01 Sep 1975

Flow Visualization Studies of Check Valve Vibrations

D. S. Weaver

F. A. Adubi

Recommended Citation

Weaver, D. S. and Adubi, F. A., "Flow Visualization Studies of Check Valve Vibrations" (1975). *Symposia on Turbulence in Liquids*. 23.
https://scholarsmine.mst.edu/sotil/23

FLOW VISUALIZATION STUDIES OF CHECK VALVE VIBRATIONS

David S. Weaver,
Department of Mechanical Engineering,
McMaster University,
Hamilton, Ontario, Canada.

Francis A. Adubi,
Nigerian National Oil Company,
Lagos, Nigeria.

ABSTRACT

A swing-type hydraulic check valve was found to vibrate continuously when a damper was applied to prevent slamming. A research program was conducted to develop an understanding of the phenomenon and thereby determine a method of alleviation. This paper describes the flow phenomena observed during flow visualization studies of a two dimensional full scale model of the valve. Such observations give insight into the mechanism of vibration.

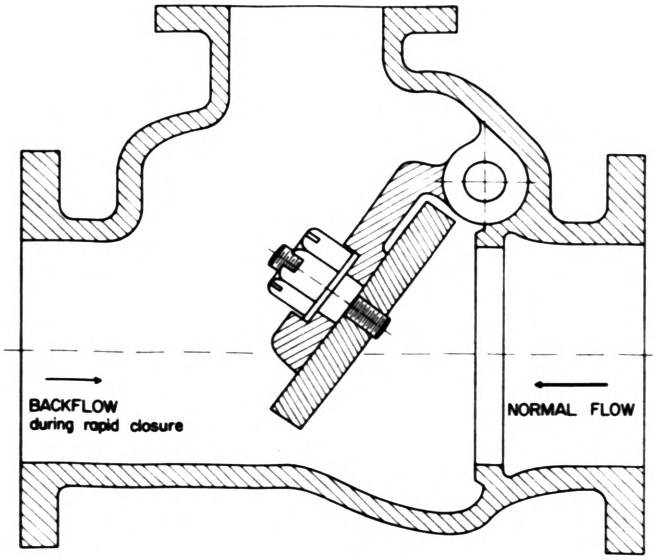

Figure 1: CROSS-SECTION OF CHECK VALVE

INTRODUCTION

Violent slamming of non-return or check valves with the attendant water hammer is a common problem in hydraulic systems. One obvious solution is to attach a spring-damper mechanism to the valve clapper arm to reduce the rate of closure. However, in the case of the swing check valve shown in figure 1, such a damper proved ineffective. When the damping was sufficiently large to prevent rapid closure, small amplitude limit cycle oscillations, and hence multiple slammings, were produced. Increasing the spring stiffness served only to reduce the frequency and increase the amplitude of oscillation (Adubi, 1975).

Interestingly, similar phenomena are found in certain mechanically operated sink and bathtub stoppers as well as rubber seals on hydraulic gates. These vibrations are extremely persistent and accompanied by a loud chattering or humming sound. In fact, the vibration of J-bulb rubber seals is so commonplace that they are often referred to as musical note seals. A discussion of such problems as well as a number of references is included in the paper by Weaver (1974).

These phenomena are hydroelastic, resulting from the mutual interaction of inertia, elastic and hydrodynamic forces, the motion being perpetuated by the transfer of energy from the flow to the structure. Our inadequate understanding of this behaviour is demonstrated by our apparent inability to design vibration free components and by numerous abortive attempts to alleviate the difficulties; see, for example, Schmidgall (1972) and Wonik (1972). While most of the literature on seal vibrations has reported field observations, at least one paper, Lyssenko and Chepajkin (1974), has attempted to model the vibrations mathematically. However, the latter is really a negative damping model based on an assumed linear discharge coefficient and does not appear capable of representing the unsteady flow phenomena occurring during vibration. In particular, it cannot represent the rapid reduction in discharge at closure, nor the known effect of varying spring stiffness. (Increasing the stiffness of a damped simple harmonic oscillator, increases the frequency of oscillation which is contrary to experimental observations in this case.)

