SYMPOSIUM 1970
STOCKHOLM

FREQUENCY AND AMPLITUDE OF PRESSURE SURGES GENERATED BY SWIRLING FLOW

H. T. Falvey, Head, Hydraulic Research Section — U. S. Bureau of Reclamation — Denver, Colorado USA

J. J. Cassidy, Professor of Civil Engineering — University of Missouri — Columbia, Missouri USA

SYNOPSIS

This paper reports on the investigation of swirling flow through tubes. Frequencies and amplitudes of pressure surges produced by the swirling flow were measured and found to be essentially independent of viscous effects for Reynolds numbers larger than 1×10^5. Dimensionless frequency and pressure parameters as well as the onset of surging were correlated with the parameter $\Omega D/\rho Q^2$ where Ω and Q are respectively the fluxes of angular momentum and volume through the tube, D is the tube diameter and ρ is the fluid density. Relative length and shape of the tube were also found to be important.

The results of the study were used to analyze a particular draft tube for potential surging. Using performance data obtained from the model of the turbine and draft tube, a region of surge-free operation was outlined on the efficiency hill for the unit. Comparison was made between measured power swings and the predicted pressure fluctuations in the draft tube.

RÉSUMÉ

Cette communication a trait à une recherche sur un courant tourbillonnaire passant dans des tubes. Les fréquences et les amplitudes des surpressions créées par le courant tourbillonnaire ont été mesurées et trouvées essentiellement indépendantes des effets de viscosité pour des nombres de Reynolds supérieurs à 1×10^5. Des paramètres sans dimension liés à la frequence et à la pression et l'établissement de la surpressions ont été reliés au paramètre $\Omega D/\rho Q^2$ où Ω et Q sont respectivement les flux du moment angulaire et du volume dans le tube, D le diamètre du tube et ρ la densité du fluide. On a trouvé que la longueur relative et la forme du tube sont egalement importantes.

On a utilisé les résultats de cette étude pour analyser l'établissement de la surpressions dans une conduite d'ecoulement particulière. En utilisant les donnees de fonctionnement obtenues a partier d'un modèle de turbine et de conduite d'ecoulement, on a trouvé un domaine de marche sans surpressions sur le diagramme de rendement de l'unité. On a comparé les résultats des mesures des amplitudes de puissance et ceux des calculs des fluctuations de pression dans la conduite d'ecoulement.

INTRODUCTION

Draft-tube surging has been recognized as a problem in the operation of hydro-power plants for several decades [1]. History shows that the attack on draft-tube surging has been one of alleviation rather than prevention. New units are placed in operation and if objectionable surging occurs within the operating range various types of remedial action are attempted or, if possible, operation is not permitted in the critical range. However, because of automation of the generating systems, plants are frequently required to operate over a wide range of turbine gate openings. Remedial measures including air admission to the draft tube or the placement of fins in the draft-tube throat are only occasionally successful. If it were possible to predict the possibility of surging prior to installation and operation of a unit, considerable advantage could be realized.

It appears to be generally accepted that draft-tube surging arises as a result of rotation remaining in the water as it leaves the turbine runner and enters the draft tube [2]. Moreover, the phenomenon is complicated by many factors including vaporization of the water, geometry of the turbine and draft tube, and the dynamic characteristics of the penstock and electrical network. Swirling flow has been investigated analytically and experimentally by several people in the field of fluid mechanics [3,4,5,6]. These studies have shown that when a fluid flows in a swirling fashion (axial flow with superimposed rotation) the resulting flow pattern depends upon the relative amount of rotation in the fluid. If the flux of angular momentum entering the tube is sufficiently large as compared to the flux of linear momentum, a reversal in flow direction occurs along the centerline of the tube [3]. For large Reynolds numbers the flow becomes unsteady forming a helical vortex with the reversed flow occurring along the spiral core of the vortex [7]. This phenomenon is known as "vortex breakdown" in the fluid mechanics literature and as draft-tube surging in the hydraulic-machinery literature [8].

This paper describes a study initiated in order to investigate the characteristics of swirling flow in tubes and to correlate these characteristics with the occurrence and nature of draft-tube surging.

THE BASIC STUDY

Because of the excessive number of variables which could be involved it was not only desirable but necessary to simplify the experimental model as much as possible. Air was used as the working fluid in order to eliminate the possible effects of a two-phase flow. No turbine runner was installed in order to simplify geometry. Instead the air entered the tube (Figure 1) radially through wicket-gate type vanes. It was possible to accurately determine the amount of swirl imparted to the flow as it entered the tube. The angle between the vanes and a radial line could be set at any angle between 0° and 82°. At 0° the vanes were

