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# INTERMITTENT-FLOW COEFFICIENTS OF A POPPET VALVE

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## SUMMARY

Flow coefficients were determined for the inlet valve of a modern air-cooled cylinder during operation of the valve. The cylinder head with valves was mounted on a large tank that could be evacuated. Operating the valve with a rotating cam allowed air to flow through the valve into the evacuated tank. The change of pressure in the tank was a measure of the amount of air flowing through the valve in a given number of cycles. The flow coefficients were determined from the pressure across the valve, the quantity of air flowing, and the valve-lift curve. Coefficients were measured with lifts of 0.1 to 0.6 inch and speeds of 130 to 1,200 r.p.m. The results obtained with intermittent flow were compared with the results of tests made with steady flow through this cylinder head. This comparison indicated that steady-flow coefficients can be used for intermittent flow.

## INTRODUCTION

The inlet air or the combustion mixture in the conventional internal-combustion engine flows intermittently through poppet valves into the cylinders. The air velocity through the valves is zero when the valves start to open and builds up at some later time in the cycle to the maximum velocity attainable with the available pressures. The flow coefficients with valves held open and a continuous flow of air have been measured by several investigators (references 1, 2, and 3). Steady-flow coefficients have been considered applicable to intermittent flow in short inlet passages (reference 4) because the mass of air to be accelerated is very small.

Aschenbrenner (reference 5) gives a mathematical treatment of unsteady flow through systems with varying cross sections. His work being entirely theoretical, it is difficult to apply to poppet valves and gives little information about intermittent-flow coefficients.

Lucke (reference 6), in an investigation of the flow of air through poppet valves under both intermittent-flow and steady-flow conditions, computed the coefficients for the steady flow but not for the intermittent flow. The intermittent flow was investigated by motoring an engine and measuring the air consumption with a meter. The engine and the meter were connected by long pipes in which pulsations undoubtedly existed; such pulsations would make the computation of coefficients difficult and subject to large errors.

When an attempt is made to check steady-flow coefficients with intermittent-flow coefficients for an engine, many complicating factors are introduced, such as piston motion, volume of residual exhaust gas, engine temperature, and pulsations in the cylinder and the inlet manifold. It is also difficult to obtain accurate simultaneous values of the air pressures on the inside and the outside of a valve on an engine in operation.

The purpose of this work is to eliminate as many of the complicating factors as possible, to determine whether intermittent-flow coefficients actually are the same as steady-flow coefficients, and to find how the coefficients for each value of valve lift should be applied to the entire induction period.

## APPARATUS

The apparatus used in this investigation (see fig. 1) consisted of a large tank, a vacuum pump, a cylinder head, a valve-operating mechanism, and a micromanometer.

The tank has a capacity of 81.65 cubic feet, is rigid and almost airtight, and is equipped with connections for the micromanometer, the vacuum pump, and the cylinder head. The temperature of the air inside the tank was measured by a thermometer. Room temperature was measured by a thermometer near the apparatus.

The vacuum pump was an electrically driven reversed air compressor, capable of quickly evacuating the tank to the vacuum of 16 inches of mercury used in the tests.

The cylinder head, from a Wright 1820-G cylinder, was equipped with valves and springs. The cylinder end of the head was mounted with an airtight joint over a large hole in the side of the tank. The exhaust valve was held closed and airtight by the springs, and the spark-plug openings were sealed. The intake valve was operated by a push rod from the valve-operating mechanism.

The valve-operating mechanism was driven by a 3-horsepower variable-speed electric motor connected by a belt to a shaft having a very heavy flywheel (moment of inertia of 3 lb. ft. sec.$^2$). This shaft could be driven at steady speeds ranging from 100 to 1,200 r.p.m. It was connected to a clutch, similar to those used on punch presses, that gave a driven shaft one or more revolutions, as desired. The clutch was engaged by an operating handle that could be held back for any desired number of valve cycles. Mounted on the end of the driven shaft was a revolution counter that indicated the number of valve cycles. The heavy flywheel was installed to keep the speed decrease as low as possible each time the clutch apparatus operated. The valve-operating cams were mounted on the driven shaft of the apparatus.