Experiments with the prototype check valves were plagued with mechanical failures because of the severity of the slamming. In addition, such experiments could only provide limited information regarding the flow behaviour during oscillation. As a result, a two-dimensional model was constructed which would ensure structural integrity during experimentation and permit flow visualization. Experiments were then conducted in order to determine the fundamental nature of the vibration phenomenon and hence ascertain a simple method of alleviation. This paper presents the results of the flow visualization studies carried out as part of this research program.

TWO DIMENSIONAL MODEL

Flow visualization required the construction of a model and since both construction and experimentation would be a great deal simpler, the model was made two-dimensional. Of course, this assumes that any three-dimensional flow effects in the neighbourhood of the circular valve disc and seat are not a significant part of the vibration phenomenon. It appears that this is not an unreasonable assumption as the vertical cross-section through the valve as shown in figure 1 is also a plane of symmetry. Subsequent tests validated the use of this model as both the static discharge characteristics and dynamic stability behaviour of the model and prototype were very similar.

Details of the model construction are shown in figure 2. The width was made 50% greater than the nominal 6-inch diameter of the valve being modelled. While the upper blocks were made of aluminum, the bottom section was made of maple (hardwood) to facilitate fabrication and inclusion of the 2-inch perspex lamination. The use of wood in such applications is not recommended as, in spite of precautions taken to obtain properly dried hardwood and seal out the moisture, considerable reinforcement was necessary to prevent warping.

The length of the model was 24 inches with a 36-inch rectangular 6" x 9" section upstream and downstream of the model. Flow straighteners and screens were used at the beginning of the upstream duct. The front, back and top of the model were constructed of 1-inch thick perspex, bolted and sealed with silicone sealent. In addition, the upstream and downstream ducts were drawn together by four tension bolts, compressing the model between them. The latter was thought necessary to prevent excessive tensile loads on the flanges due to the water hammer waves produced at each vibration cycle.

The experiments were conducted with the flow in the "backflow" direction and sustained by a reservoir which held the head constant at about 11 feet. The valve disc was held at some small initial angle between 1 and 7 degrees by a spring assembly attached to the disc arm. This permitted a limited and restrained swing of the disc as well as the ability to vary spring stiffness.

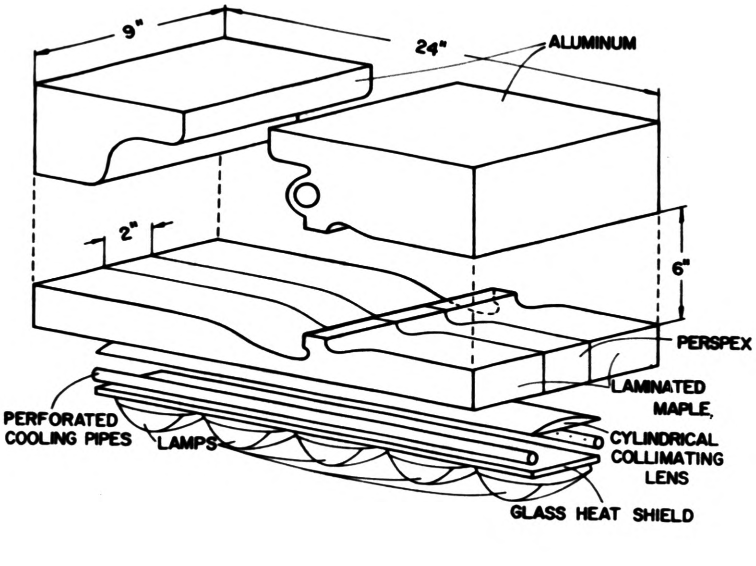

Figure 2: SCHEMATIC OF FLOW VISUALIZATION MODEL

FLOW TRACER AND ILLUMINATION

Of the many flow visualization techniques available (see, for example, Merzkirch, 1974), the use of aluminum powder as a tracer was considered most appropriate in view of the turbulence and unsteady flow. The particles used were aluminum flakes A9432 produced by Canbro Division of International Bronze Powders Ltd. This material is specified to give a 20% retention on a Tyler 325 mesh, i.e., most of the particles will be a little smaller than 0.043 mm.

The powder was wetted using methyl alcohol and then made into an aqueous solution in an injection tank. This tank was then pressurized at 15 psig and the tracer injected through a series of 3/32-inch diameter holes in a 1/4-inch pipe positioned vertically about 30 inches upstream of the valve disc. A valve was used to control the injection rate. This system worked quite well and no difficulties with aluminum particles adhering to the model surfaces were encountered.