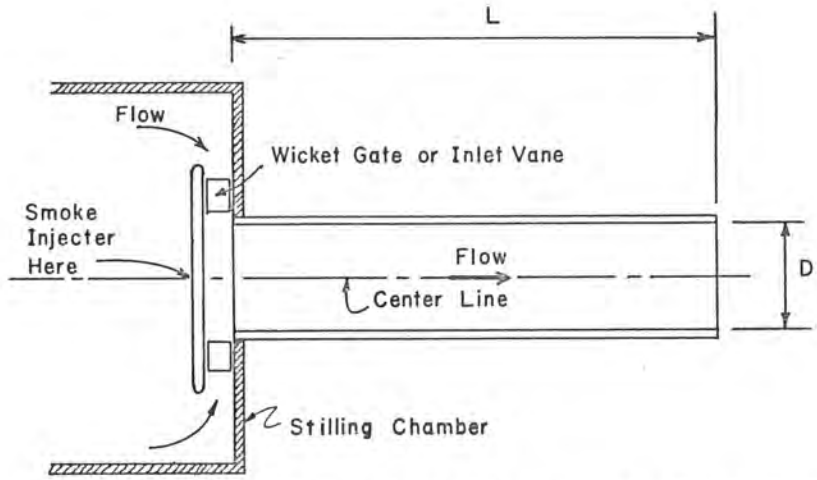


FIGURE 1 EXPERIMENTAL APPARATUS SHOWING STRAIGHT TUBE

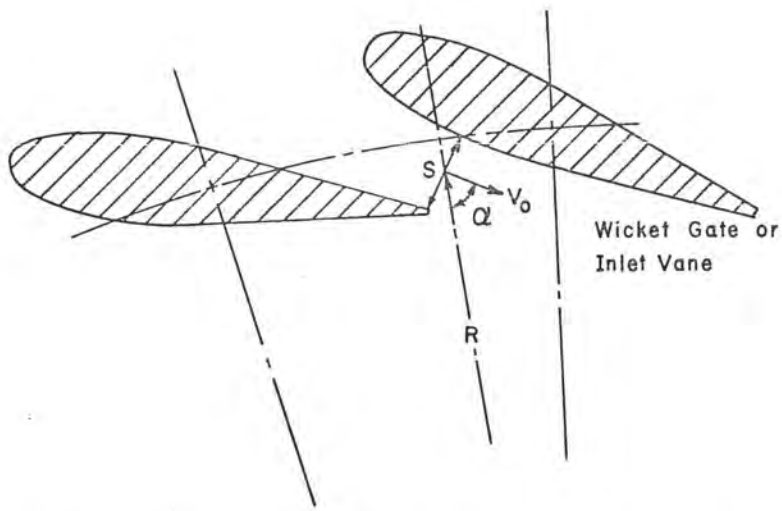


FIGURE 2 DEFINITION SKETCH OF INLET FLOW TO TUBE OR RUNNER

radial and no rotation was applied to the flow (Figure 2). Overall configuration of the apparatus is shown in Figure 1 and is described in more detail by Cassidy [7]. Clear plastic was used in construction to facilitate flow visualization with smoke. Pressures at points on the tube wall were measured using pressure cells and a root-mean-square meter. Frequencies of the unsteady pressure were determined using an oscilloscope with a retentive screen.

Flow through three types of tubes was studied:

1) circular tubes of uniform diameter; 2) a model of a draft tube used in Fontenelle Dam; 3) an expanding cone having the same length-to-area relationship as the draft tube.

ANALYSIS

It was assumed that, for a particular draft-tube shape, the root-mean-square amplitude ΔP and frequency f of the unsteady pressure surge are both functions of the fluid density ρ and viscosity ν, draft-tube diameter D and length L, discharge Q, and flux of angular momentum Ω. Standard techniques of dimensional analysis yield the following functional relationships

$$\frac{D^4 \Delta P}{\rho Q^2} = \phi_1\left(\frac{\Omega D}{\rho Q^2}, \frac{L}{D}, \mathbb{R}\right) \tag{1}$$

$$\frac{f D^3}{Q} = \phi_2\left(\frac{\Omega D}{\rho Q^2}, \frac{L}{D}, \mathbb{R}\right) \tag{2}$$

where $\mathbb{R}$ is the Reynolds number $4Q/\pi D\nu$.

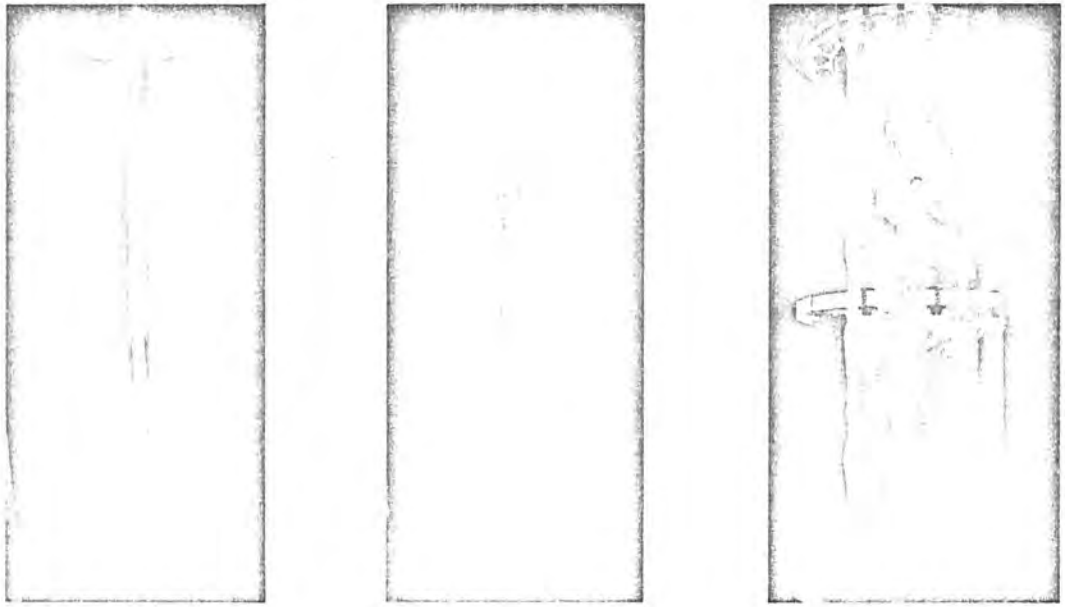
The angular momentum flux Ω entering the tube was computed as (see Figure 2)

$$\Omega = \rho Q R V_0 \sin\alpha \tag{3}$$

in which $Q = SBNV_0$. The variables S, α, V_0, and R are defined in Figure (2); N is the number of vanes around the inlet; and B is the dimension of the vanes perpendicular to the plane of Figure 2. The momentum parameter contained in Equations (1) and (2) is then computed as

$$\frac{\Omega D}{\rho Q^2} = \frac{DR \sin\alpha}{BNS} \tag{4}$$

and is seen to be a function of geometry only for the particular inlet conditions of this study. In all experiments the momentum parameter $\Omega D/\rho Q^2$ was calculated according to Equation (4).



a.) Steady Swirling Flow, no surging, D=3.46 inches, L/D=7.15, $\Omega D/\rho Q^2=0.150$

b.) Stagnation Point at the Centerline With Surging downstream D=3.46 inches, L/D=7.15, $\Omega D/\rho Q^2=0.330$

c.) Fully Developed Surging, $\Omega D/\rho Q^2=0.454$, D=6.13 inches, L/D=3.26.

Figure 3. SWIRLING FLOW IN STRAIGHT TUBES

RESULTS

Flow patterns at low velocities were readily made visible with smoke. With the inlet vanes in the radial position no rotation was imparted to the fluid and streamlines in the test section were essentially straight and parrallel to the tube centerline. With the vanes inclined slightly from the radial position (discharge held constant) streamlines in the test section became steady spirals. That condition is shown in Figure 3a. Further closure of the vanes eventually produced vortex breakdown. Figure 3b shows the stagnation point occurring near the right quarter point of the tube with a helical vortex occurring downstream from the stagnation point. Still further closure of the vanes moved the stagnation point to the upstream limit of the tube. Figure 3c shows the helical vortex filling the tube. A comparison of this helical vortex pattern with those observed below a turbine runner [2,8] establishes the similarity.

Although flow patterns could be observed only at low velocities (below 5 feet per second) unsteady pressures could be measured only at higher velocities. To insure that similar flow patterns occurred at these higher velocities, a hot-wire anemometer was used to measure air velocities near the tube wall and at the tube centerline. These measurements showed that surging began at the same gate setting (same $\Omega D/\rho Q^2$) regardless of discharge.

In the experiments pressure amplitudes and frequencies occurring after surging commenced were measured near the open end of the straight tubes, upstream from the elbow of the draft tube, and near the entrance of the straight cone. The onset of surging (vortex breakdown) was correlated with the value of $\Omega D/\rho Q^2$. It was also found to be a function of L/D and shape. For the elbow draft-tube and the straight cone the critical value of $\Omega D/\rho Q^2$ was 0.4. Above this value surging occurred.