During the investigation, several cams were used. Two of the cams had maximum lifts of 0.2 inch and 0.4 inch with 40° dwell and were used in the determination of coefficients for the 0.2- and the 0.4-inch lifts at 130 and 1,200 r.p.m. For the tests at the 0.1- and the 0.3-inch lifts, the tappet roller was given a clearance of 0.1 inch and contacted the cam after the lift of the cam had started. Cam acceleration and rotational speed had to be low to prevent excessive shock loading. These cams had low acceleration so that flow coefficients could also be measured at the 0.1- and 0.3-inch lifts at 130 r.p.m. Flow coefficients for a 0.6-inch lift at 1,200 r.p.m. and for a 0.5-inch lift at 130 r.p.m. could also be determined by means of another cam with a 0.6-inch lift and 40° dwell. A fourth cam had a lift curve similar to that of the cam on the Wright 1820-G engine and thus provided a means of checking the application of the coefficients for different lifts to an engine-valve lift curve.

A 1-1/4 inch roller in a tappet bore on the cams. Between the tappet and the end of the intake valve stem was an adjustable push rod. There was no rocker arm in the cylinder head; and the tappet, the push rod, and the intake valve were all in a straight line.

Moving pictures of the cam and the roller were taken at a rate of 1,500 frames per second to make sure that the roller followed the cam during the most severe conditions of operation. No bounce or throw off of the roller was observed.

The special micromanometer, described in reference 7, was connected to the tank and is capable of registering a maximum pressure difference of 16 inches of mercury between the outside and the inside of the tank. The instrument gave readings correct to within ±0.004 inch of mercury.

Figure 2 is a cross section of the valve, the seat, and the intake passage of the head used. The seat angle is 30° and the approach has been faired smoothly into a rounded section in the intake passage, giving the valve opening a rounded entrance. The intake passage is very short and a rounded mouthpiece was placed at its entrance. Very little resistance to flow should have been offered by the intake passage and its entrance.

Figure 3 shows the apparatus used for measuring coefficients with steady flow. The blower forced air through an accurately calibrated orifice into the tank upon which the head was mounted. The same rounded intake passage entrance was used that was used when measuring coefficients with intermittent flow. A manometer and a thermometer measured the pressure and the temperature of the air in the tank. The valve was held open by a screw and lock nut and the lift was set with a micrometer to ±0.0005 inch.

## METHODS

The method of measuring coefficients for intermittent flow was to evacuate the tank shown in figure 1, to operate the valve at any desired speed for any desired number of cycles, and then to find the change in pressure in the tank caused by the air that had entered through the valve. From this change of pressure and the known volume of the tank, the weight of the air that flowed through the valve could be determined. From this weight of air, the pressure across the valve, and the valve-opening area, the coefficients of flow were determined.

The true change of pressure was found by measuring with the micromanometer the pressure in the tank before the valve had been operated and then, after the valve had been operated, by taking accurate readings of pressure in the tank at intervals of 2 or 3 minutes. The first few readings showed the pressure in the tank to be decreasing. Succeeding readings showed it to start rising and to continue doing so at a steady rate. The initial decrease was probably due to the cooling of the air after it had been compressed and heated by the air entering the tank during the valve operation. The subsequent slow rise of pressure was caused by air leaking into the tank.

Figure 4 is a plot, with an exaggerated scale, of pressure in the tank against time. The pressure in the tank before the valve operation is indicated by A. The first reading of pressure after admitting air by operating the valve was taken at B and readings taken as the pressure decreased are shown at C, D, and E. Readings taken during the final slow rise of pressure caused by the leakage are indicated at E to K. Readings E to K always fell almost on a straight line. Extending a straight line from K through E to X, which is directly above A, extends the leakage curve back to zero leakage time and shows what pressure would have existed in the tank immediately after the valve operation if there had been no leakage nor temperature change. The distance between X and A represents the true change in pressure caused by air flowing through the valve into the tank. The vertical distance between A and X was usually about 40 times as great as the vertical distance between X and B.

Flow coefficients vary with the amount of valve lift. Valves cannot be lifted in zero time to large lifts. The flow therefore always passes through conditions to which coefficients for low lift apply before it reaches the condition at which the coefficient for large lift applies.