As the direction of maximum light scattering from the tracer is at an angle of about 90° from the direction of the incident light, the flow was illuminated from below through the perspex lamination. A plano-convex cylindrical collimating lens was machined from a 5-inch diameter cast acrylic rod. This provided a thin vertical sheet of light along the centre of the model.

Light was provided by five 300 watt Kodak Carousel projector lamps as shown in figure 2. As these also produced considerable heat, cooling was provided by two perforated cooling pipes connected to a 20 psi compressed air supply. In addition, the lens was protected by a 1/4-inch thick heat resistant glass plate.

Photography was done using a Bolex 16 mm. cine camera with a 25 mm., f1.1 lens. This camera allowed the framing rate to be varied from 1 to 64 frames per second. Kodak Plus X negative film was found best for this particular application.

EXPERIMENTAL OBSERVATIONS

Numerous experiments were conducted using different initial angles of valve opening and spring stiffnesses. During each experiment, the pressure difference between the upstream and downstream side of the valve plate, the angular displacement of the plate and the reactive load on the restraining spring were monitored. At the same time, films were taken of the flow pattern during oscillation.

Precise displacement measurements were made using a capacitive probe together with a spring-loaded piston operated by a cam on the valve shaft. Pressure measurements were obtained using a Pace Engineering Co. Model P7D differential pressure transducer. The diaphragm probes were placed flush with the back wall of the model about 1/2" up from the bottom, one about 1/2" downstream (in the backflow direction) and the other about 2" upstream of the valve seat. All instruments were carefully calibrated, had a flat response in the frequency range of interest, and records were taken using a U-V strip chart recorder.

Typical results are shown in figure 3. Clearly, the motion is not sinusoidal and, incidently, the frequency of oscillation is considerably less than the natural frequency of the valve-spring assembly in a quiescent fluid. The valve closes and remains closed until the dynamic pressure difference across the valve is sufficiently low to allow opening. The opening portion of the cycle is considerably more rapid than the closing part. As the valve moves towards the seat, the pressure gradually increases until closure occurs and the water hammer produces a large pressure difference across the valve once again. There is some extraneous noise on the pressure signal but the large spikes occurring while the valve is closed are caused by cavitation and movement of the whole test rig due to the water hammer waves.

The photographs in figure 4 show the flow pattern during a cycle of vibration with emphasis on the flow on the upstream side of the valve. The numbers on each