The dimensionless frequency and pressure parameters of Equations (1) and (2), calculated from the experimental measurements, are shown in Figures (4) and (5). These parameters were found to be independent of viscous effects for Reynolds numbers greater than 1×10^5 and the results for smaller Reynolds numbers are not shown here.

In order to obtain reproducible results, it was necessary to measure the root-mean-square values of the pressure fluctuations and it is these values which were used in the preparation of Figure 4. Straight tubes are seen to have pressure characteristics which are strongly influenced by relative length. The pressure characteristics of the elbow draft-tube and straight cone are striking since the pressure parameters exhibit a maximum value as $\Omega D/\rho Q^2$ is made larger.

Frequencies of the unsteady pressure were quite regular and, thus, relatively easy to measure. The regularity indicates that the helical vortex precesses about the tube centerline at a reasonably regular rate. Figure (5) shows that the divergent tubes significantly reduce the frequency of surging. However, the bend in the draft tube has an insignificant effect upon frequency.

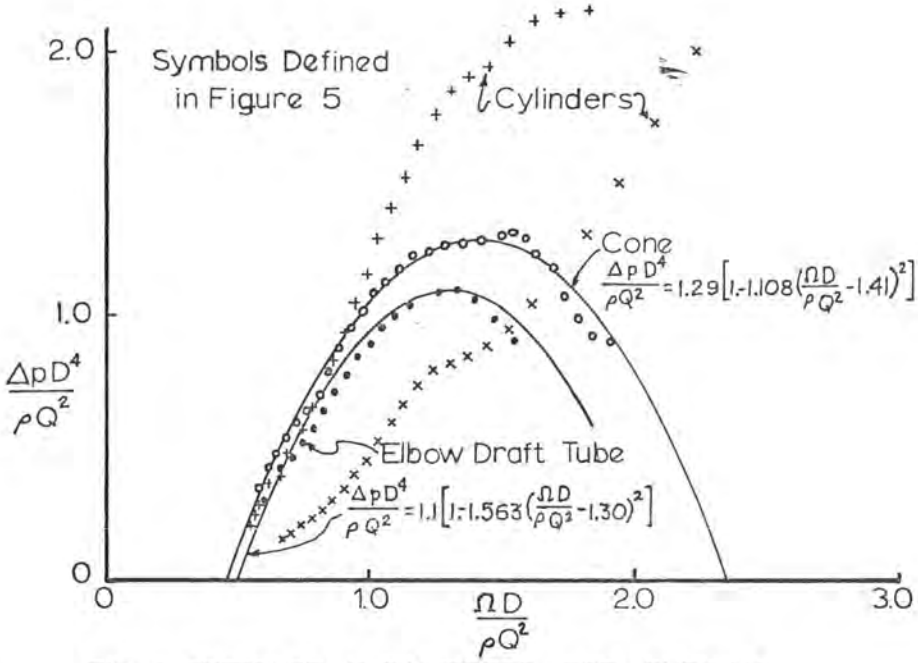


Fig. 4-PRESSURE PARAMETERS FOR SURGING

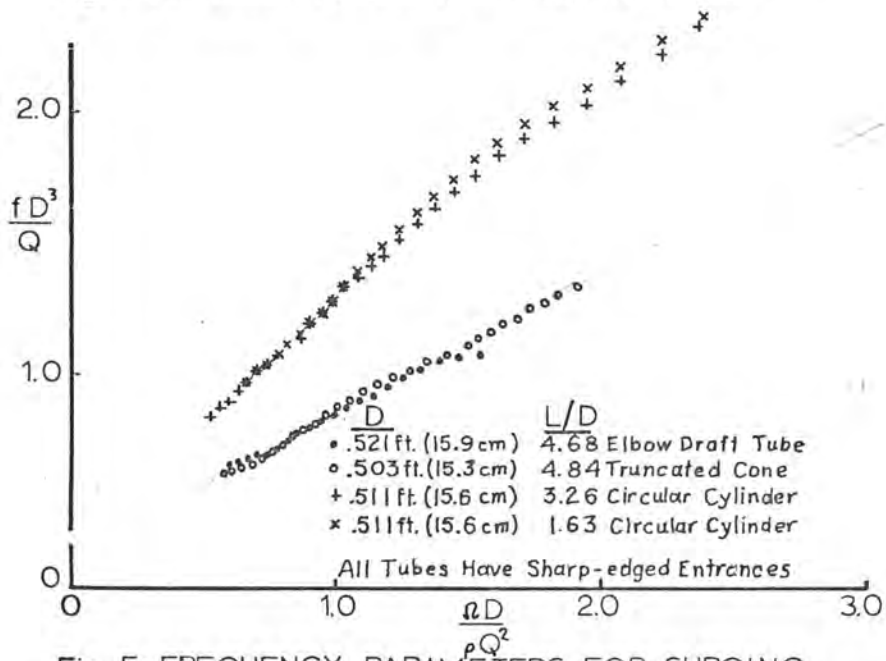


Fig. 5-FREQUENCY PARAMETERS FOR SURGING

E 1

ANALYSIS OF DRAFT-TUBE SURGING IN A TURBINE

Occurrence and severity of draft-tube surging in a given hydraulic turbine and associated draft tube can be predicted using the experimental results of this study. To do this it is necessary to be able to compute the parameter $\Omega D/\rho Q^2$ of the flow as it leaves the turbine runner and enters the draft tube.

Analysis of Turbine

Horsepower P imparted to the turbine runner by water flowing through it is given by the classic expression

$$P = \frac{2\pi n}{60}(\Omega_1 - \Omega_2)/550 \qquad (5)$$

where n is the rotational speed of the turbine in revolutions per minute, and $(\Omega_1 - \Omega_2)$ is the rate of change in angular momentum occurring in the flow as it passes through the runner. Subscripts 1 and 2 refer to the entrance and exit sides of the runner respectively. Multiplying both sides of Equation (5) by $550\ D/n\rho Q^2$ and using Equation (4) yields

$$\frac{\Omega_2 D}{\rho Q^2} = \frac{DR\sin\alpha}{BNS} - \frac{550}{2\sqrt{2g}}\frac{P_{11}}{\rho Q_{11}^2 \phi}\frac{D}{D_2} \qquad (6)$$

where ϕ, P_{11}, Q_{11}, and D_2 are respectively the specific speed, power, discharge and runner diameter and are defined in Figure 6 and $\Omega_2 D/\rho Q^2$ is the momentum parameter associated with the flow leaving the turbine runner and entering the draft tube. The first term on the right of Equation (6) is the momentum parameter associated with the flow leaving the wicket gates and entering the turbine and can be evaluated if the wicket-gate geometry is known. The second term on the right of Equation (6) can be evaluated if the performance characteristics and geometry of the turbine are known.