The method of making allowance for the coefficients applicable to low values of valve lift can be explained by the use of figure 5. This figure is a plot of valve-opening area against crank degrees when the maximum valve lift was 0.2 inch. The points b and e correspond to 0.1-inch valve lift. For purposes of explanation, it will be assumed that the coefficient for 0.1-inch valve lift has been determined and that the coefficient for 0.2-inch valve lift is to be determined. The coefficient for 0.1-inch lift is applied to the areas abj and efg by the use of the following equation (reference 8):

$$w = 158.17 \text{ Atc} \sqrt{\frac{(p_1 - p_2)\ [p_2 - 0.0755\ (p_1 - p_2)]}{T_1}} \tag{1}$$

where

W is weight, pounds.

A, opening area, square feet.

t, time of flow, seconds.

At, area under the area-degrees curve of the valve, multiplied by the proper factor to change crank degrees to seconds.

c, flow coefficient.

$p_1$, pressure of air before going through valve, pounds per square inch.

$p_2$, pressure of air after going through valve, pounds per square inch.

$T_1$, absolute temperature of air before going through valve, °F.

The flow area is the lateral area of the frustum of a cone having sides perpendicular to the valve seat. This area is given by the following equation:

$$A = 2\pi \left(R_o + \frac{L}{2} \sin \alpha \cos \alpha\right) L \cos \alpha \tag{2}$$

where

L is the valve lift, inches.

$\alpha$, seat angle, degrees.

$R_o$, radius of inside of seat, inches.

The weight of air flowing through the valve while it is lifting from a to b and dropping from e to f is obtained by multiplying the area abj + efg by the proper factor and by substituting the product for At in equation (1). The flow coefficient for 0.1 inch is substituted for c and the proper values of $p_1$, $p_2$, and $T_1$ are inserted under the radical.

The weight of air flowing through the valve while it is lifting from b to c and falling from d to e is obtained by multiplying the area bcij + degh by the proper conversion factor and substituting the product for At in equation (1). The value of c to be used is an average between the known value for the 0.1-inch lift and the unknown value C for the 0.2-inch lift.

The unknown value C applies to the remaining area cdhi.

The total weight of air that will be admitted by a valve operating with this lift curve can be found from experiment; C is thus the only unknown and can be computed.

In the computation of Reynolds Number for the air flowing through the valve, the values of density, velocity, and viscosity were averages of the values for the air before and after it had passed through the valve. The value of d (diameter) used was $\sqrt{\frac{4 \times \text{opening area}}{\pi}}$.

## TESTS RESULTS AND DISCUSSION

The first tests were made with a valve lift of 0.1 inch. The cam was operated at a speed of 130 r.p.m. and a dwell of 40° at maximum lift. This combination of long dwell with low lift minimized the effect of the opening and the closing of the valve so that the over-all coefficient obtained was considered to be the true coefficient for 0.1-inch valve lift. Figure 6(a) shows the experimental values of c obtained with different values of pressure difference across the valve.

The average of the values for c was 1.00. Many of the values were greater than 1.00. This fact probably was a result of the well-rounded approach to the valve opening, which caused the air to expand after passing the entering-seat edge. The static pressure of the air at the entering-seat edge was therefore lower than that at the exit-seat edge. In order to obtain the correct flow coefficient with a flow area computed from the inside-seat radius, this lower static pressure should have been used; the coefficients would then have been lower than those presented in figures 6 and 7. The correct flow area with the tank pressure as the discharge-side static pressure would have been the lateral area of the frustum of a cone constructed on the outside-seat diameter. Results presented by other investigators (references 1, 2, and 3) have, however, been based on inside-seat diameter, and this diameter was therefore used for figures 6 and 7 to make the results directly comparable. If outside-seat diameter had been used for figure 6(a), the value of c would have been 0.86 instead of 1.00. This ratio approximately holds for all other coefficients.

Figure 6(b) gives the results of tests made at 130 and 1,200 r.p.m. with a valve lift of 0.2 inch obtained with a cam having a total lift of 0.2 inch. The results at 130 r.p.m. were not appreciably different from those obtained at 1,200 r.p.m. The points are experimental values, which are the over-all coefficients and include the opening and the closing of the valve; the dashed line is their average. The solid curve is the coefficient C that applies during the time the valve is at its maximum lift and was derived in the manner already described in the section on Methods. The values of c for the 0.1 inch lift were obtained from figure 6(a). With 0.2-inch valve lift, c was not the same for all pressure differences and reached a maximum of 1.14 at a pressure difference of 5 inches of mercury. At 11 inches of mercury, it had dropped to 1.07. With an opening area computed from the outside-seat diameter, c was 0.98 at 5 inches and 0.92 at 11 inches of mercury.