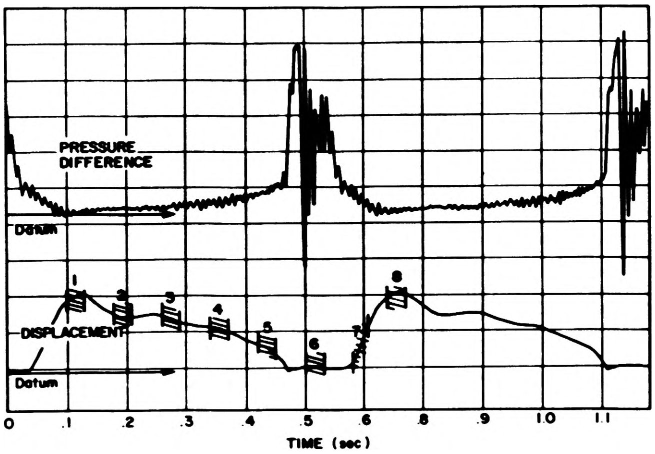

Figure 3: HYDRODYNAMIC PRESSURE DIFFERENCE ACROSS VALVE AND DISC DISPLACEMENT RECORDS OVER TWO CYCLES OF OSCILLATION

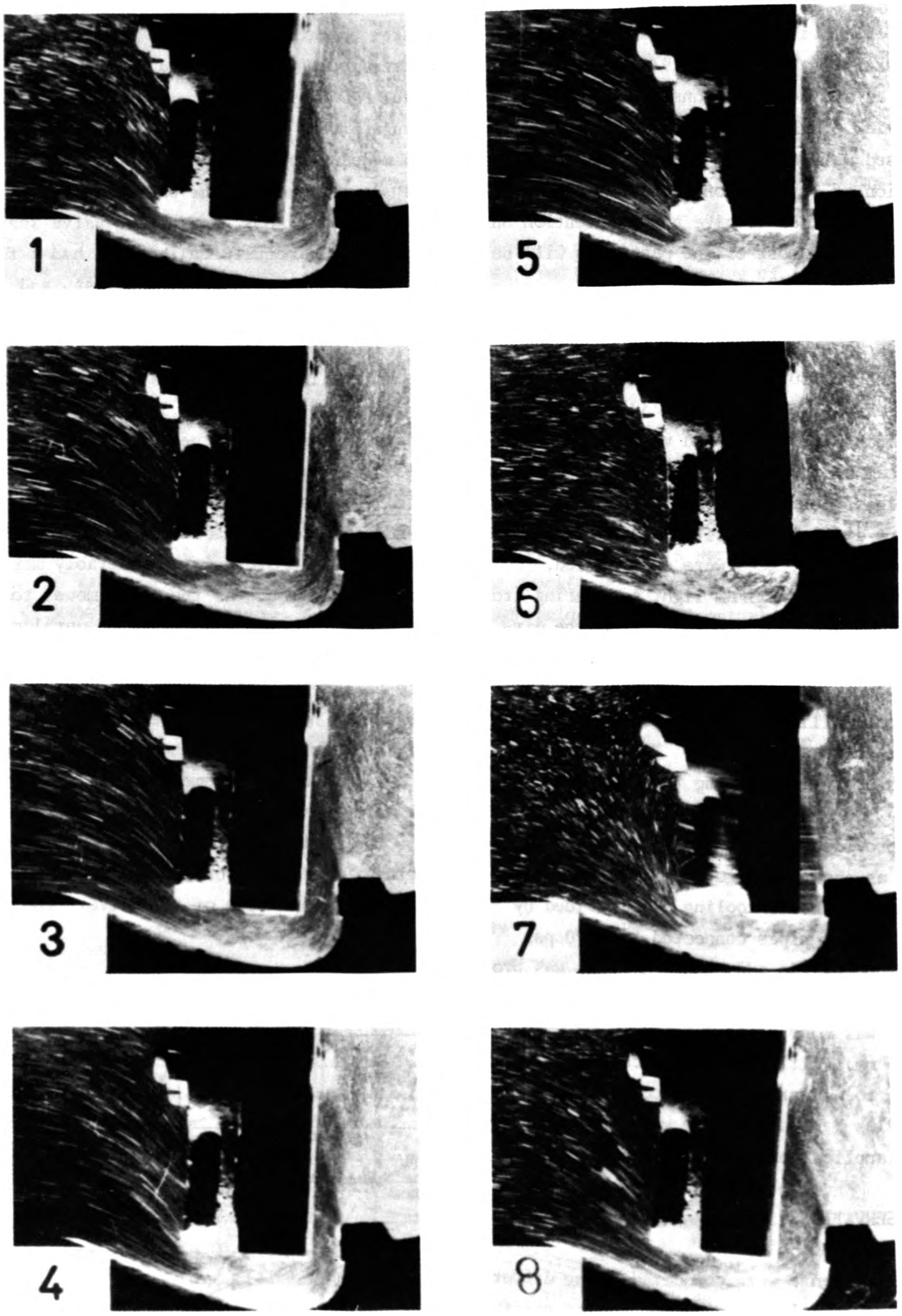

Figure 4: SAMPLE PHOTOGRAPHS OF FLOW PATTERN OVER CYCLE OF OSCILLATION (framing rate = 12 frames per second)

photograph refer to the portion of the cycle noted in figure 3. These photographs also illustrate the problems associated with recording flow patterns when the fluid boundaries are moving and large variations in flow velocity occur in the field of observation. If the flow velocity is of the same order of magnitude as that of the valve plate, any exposure time which "stops" the valve produces no streak pattern. On the other hand, a photograph showing a distinct streak pattern may necessarily also show a rather indistinct valve plate as seen, for example, in frame 7 of figure 4. In such cases, the "average" valve location was obtained by reference to the synchronized displacement record (figure 3) rather than measurements taken from the photographs. This also explains why it was found more instructive to study different parts of the flow field separately as is discussed below.