Specific Application

Equation (6) was used to analyze the hydraulic turbine at Fontenelle Dam [9]. The efficiency hill for the turbine, as determined from model tests, is shown in Figure 6. Any point on this efficiency hill yields a particular value of $\Omega_2 D/\rho Q^2$ and, thus, areas of potential surging--where $\Omega_2 D/\rho Q^2$ exceeds 0.4--can be determined. Figure 7 shows the regions in which surging can be expected and also shows a line representing operating conditions for which flow leaving the turbine and entering the draft tube has zero swirl. To the left of this line flow in the draft tube swirls opposite to the turbine rotation while to the right swirl is with the runner. The point of maximum efficiency falls on the line of zero swirl for this unit although this is not apparently true of all units.

Figure (4) predicts that for the elbow draft tube a maximum pressure amplitude occurs at $\Omega_2 D/\rho Q^2 = 1.30$. The line on which surging would produce this maximum pressure amplitude is also shown on Figure 7. A parabolic curve was fitted to the pressure data for the elbow draft tube. This parabolic expression (shown in Figure 4) was used in conjunction with Equation (6) to calculate relative values

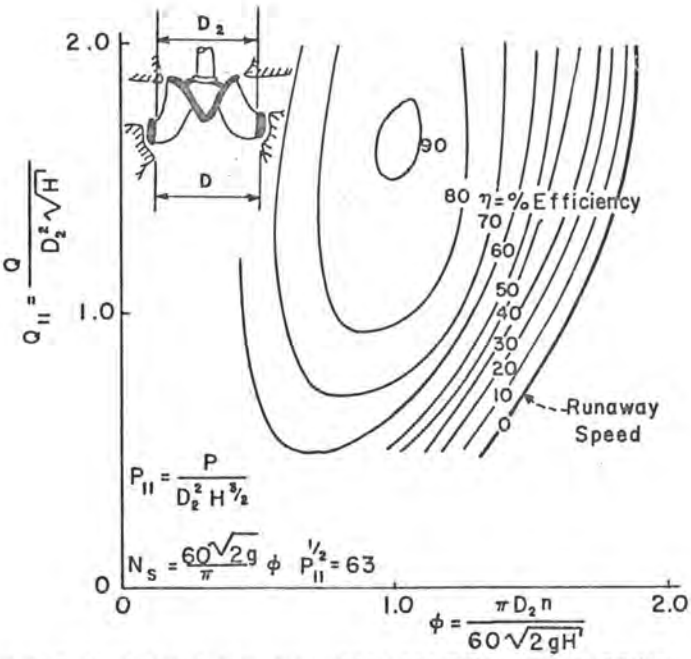


FIGURE 6 EFFICIENCY HILL FOR FONTENELLE TURBINE

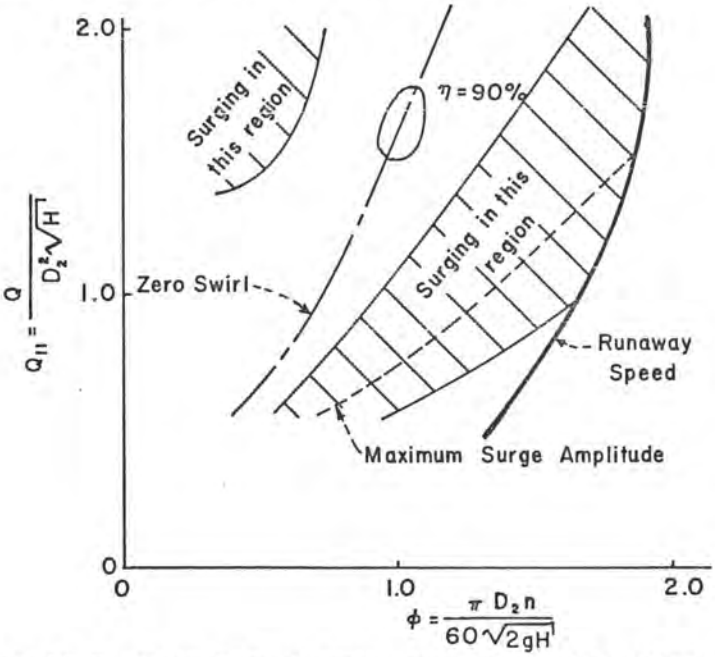


FIGURE 7 PREDICTED SURGING FOR FONTENELLE TURBINE

($\Delta P/\rho gH$ where H is net head on the unit) of pressure amplitudes for the Fontenelle unit. The results of this computation are shown on Figure 8 along with relative power swings actually measured at the plant. Excellent agreement is shown for the wicket gate opening at which maximum surging occurred. Numerical agreement of the % power swing and % pressure fluctuation was not expected because the dynamics of the mechanical and electrical system are necessarily involved in determining the power swing arising as a result of the pressure surging which is the driving force. The lack of agreement occurring at each side of Figure 8 may also be due to the use of Equation (4). Analysis of other units indicated that this expression may not give an entirely accurate estimate of the momentum parameter for all gate openings usually because the exact geometry of the wicket gate assembly is not accurately known.

CONCLUSIONS

A particular hydraulic turbine can be analyzed for possible surging within the desired range of operation by the method presented herein. Dimensionless frequencies and root-mean-square amplitudes of unsteady pressures measured herein are directly convertible to prototype conditions by similitude considerations. However, they can be expected to predict prototype conditions only for the "hard-core" vortex condition. For low-tailwater conditions, for which the vortex core becomes unwatered, a two-phase flow occurs and pressure surges are usually somewhat milder.

ACKNOWLEDGEMENTS

Mr. U. J. Palde of the Hydraulics Branch, U.S. Bureau of Reclamation, took much of the data and carefully refined the experimental methods and equipment used.

REFERENCES

1.) Rheingans, W.J., "Power Swings in Hydroelectric Power Plants," Transactions, American Society of Mechanical Engineers, Vol. 62, 1940.

2.) Hosoi, Y., "Experimental Investigations of Pressure Surge in Draft Tubes of Francis Water Turbines," Hitachi Review, Volume 14, Number 12, 1965.

3.) Harvey, J.K., "Some Observation of the Breakdown Phenomenon," Journal of Fluid Mechanics, Vol. 14, 1962.

4.) Chanaud, R.C., "Observations of Oscillatory Motion in Certain Swirling Flows," Journal of Fluid Mechanics, Vol. 21, 1965.

5.) Benjamin, T.B., "Theory of the Vortex Breakdown Phenomenon," Journal of Fluid Mechanics, Vol. 14, 1962.

6.) Squire, H.B., "Analysis of the Vortex Breakdown Phenomenon, Part I," Aero Department, Imperial College, Report Number 102, 1960.