Figure 6(c) gives the results obtained with a valve lift of 0.3 inch and a speed of 130 r.p.m. When the over-all coefficients were corrected for the coefficients for 0.1 and 0.2 inch that were effective during the opening and the closing periods, the coefficients for the 0.3-inch lift as shown by the solid curve were lower than the over-all coefficients, the maximum for the 0.3-inch lift being 1.03 at 3-1/2 inches of mercury. Figures 6(d), 6(e), and 6(f) are results obtained with cams having lifts of 0.4, 0.5, and 0.6 inch, respectively.

Figure 7 shows a comparison between flow coefficients obtained with steady flow and with intermittent flow for valve lifts of 0.1 to 0.4 inch. Steady-flow tests had an exit pressure equal to atmospheric; whereas, intermittent-flow tests had an inlet pressure equal to atmospheric. The results have been plotted against Reynolds Number for comparison. The steady-flow and the intermittent-flow coefficients are not in good agreement but do lie in the same region. It probably is safe to conclude that, for engineering purposes, steady-flow coefficients can be used for intermittent-flow conditions.

Figure 8 gives the over-all coefficients obtained with a cam having no dwell and a lift curve similar to that of an inlet valve from a Wright 1820-G engine. Runs were made at speeds of 130, 600, and 1,200 r.p.m. and the coefficients obtained were approximately the same, showing that flow conditions in a valve build up so rapidly that they are independent of lifting rate.

The solid curve below the curve for 130 r.p.m. was obtained by applying to the lift curve of this cam values from figure 6 in the manner described under Methods. The curve of computed over-all coefficients falls very close to the curve of experimentally derived over-all coefficients.

## CONCLUSIONS

1. Flow coefficients measured with steady-flow conditions can be applied to intermittent-flow conditions with no correction for valve-opening speed.

2. The over-all flow coefficient of a valve with any lift curve can be accurately computed by the method herein described if the flow coefficients of a similar valve are known at values of lift of 0.1, 0.2, 0.3 inch, and so on.

3. When a valve seat has a well-rounded approach, flow coefficients above 1.00 can be obtained if the flow area is computed from the inside-seat diameter.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., March 27, 1939.

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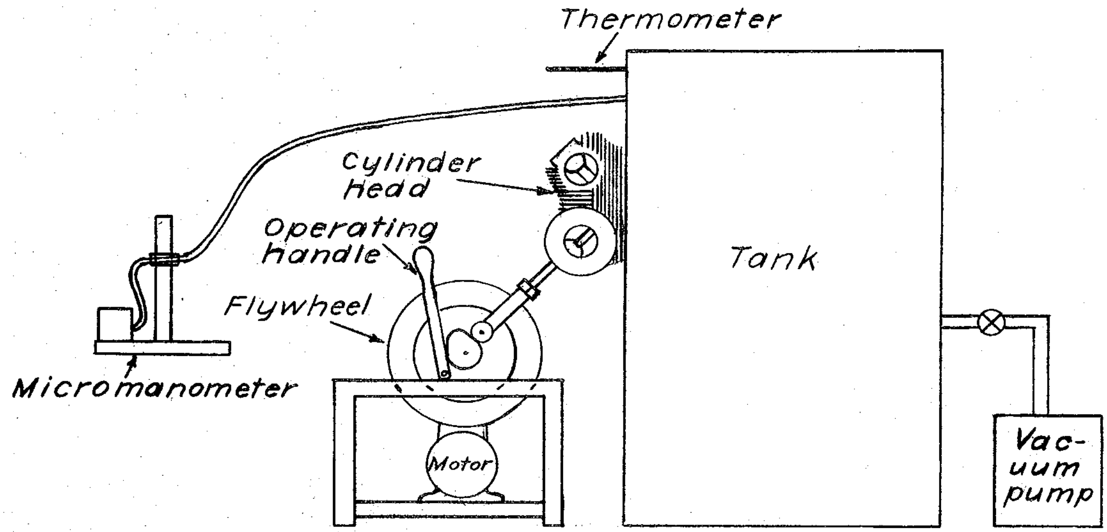


