CHARACTERISTICS AND TYPES OF HYDRODYNAMIC TORQUE CONVERTERS

Thesis for the Degree of M. S.

MICHIGAN STATE COLLEGE

Gordon Wang

1951

This is to certify that the

thesis entitled

Characteristics and Types of Hydro-
dynamic Torgue Converters.

presented by

Gordon Wang

has been accepted towards fulfillment
of the requirements for

M.S. degree in Mechanical Engineering

Geo Hobbs
Major professor

Date May 22, 1951

O-169

CHARACTERISTICS AND TYPES OF HYDRODYNAMIC TORQUE CONVERTERS

By

Gordon Wang

A THESIS

Submitted to the School of Graduate Studies of Michigan
State College of Agriculture and Applied Science
in partial fulfillment of the requirements
for the degree of

MASTER OF SCIENCE

Department of Mechanical Engineering

1951

THESIS

6/27/51

ACKNOWLEDGMENTS

The writer wishes to express his sincere appreciation to Professor G. W. Hobbs for his timely advice and guidance throughout this study.

Gratitude also is expressed to Professor J. N. Windburne for his critical reading of this manuscript.

CONTENTS

PAGE

PART 1

CHARACTERISTICS OF HYDRODYNAMIC TORQUE CONVERTERS

CHARACTERISTICS OF HYDRODYNAMIC TORQUE CONVERTERS

A hydrodynamic torque converter consists of essentially a pump wheel connected to the input shaft, a turbine wheel connected with the output shaft and a reaction member ring fixed or clutched to a stationary converter housing. Fig. 1 shows the arrangement of pump, turbine and reaction member for a typical torque converter. A fluid is used as a medium of torque and energy transfer which recirculates in a closed path formed by the converter elements.

CONSERVATION OF TORQUE. The velocity of any fluid particles circulating in the annular ring has two components: (1) Circumferential and (2) Annular. The circumferential component is perpendicular to the plane of the annular ring, and the annular component is in the plane of the annular ring which determines the rate at which the fluid recirculates.

The angular momentum of a fluid particle about an axis is equal to the product of its weight, its distance to the axis and its circumferential component. The angular momentum is designated to be positive when the circumferential velocity of the fluid particle is in the same direction of the motion of the pump, and it is negative when the circumferential velocity of the fluid is in