The difference in flow velocity between the opening and closing part of each cycle is evident in these and similar photographs. While it was never possible to obtain distinct enough particle streaks in the slot between the valve and its seat to determine precisely the flow velocity there, this was not true upstream of the valve. As a result, the average velocity in the apron across an arc about 15 degrees upstream of the seat was obtained by measuring the streak length in films taken at 12, 24 and 48 frames per second. These data are shown in figure 5. Clearly, the inertia of the fluid produces a much greater flow velocity during the closing portion of the cycle than at the same angle during the opening part. This hysteretic effect is seen in the pressure difference records as well and leads to a net energy transfer from the fluid to the structure during each cycle.

The flow velocity at zero angle of opening in figure 5 is due to leakage, mostly past the sides of the valve plate. It was necessary to leave some clearance between the plate and the sides of the model to avoid friction and scratching of the transparent perspex.

A series of films were taken closer to the model in order to examine more carefully the flow in the slot between the valve and its seat. Examples are shown in figure 6 where both the angle of opening is given as well as the frame number. The photographs are paired at the same angles to facilitate comparison. It is readily seen that the high velocity jet in the slot is more nearly parallel to the valve plate during closure than during opening where the jet is seen to impinge on the downstream side of the plate. These local flow effects undoubtedly contribute to the difference in hydrodynamic load on the valve between opening and closing.

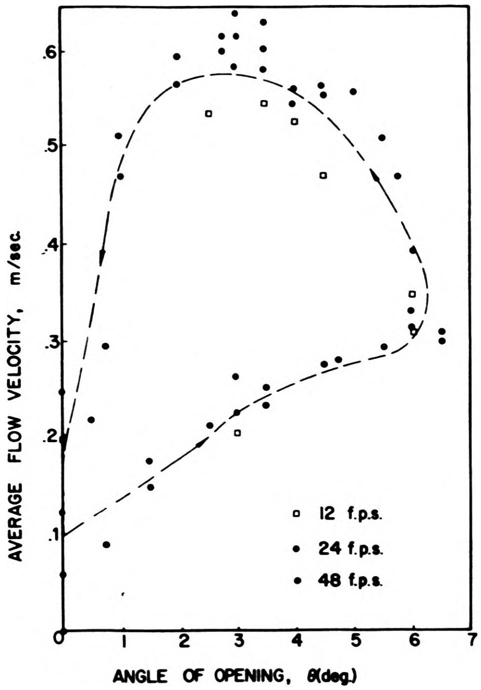

Figure 5: VARIATION OF FLOW VELOCITY IN VALVE OVER VIBRATION CYCLE

It was suggested by Wood (1972) that perhaps a vortex in the downstream cavity plays an important role in the vibration phenomenon similar to that in the mammalian antic heart valve. In order to examine this possibility, a further close-up series of films were taken, an example of which is shown in figure 7. This shows that, as the valve begins to close, a vortex is formed which rolls up and is swept downstream shortly afterward. While this coincides with a momentary pause in the valve's motion, it apparently plays no significant role in the vibration mechanism.

It is not surprising to see a vortex created as the valve plate changes its direction of motion at maximum amplitude. However, this vortex is not responsible for either opening or closing the valve. The reflected water hammer waves and the valve spring control the initiation and opening portion of each oscillation. The momentary pause after full opening (frames 14-16) results from the overshoot of the valve plate past the zero deflection angle of the restraining spring and the subsequent settling down. Up to this point the maximum flow rate has not yet been