Figure 1. - Diagrammatic sketch of intermittent-flow apparatus

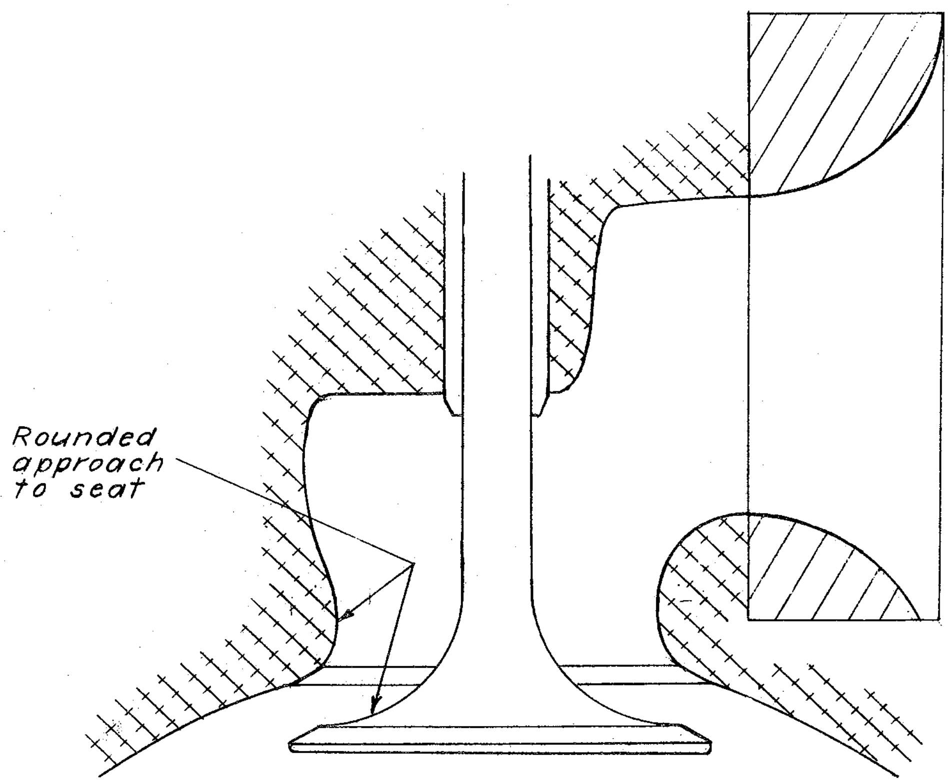


Figure 2. - Section through flow passage.

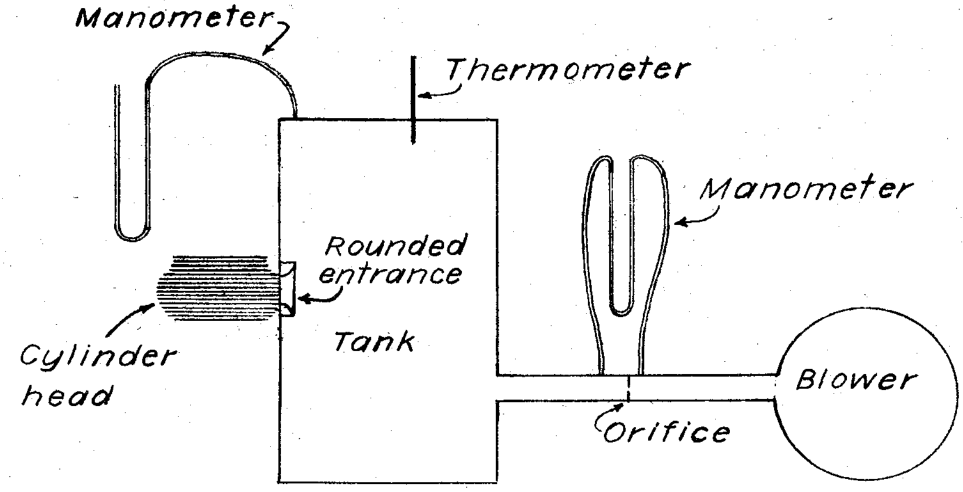


Figure 3.- Diagrammatic sketch of steady-flow apparatus.

