Brix. Q-H curves for the impeller at 84 rpm, 76 rpm, 71 rpm, and 65 rpm are plotted. Shown in dotted lines below the 0° Brix curve for the original pan is a plot of the head loss curve for 0° Brix for the three pans which were designed subsequent to this first pan. The later design is for a 13-foot diameter pan with 10 percent more tubes of 2¾-inch diameter and a length of only 30 inches. These later pans have the same total heating surface as installed in the original pan.

Test points are indicated for data taken at the various impeller speeds when the pan was filled with boiling water at 75° C. The head loss data for water as taken in the test agreed almost exactly with calculated design values.

This close agreement between the head losses calculated and found in test strengthened my confidence in the method of calculation. Therefore, I calculated the impeller efficiencies for water at the four impeller speeds and applied these efficiencies to the cases for liquor and fillmass. Later tests at the four impeller speeds for liquor and fillmass showed remarkably close agreement between the total horsepower required and the calculated total horsepower. Agreement was within 3 percent at a fillmass Brix of 90°.

Figure 6 shows a plot of Q vs. H for impeller speeds of 84 rpm and 76 rpm. Also shown are calculated total horsepower curves for 84 rpm and 76 rpm.

Initial operation at 84 rpm indicated that there was considerable power surging during the tightening of the strike; this suggested that cavitation was taking place at the impeller blades. Therefore, we reduced the speed of the impeller to 76 rpm and eliminated the power surging. There was no indication that the performance of the pan suffered from this speed reduction.

Sugar-boiling operations at the lower speeds of 71 rpm and 65 rpm seemed to indicate that at these lower speeds the uniformity of sugar crystal sizing suffered. This was observed visually on the viewing screen of the Sucroscope and was verified by the increase in CV of the screened sugar samples produced at these speeds. The spread in crystal sizing appears to come about during the first half of the boiling of the strike. Therefore, we feel that the maximum circulation rate which will not produce cavitation during the tightening of the strike is necessary, inasmuch as it produces a more uniform crystal size during the important early phases of the strike when the grain is being formed.

References

(1) McAdams, W. H. Heat transmission, McGraw-Hill Book Company, 1933, pps. 99 to 134, Ch. V.

(2) Ibid., pp. 109, Ch. V.

(3) Ibid., pp. 121, Ch. V.

(4) Ibid., pp. 122, Ch. V.

(5) Stepanoff, A. J., Centrifugal and axial flow pumps, John Wiley and Sons Company, 1948, pps. 143 to 169, Ch. VIII, and pps. 170 to 192, Ch. IX.