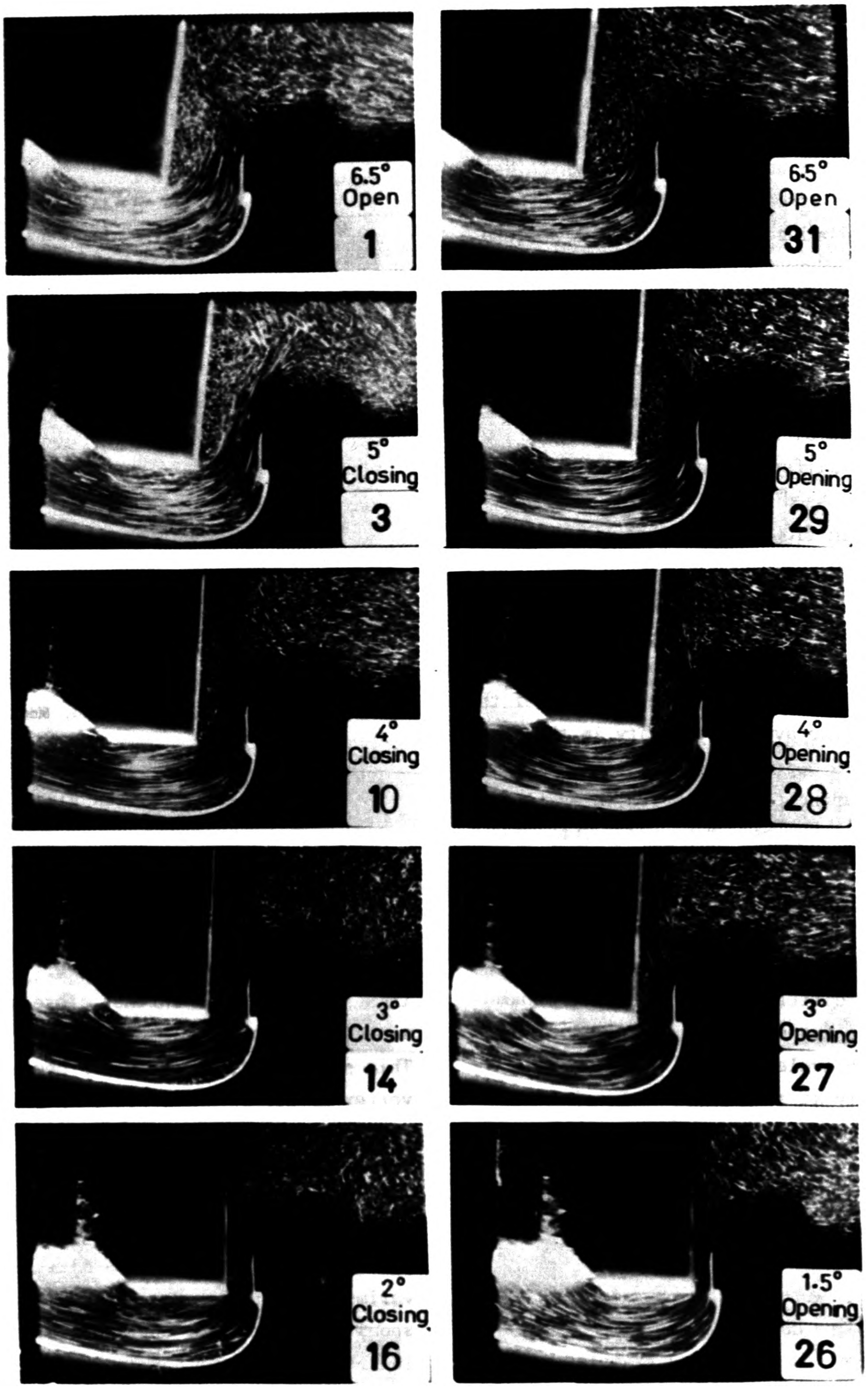

Figure 6: COMPARISON OF FLOW PATTERN AT VARIOUS ANGLES OF DISC DURING OPENING AND CLOSING (framing rate = 64 frames per second)