Fontenelle
relative
own for
greement
the
d in
which is
re 8 may
ed that
um para-
cket gate

within
nless
ed herein
tions.
e "hard-
ex core
y some-

, took
ment

actions,

bes of

l of

g Flows",

of

Aero

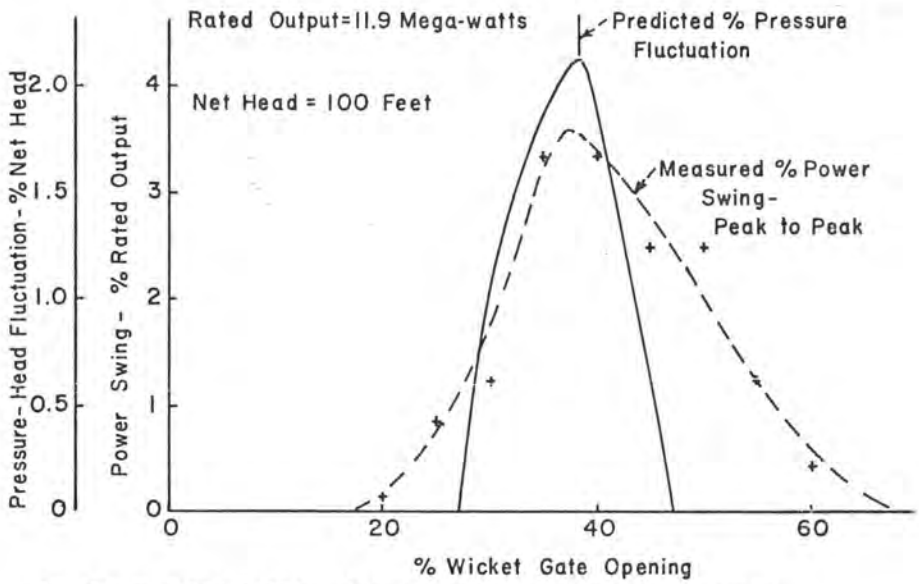


FIGURE 8 PREDICTED AND MEASURED SURGING EFFECTS

Contribution by J. Raabe on Paper E1

Since the publication of Meldau:"Drallströmung im Drehhohlraum" in 1935 we know that steady potential flow in a straight tube with an axial component and a circulation as Dr Falvey and Prof. Cassidy describe is usually connected with a dead water core.The ratio of core diameter to internal tube diameter is controlled by the law of minimum of kinetic flow energy content in the tube. This law is a special case of the law of minimum resistance.Bammert and Kläukens have enriched 1950 our knowledge of straight tubular potential flow with circulation in so far as they stated and observed a back flow in the core for those cases in which the pressure on the external boundary of the core drops below the back pressure. In a longer tube with pressure losses in axial direction this return of the flow in the core may happen at variable stations on the axis.The position of this depends on circulation and throughflow.

In reality the dead water core is also set in rotation by boundary friction.Therefore it can be taken as a vortex.Slight irregularities in the axial flow with enlarged discharge displace the vortex center from the axis. These irregularities belong strongly to the inlet conditions in the tube.

Hereby the displaced vortex core element induces at other vortex core elements a radial displacement. The displaced element does the same with a following element, by this forming a helical vortex. This vortex rotates in circumferential direction in the sense of circulation of the surrounding flow, also due to induced velocity. The tangential induced velocity at any core element and its approximation to the wall due to suction effect of surrounding flow enlarges the radial deviation of the vortex core so that it touches the tube wall at last.This is seen clearly by the experiments of Dr Falvey and Prof. Cassidy.

They stated that viscosity ~~until~~ after the observed Reynolds number of 10^5 is negligible. Therefore rotational speed of the vortex and the pressure pulsations in connection herewith can only be controlled by inertia

forces of rotating and axial-flow masses. Dimensionless pressure amplitude and its frequency is therefore a function of dimensionless ratio of kinetic energy of rotational and axial flow as formulae 1 and 2 show.

The dependence of the ratio L/D and Reynolds number describes the dependence of the flow returning point in the axis as function of pressure drop as mentioned above.

The curiousity not known in real turbines seems to be that the formatio of the helicoidal whirl begins at large distances from the inlet of the tube. It is in contrast to the swirl of a turbine, beginning mostly behind the runner, at the hub or in the runner itself.

The use of the model formula in the shell diagramm of a turbine therefore rises some doubts:

1/ Flow in a turbine runner, especially a Francis runner of high specific speed, can only be described by one circulation before and behind the runner after thoroughful measurements of flow velocity at these stations.

2/ Due to fig 4 pressure pulsation will be higher on a cylinder. This in contrast to the experience that an internal coaxial cylindrical tube in the draft tube quiétens the pressure pulsations remarkably.

3/ Fig 7 shows surging regions but not the magnitude of pressure pulsation. The limits of these regions are gained by equations 6 which uses the shell diagramm fig 6. This diagramm doesn't distinguish a helical whirl region and a pulsating straight whirl region. Especially the pulsations of a central whirl at higher discharges with circulation against the speed causes the stronger shocks.

4/ From fig 5 follows for a runner with given D,N,B that

$$f/Q \sim \sqrt{\sin\alpha} \sim \sqrt{\cos\alpha_0}$$

where α_0 is the guide opening angle.

For Francis turbines one can estimate

$$\cos\alpha_0 \sim \text{constant}$$

By this

$$f \sim \sqrt{H}$$

Observing

$$\sqrt{n} \sim u$$

we have

$$f/n \sim \text{constant}$$

Concluding I can state that the air tests of the authors have a preliminary value in predicting qualitatively the Δp-region but not the exact strength of Δp. On behalf of a constant frequency/ speed ratio the paper may give some approximation to the prototype. The resting guide apparatus with its wicket gates and its sharp inlet edges in the tube doesn't coincide with the runner as the real boundary condition.

H. T. Falvey, J. J. Cassidy, "Frequency and Amplitude of Pressure Surges Generated by Swirling Flow"

Answered

DISCUSSION

by Rex A. Elder, Director, TVA Engineering Laboratory, USA and
Svein Vigander, Research Engineer, TVA Engineering Laboratory, USA

The authors have presented model data in dimensionless form from which the frequency and amplitude of pressure pulsations in draft tubes of hydraulic turbines can be predicted. This type of information is needed both to better understand the generation of periodicity in the swirling flow and to enable prediction of the frequency and severity of disturbances to be expected in new turbine designs.

We would like to add a few comments based on the studies made by TVA's Engineering Laboratory on the vibrations of a 36-MW fixed-bladed turbine unit. Mention of these vibration studies were made earlier, at the 1968 Lausanne Symposium and at the 13th Congress in Kyoto.