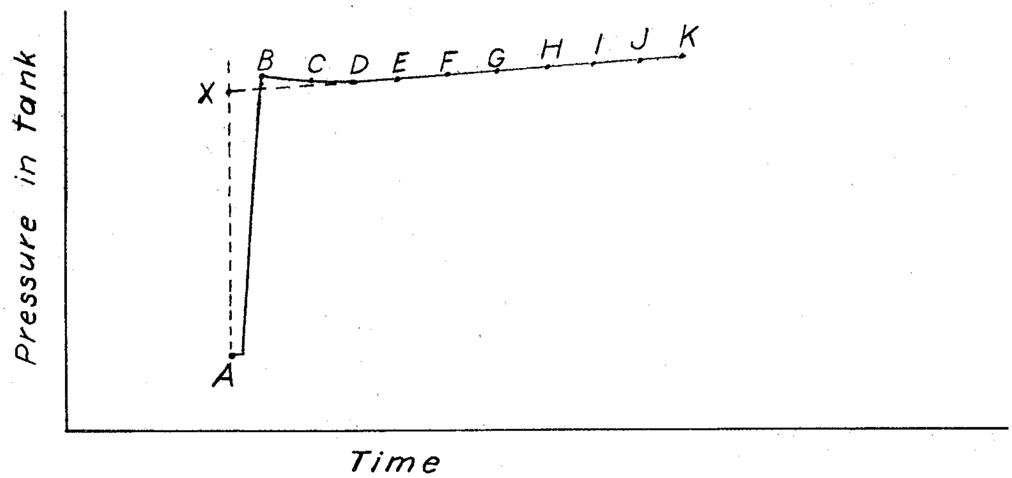


Figure 4.- Method of correcting for leakage and heating.

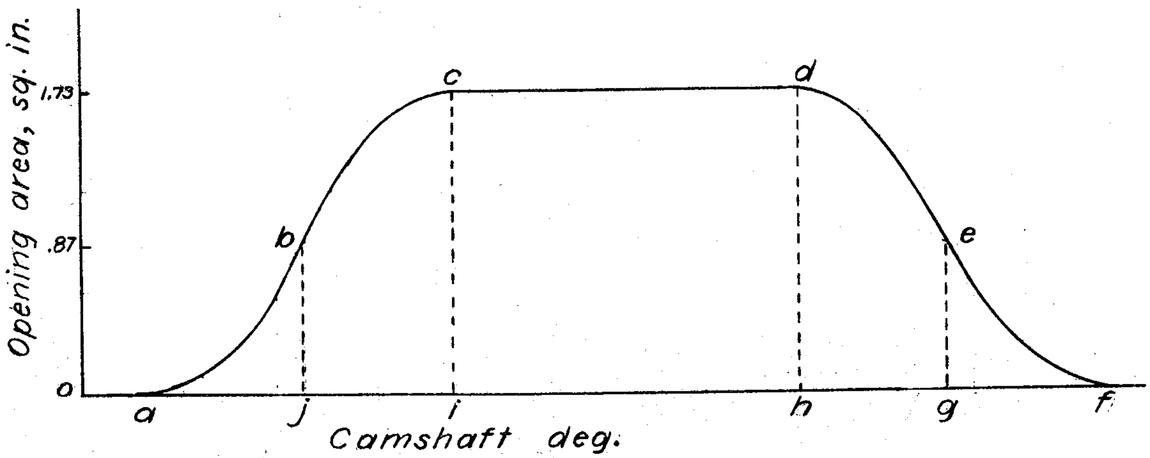


Figure 5.- Valve-lift diagram.