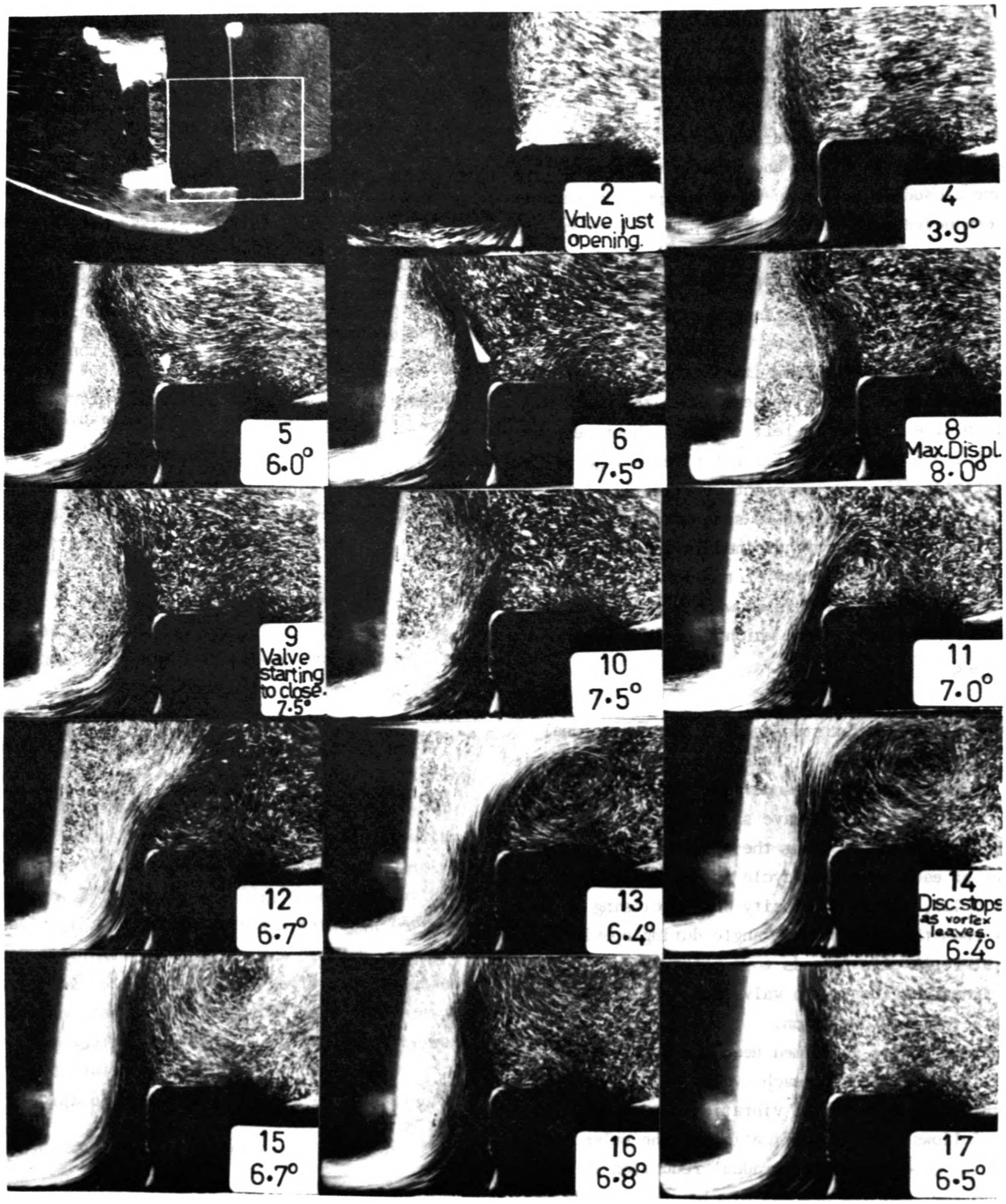

Figure 7: SEQUENCE OF PHOTOGRAPHS SHOWING DEVELOPMENT OF VORTEX
(framing rate = 64 frames per second)

established and hence, the behavior is still essentially the natural response of the spring-valve system. By frame 17, the flow is sufficiently reestablished that it now begins to dominate the behavior of the valve which commences to drift shut under the increasing hydrodynamic load. Closure is controlled by the increase in pressure drop across the valve as the angle of opening decreases and finally by the inertial pressure caused by reduction in discharge. In experiments conducted with weaker springs, the overshoot and pause seen in this sequence of photographs did not occur. In such cases the vortex is still formed but, just as shown above, it has long been swept away by the time final closure occurs.

Note also that the flow immediately above the downstream side of the valve seat is in the upstream direction, i.e., right to left. This is best seen by the fortuitous appearance of a relatively large piece of material at the extreme right of frame 2. In frames 4 and 5, this object is seen moving to the left until, in frame 6, it is entrained in the jet and swept away. An overall view of the downstream side of the model shows that this flow is a portion of a large recirculating pattern which fills the duct.