On the enclosed Figure 1, we show the results of measurements of pressure pulsations made at the draft tube man door below the turbine runner. The pressure pulsations were measured with a flush-mounted, diaphragm-type, electronic pressure transducer. The data has been reduced to the dimensionless coordinates shown on the author's Figure 4 and Figure 5. The applicable parts of these figures are also shown on the enclosed Figure 1. The prototype data scatter right around the curves drawn through the model results for both frequency and amplitude of pulsations. This agreement, we feel, is quite remarkable, particularily in view of the fact that geometric similarity between the model and the prototype in this case was limited to whatever is implied in the general phrase "Elbow Draft Tube". In light of these results, it would be interesting to see data from Fontenelle Dam and other dams plotted in a similar manner. We hope that the addition of this information may make engineers even more confident in the use of the author's results.

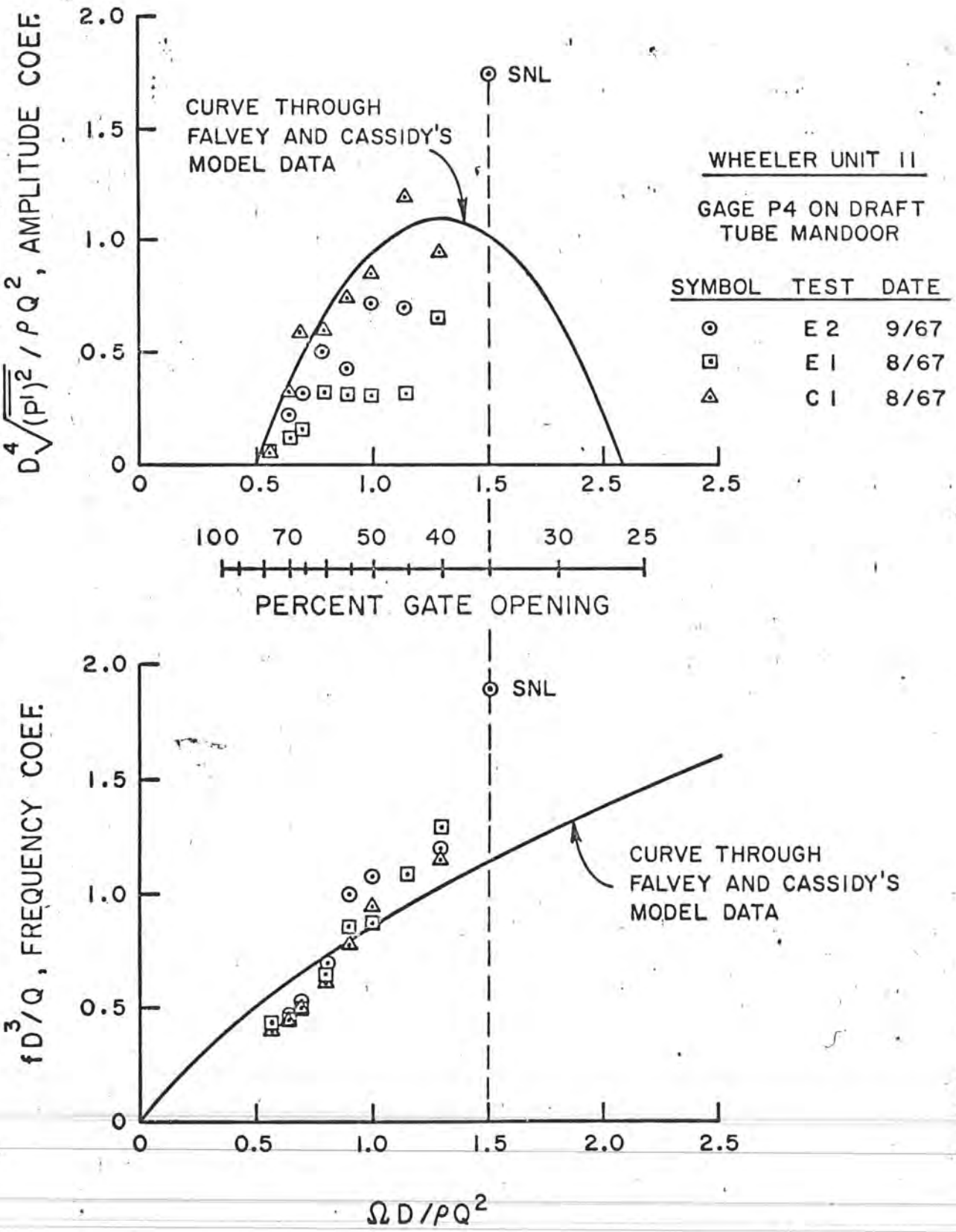


Figure 1 – Draft tube pressure pulsations

Mr. S. CASACCI
Doctor of Math Sciences
Technical Director
Energy Division
ALSTHOM - NEYRPIC
Grenoble - France

Remarks on report E_1 of Mr. H.T. FALVEY and J.I. CASSIDY "Frequency and amplitude of pressure surges generated by swirling flow"

The paper of Mr. Falvey and Mr. Cassidy comprises an interesting attempt to explain pressure surge phenomena which are produced in the draft tubes of hydraulic turbines ; this explanation gives an account of the general aspect of their experimental results.

On the characteristic hill curve of a Francis turbine, can indeed be seen a surge-free zone corresponding to slight swirling flow at the runner outlet. Outside this zone, swirling flow appears rotating in the same direction as the runner or in the opposite direction, depending on whether the operating point happens to be above or below the surge-free zone.