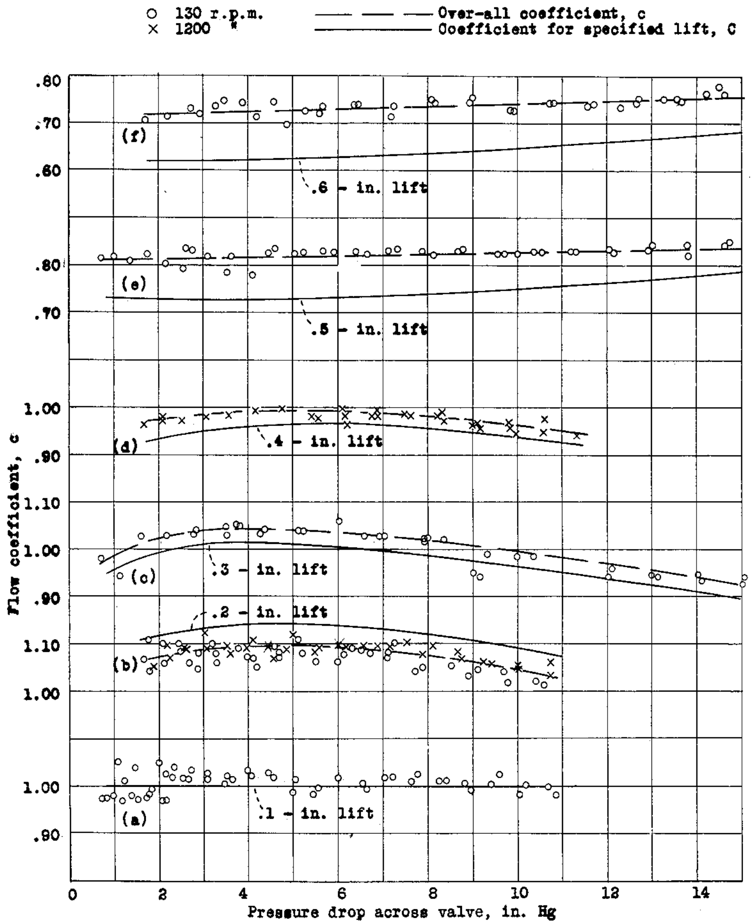


Figure 6.- Flow coefficients with intermittent flow.

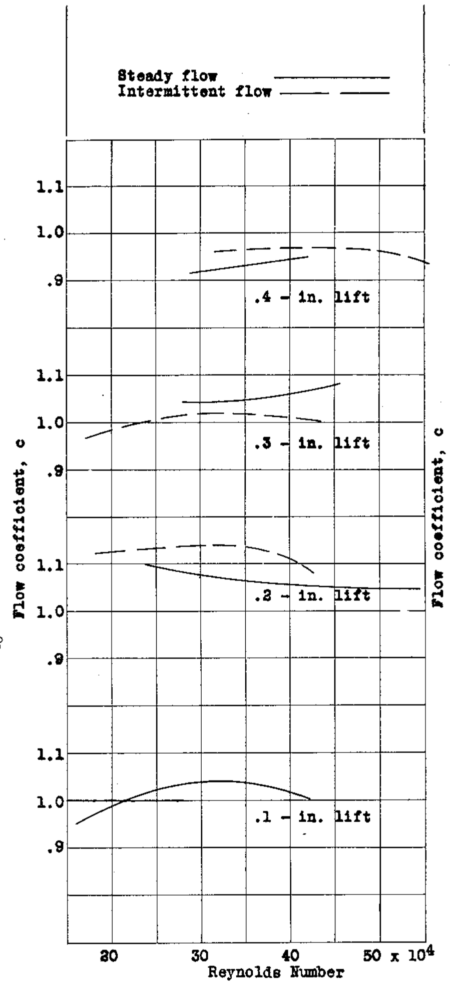


Figure 7.- Comparison of intermittent- and steady-flow coefficients.

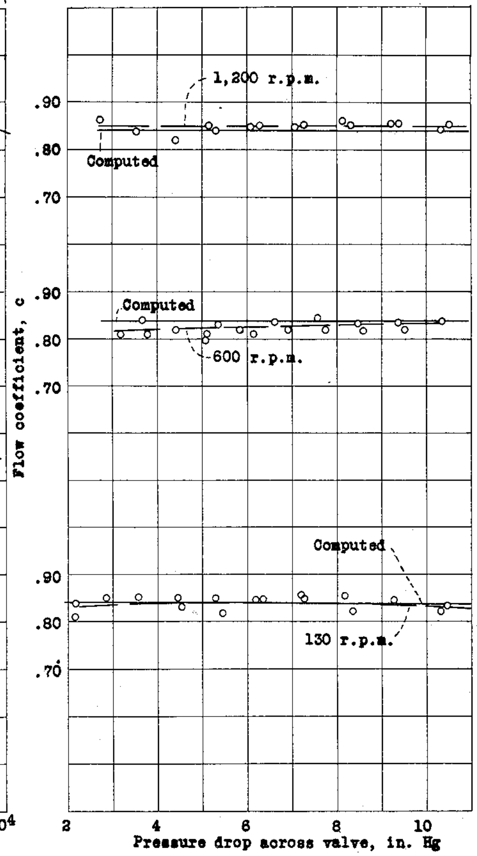


Figure 8.- Flow coefficients with regular engine cam.