CONCLUSIONS

Flow visualization studies of a two-dimensional model of a vibrating swing check valve have been carried out in order to develop an understanding of its dynamic behaviour. These studies have shown that:

1.) The flow pattern during the opening and closing portions of each vibration cycle is quite different. In particular, the flow velocity during closing is much greater than that at the same angle during opening. This hysteretic effect leads to a net transfer of energy from the flow to the valve during each cycle, thus perpetuating the motion.

2.) While a vortex is formed near the downstream side of the valve plate during each cycle, it does not play any significant role in the vibration phenomenon.

It follows that a modification of the valve geometry which produces a more gradual reduction in discharge in the last few degrees of closure, as well as eliminates the hysteretic hydrodynamic pressure effect in this region, will prevent this vibration phenomenon. These observations led to simple geometric alterations to the valve disc and seat which proved successful in eliminating the vibration problem completely.

REFERENCES

1. Adubi, F. A., 1975, "The Hydroelastic Vibration of a Hydraulic Swing Check Valve", Ph.D thesis, McMaster University, Hamilton, Ontario, Canada.
2. Lyssenko, P. E. Chepajkin, G. A., 1974, "On Self-Excited Oscillations of Gate Seals" IUTAM/IAHR Symposium on Flow-Induced Structural Vibrations, Springer-Verlag, 278-296, (Ed. E. Naudascher).
3. Merzkirch, W., 1974, "Flow Visualization", Academic Press Inc., New York.
4. Schmidgall, T., 1972, "Spillway Gate Vibrations on Arkansas River Dams", ASCE Jo. Hyd. Div., HY1, 219-238.
5. Weaver, D. S., 1974, "On Flow Induced Vibrations in Hydraulic Structures and their Alleviation", Second Symposium on Applications of Solid Mechanics, McMaster University, Hamilton, Ontario, Canada, 400-429.
6. Wonik, G., 1972, Discussion of reference 2 above, IUTAM/IAHR Symp., Karlsruhe, Germany (Ed. E. Naudascher).
7. Wood, C. J., 1972, Discussion of paper by Weaver, Kouwen and Mansour, IUTAM/IAHR Symp. Karlsruhe, Germany (Ed. E. Naudascher).

DISCUSSION

Rober King, BHRA: We had an almost identical problem with a large river control flap gate. The operator was rather worried about the steady drag forces and the discharge coefficient and we tested a model gate in the laboratory. Because I was interested in vibrations I put a spring in the operating mechanism to simulate the elasticity of the hydraulic lifting gear. Sustained oscillations of the gate were observed at small angles of opening. The way I explained it at the time was that when the gate was cracked open the jet at the back of the gate would cause a low pressure bubble to form and this tended to close the gate slightly and the increased heat across the gate caused the gate to shut. With the gate shut, the flow ceases and the low pressure region behind the gate is removed. Because the spring has been extended, the spring snatches the gate back again, to the original cracked open position and the cycle is repeated. We too, overcame that in the same way as Dr. Weaver by redesigning the flow paths.

Weaver: That's true! The sort of mechanism that you suggest is qualitatively what causes the problem. The real difficulty is associated with the fact that during the opening portion of the cycle, the flow velocity and therefore, the pressure difference across the structure is considerably less than during the closing portion. If the flow were the same, there would be no net energy transfer from the fluid to the structure. I haven't seen it examined before, but the problem is common enough, I think. Typically, especially with rubber seals, the intuitive solution has been to increase the stiffness which makes the problem worse. In fact, in the case of hydraulic seals, the solution is to make the seal so flexible that once it closes, it can't open again. While increased flexibility works for hydraulic seals, it wouldn't be of any help in this particular problem.

V. W. Goldschmidt, Purdue University: Which are the proper scaling parameters?

Weaver: Valves such as that modelled here are produced in several geometrically similar sizes from 6 inches up to something in excess of 12 inches. Hence, the model is a two-dimensional full scale replica of the smallest production valve. The geometric distortion resulting from the two-dimensionalization of the model was based on the conjecture that the three-dimensional flow effects produced by the spherical valve body and circular disc valve were unimportant to the phenomena being studied. Subsequent comparisons of model and prototype discharge characteristics and dynamic behavior have borne this out.

Other important scaling parameters are Reynolds number, Strouhal number, and Euler number, the latter being essentially the ratio of hydrodynamic load to velocity head. With the exception of the effect of the geometric distortion on Euler number, these parameters were scaled properly.