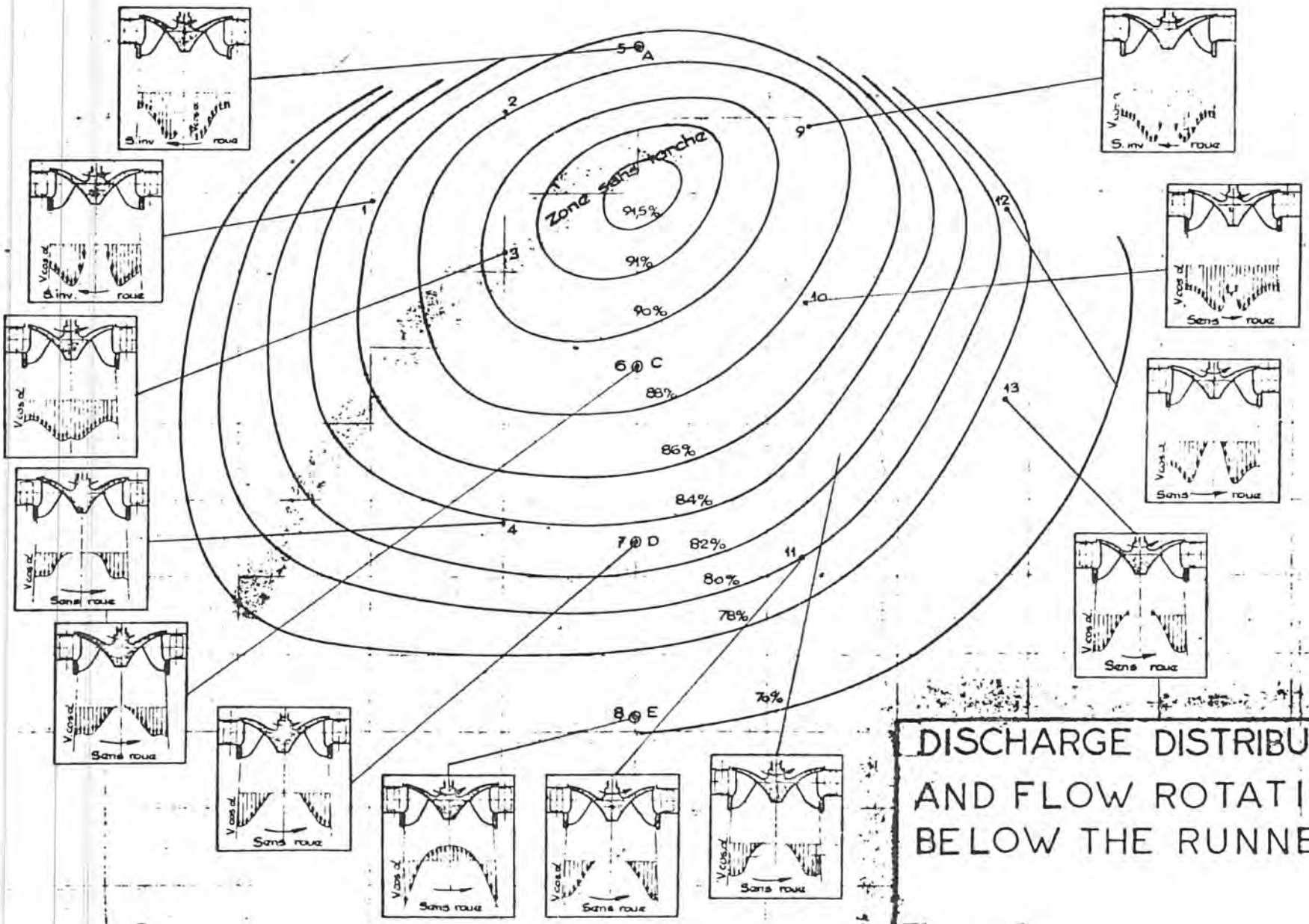
Starting from the hill curve and knowing Ω_1, it would be possible to calculate Ω_2, but it is in fact very difficult to calculate Ω_1 from the geometry of the distributor. Moreover, Ω_2 defined from the output corresponds to an overall value. There exist operating points where the water, leaving the runner, turns in one direction for one operating zone and in the other for another. When swirling flow appears at the runner outlet, what value must be chosen for Ω_2 to define $\Omega_2 D/pQ^2$?

On either side of the "calm" area, having an absolute comparable value $\Omega_2 D/pQ^2$, the contours of the axial velocities differ widely ; the stability conditions should not therefore be the same ; furthermore the distribution of the axial velocities in the model without a runner used by the authors is very different from those of a model with a runner. It is worth pointing out that the vortices do not take the same form on each side of the swirl free zone : they are roughly axial for the large Q_{11} and helical for the small Q_{11}. It seems therefore difficult to characterise the instability of a flow so complex by a single parameter $\Omega D/\rho Q^2$.

The attached figure shows the distribution of axial velocities at the draft tube inlet for various operating conditions. For points above and below the "calm" zone, the maximum axial velocities for this machine occur at the centre and near the draft tube wall respectively. For a given $\Omega D/pQ^2$, the stability conditions for these two types of flow should be different.

It therefore seems that tests on a model equipped with a runner are necessary to define the pressure surges and their transposition onto the prototype machine. The frequency of these pressure surges is of as much if not more interest than their amplitude. In fact, the problematical pressure fluctuations do not always have well defined frequencies. For partial loads, more or less periodic fluctuations can be noted ; the frequencies of the maximum amplitudes depend on the machine's speed of rotation and are little influenced by cavitation. For high loads, the frequencies, less well defined, are a function of σ. Their problematical character would require the use of the energy spectral densities and root mean square values. On this subject, the authors compare the peak-to-peak amplitude variations of the power fluctuations to the root mean square values calculated from the pressure surges. In fact, the root mean square values should be compared, allowing for the transfer function linking the variations in output to the variation in pressure in the draft tube, considered as the only important disturbances. Readings taken on industrial machines do seem to indicate that the spectral densities relating to shaft torque variations are difficult to link to those concerning draft tube pressure fluctuations.

Finally, it must be stressed that it is the whole range of frequencies of several tens of Hertz and under which is of practical interest and not the lowest frequencies of about 1 Hertz. Knowledge of this frequency range also requires tests on a complete turbine model.



DISCHARGE DISTRIBUTIC
AND FLOW ROTATION
BELOW THE RUNNER

DISCUSSION TO PAPER E1 BY CASSIDY AND FALVEY:

Authors' Reply by John J. Cassidy and Henry T. Falvey

It is both frustrating and rewarding to receive a wealth of discussion to one's paper. We have demonstrated a basic approach to the analysis of an old problem. Because the problem is indeed complex and has received attention from many investigators over at least four decades, it was natural that both favorable and unfavorable comments should arise. The authors' contention is simply that draft-tube surging is a fluid-flow phenomenon and as such must by analyzed in terms of fluid-flow parameters. The hydraulic turbine, as a machine, changes not only the energy content of the water as it passes through the runner but the momentum content and distribution as well. We will attempt to answer each discusser's comments in the light of our contention.

The prototype measurements plotted in accordance with our parameters and presented by Elder and Vigander were quite encouraging. However, in their Fig. 1 (Draft-tube pressure pulsations) the curve denoted as "curve through Falvey and Cassidy's model data" should not pass through the origin of the plot of fD^3/Q versus $\Omega D/\rho Q^2$. The critical value of $\Omega D/\rho Q^2$ as measured by the authors was 0.4 below which surging did not occur. A drawing showing the geometry of the Fontennelle draft is included in this reply. Unquestionably the shape of the draft tube has an effect upon both the occurrence and the magnitude of pressure surges. This effect of shape is one which should receive considerably more attention.

NOHAB's measurements as presented by Mr. Holmen provide an extremely gratifying support and simultaneously illustrate the importance of the plant σ or cavitation coefficient. The effect of σ was alluded to in the author's paper but could not be supported with data. Mr. Holmen's results show that as σ decreases the amplitude of the pressure surge decreases. It is the author's contention that as σ is reduced the flow becomes two-phase to an increasing degree. Because of the rotation in the draft all air and vapor migrate to the tube center. Apparently this effect stabilizes the vortex. This σ effect is in keeping with the field observation that

surging is alleviated when tail-water elevation is decreased or when air is injected into the draft tube. The authors deliberately chose air as the medium for our study in order to eliminate σ as one of the important parameters. However, detailed study of the effect of σ is certainly desirable and is a logical second step in the investigation of draft-tube surging.

Dr. Casacci makes several interesting points. In a given hydroelectric plant it is possible for resonance to occur in the penstock, the electrical network, the governor system, or in a combination of all. However, the draft-tube surge frequency is the driving frequency and its use is mandatory in a study of possible resonance conditions. For the Fontenelle-dam surge tanks prevented penstock resonance and the plant output of approximately 10 megawatts is negligible as compared to the connecting system. Resonance was not expected and, thus, it is not surprising that the maximum power swing coincided with the gate opening producing the maximum amplitude of pressure surge.

The calculation of $\Omega_2 D/\rho Q^2$ seems to raise difficulty but not in the way Dr. Casacci questions. Equation (6) is a momentum equation and is, thus, affected only by forces which produce a torque. Regardless of what the velocity distribution is at the draft-tube entrance $\Omega_2 D/\rho Q^2$ will be correct if $\Omega_1 D/\rho Q^2$ and turbine characteristics are correct. Equation (4) gives a reasonable value for $\Omega_1 D/\rho Q^2$ in all cases we have studied.

The effect of velocity distribution on the critical value of $\Omega_2 D/\rho Q^2$ is certainly a valid question. It was also of considerable concern in our study. In our complete study (Journal of Fluid Mechanics (1970), vol. 41, part 4, pp. 727-736) we deliberately varied the inlet geometry so as to vary the inlet-velocity distribution. No apparent effect on the critical value of $\Omega_2 D/\rho Q^2$ was noted. We conclude, therefore, that there is a unique flow pattern for a given value of $\Omega_2 D/\rho Q^2$ even though the development of the flow pattern may differ.

Dr. Casacci's statement that dynamic characteristics of shaft torque characteristics are difficult to link to the frequencies of the draft-tube surges is in contradiction to other studies. One

thing we have often noted is that pressure taps in prototype draft tubes frequently are located where eddies shed from upstream geometry are measured rather than the pressure surge of interest. Also, the dynamic effect of the connecting electrical system frequently plays an important part in determining the generator motion and hence the shaft torque.

Professor Raabe questions essentially the validity of our contention that "vortex breakdown" and draft-tube surging are one and the same. Our observations as seen in Fig. 3 show the flow pattern in each to be the same. Simultaneous measurements of pressure and velocity show that the unsteady pressure is due to an unsteady velocity. Observations of flow patterns at low velocities and pressures at high velocities show that surging occurs at the same value of $\Omega_2 D/\rho Q^2$ regardless of the magnitude of Q (with some allowances for Reynolds number effect). Prototype measurements and independent measurements in other laboratories support our contention. Therefore, we rest our case.

Placing a coaxial cylinder in a draft tube alleviates surging because it effectively decreases $\Omega D/\rho Q^2$ since the effective diameter D is decreased.

Finally, some observations should be made relative to further study of draft-tube surging. Fixed-blade turbines will always impart angular momentum to the flow entering the draft tube at some settings. It is logical that some changes in runners can be made to improve on surging characteristics. This, however, will probably reach the point of diminishing returns quite rapidly. It would appear that changes in draft-tube design offer the greatest possibility for improvement. In this light, the effect of divergence of the draft tube is currently under study. However, even more imaginative studies should be pursued. Presently, draft-tube designs are satisfactory only when the turbine is operating at maximum efficiency. However, operation is most frequent at conditions other than maximum efficiency for which flow is frequently upstream in one half of the draft tube. For high-head plants, surging might well be avoided entirely if no draft tube were provided and efficiency would be changed very little percentage-wise. In short, it would seem that less attention could be paid to the peak efficiency and more to other operational characteristics.

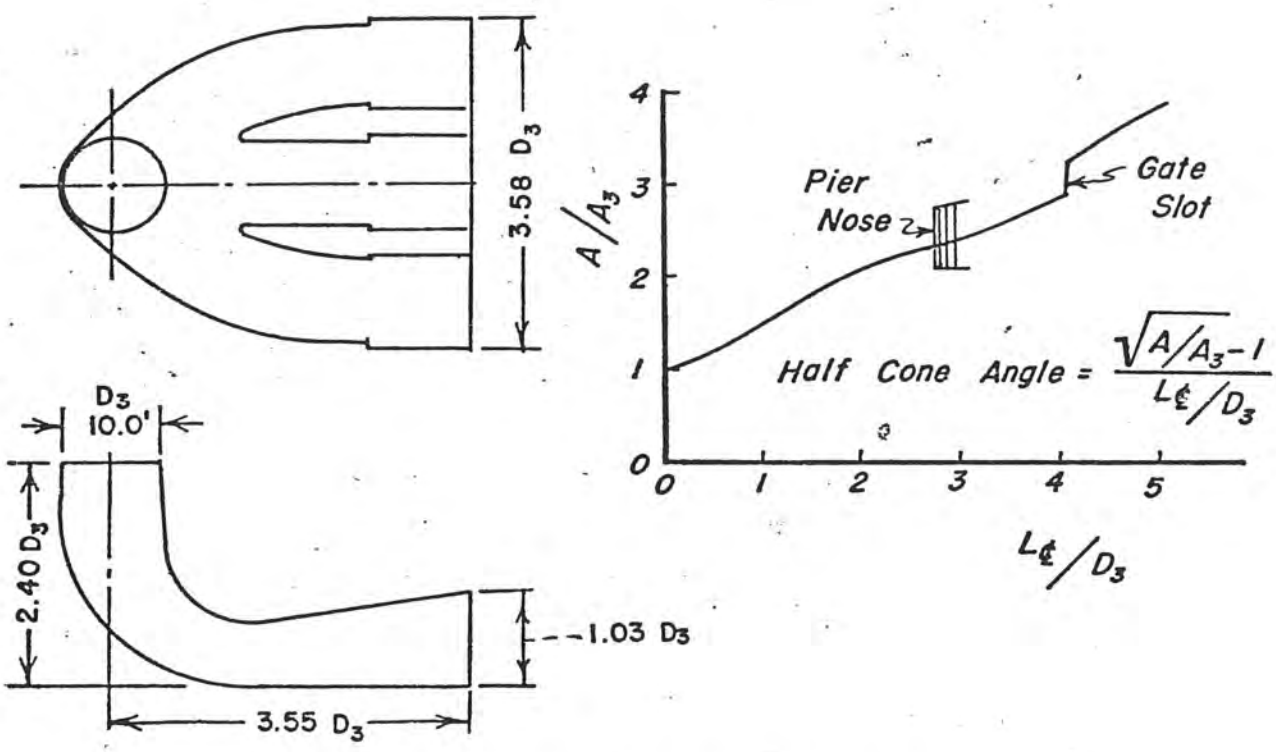


Figure 1 Fontenelle Draft Tube