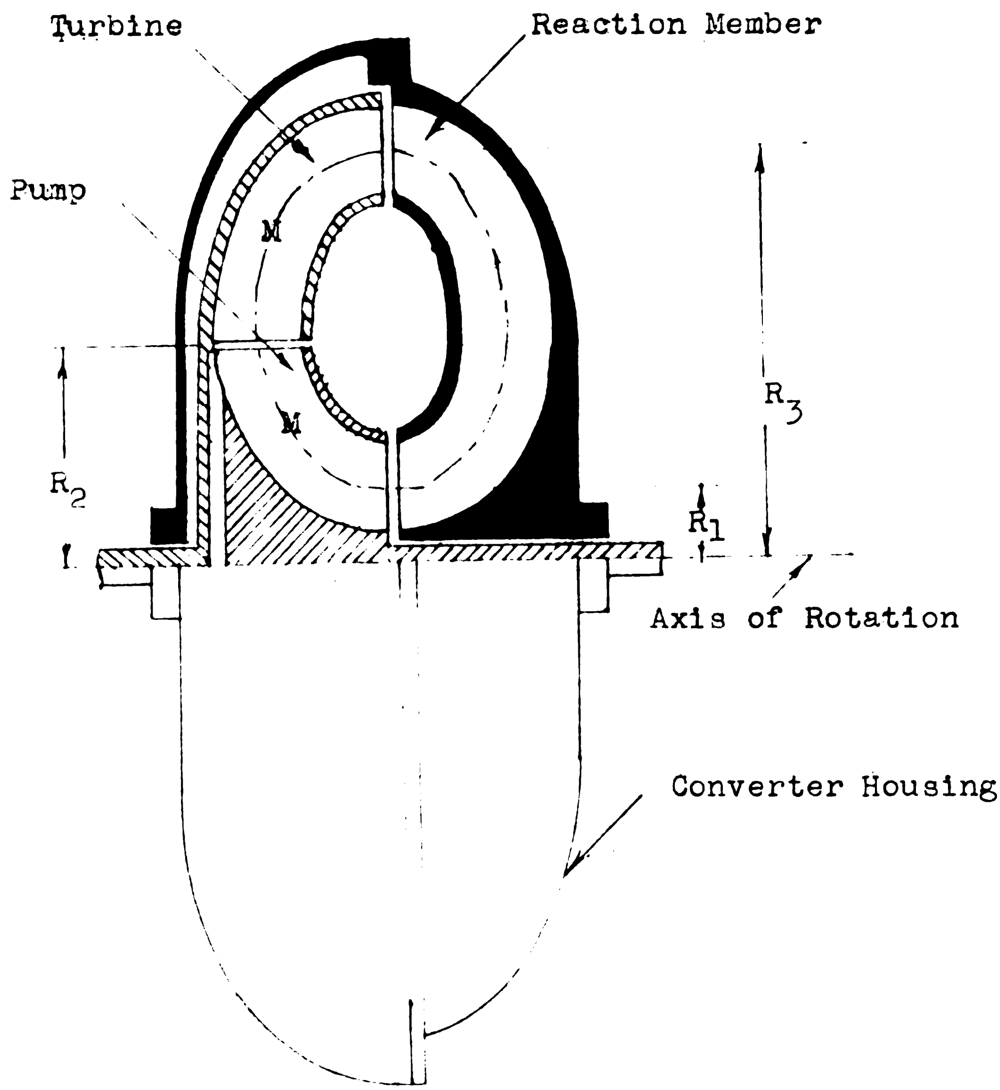


Fig. 1 Hydrodynamic Torque Converter

the opposite direction of the motion of the pump.

The mechanical torque exerted upon the wheel by the fluid is the difference of angular momentum of the fluid entering and leaving the wheel per unit time.

The angular momentum of the fluid leaving the pump is larger than that entering it due to the energy input at the pump, therefore the torque on the pump, which opposes the motion of the pump, is negative.

On the other hand, the turbine absorbs the kinetic energy of the fluid thereby the angular momentum of the fluid entering the turbine is larger than that leaving the turbine, consequently the torque exerted upon the turbine bears a positive sign which assists the motion of the turbine. In many cases, the angular momentum of the fluid leaving the turbine is very low and even changes its sign into negative: that means the fluid has reversed its motion in the circumferential direction. If there is no reaction member, as in the hydraulic fluid coupling, the pump has to raise the negative angular momentum to a certain positive value. It would require the entire engine torque to stop this backward rotation when it hits the entrance to the pump. And there would be nothing left to get the fluid reenergized into a positive rotation.

To correct this difficulty, the reaction member consisting of a set of stationary curved vanes is in-

stalled between the turbine exit and pump entrance. These vanes are properly shaped to receive the backward spinning fluid, bring it to a stop and then to direct it to a forward rotation. The angular momentum of the fluid leaving the reaction member is therefore larger than that entering it. A negative torque results which tends to rotate the reaction member opposite to the motion of the pump.

Since the total change in angular momentum about the axis of rotation is nil in a complete circulation, the resultant torque exerted upon the wheels therefore is also nil. The torques on the pump, turbine and reaction member are T_p, T_t and T_r respectively. Then

$$- T_p + T_t - T_r = 0$$

$$\text{or, } T_t = T_p + T_r \tag{1}$$

Thus the turbine torque is equal to the engine or pump torque augmented by the torque reaction of the reaction member.

<u>VELOCITY DIAGRAMS FOR THE PUMP</u>. From the foregoing discussion, only the entrance and exit of the wheels are considered important. Therefore, a knowledge of the velocity relations at the entrance and exit of the wheels is necessary in order to analyze the torque converter. The

diagrams showing the velocity relations at the entrance and exit of the wheels are called velocity diagrams.

(Refer to Fig. 1.) A central portion of the annular ring M is chosen as the characteristic section of the torque converter. It is called the mean stream layer.

Before discussing the velocity diagrams for the pump, the different parameters necessary to construct the velocity diagrams are given the following notation:

Q, flow rate of the fluid, cu. ft. per sec.

D, density of the fluid, lb. per cu. ft.

W_p, speed of the pump, radians per sec.

R_1, mean radius between the pump entrance and reaction member exit, ft.

R_2, mean radius between the pump exit and turbine entrance, ft.

A_1, entrance area of the pump, or exit areas of the reaction member, sq. ft.

A_2, exit area of the pump, or entrance area of the turbine, sq. ft.

C_1, absolute velocity of the fluid between the reaction member and the pump, ft. per sec.

C_2, absolute velocity of the fluid between the pump the turbine, ft. per sec.

V_{p1}, relative velocity of the fluid in regard to the pump at entrance, ft. per sec.

V_{p2}, relative velocity of the fluid in regard to the pump at exit, ft. per sec.

B_{p1}, blade angle of the pump at entrance, deg.

B_{p2}, blade angle of the pump at exit, deg.

V_{shp}, shock velocity at the pump, ft. per sec.

V_{whp}, relative velocity of the fluid in regard to the pump at entrance before the guiding effect, ft. per sec.

g. the gravity constant, 32.2 ft. per sec. per sec.

The velocity diagrams at the pump exit will be discussed first. (Refer to Fig. 2.) The fluid leaves the pump with a relative velocity V_{p2} at an angle B_{p2} with the motion of the pump. And the velocity of the pump at that point is W_pR_2. The resultant force C_2 is the velocity of the fluid at the pump exit. The annular component of C_2 is Q / A_2.

There is a slight difference at the pump entrance. (Refer to Fig. 3.) The fluid leaves the reaction member at an absolute velocity C_1. The velocity of the pump at the entrance is W_pR_1, therefore, the relative velocity of the fluid before meeting the blade is V_{whp}. However, upon entering the pump the fluid changes its relative velocity from V_{whp} to V_{p1} due to the guiding effect of the blade. B_{p1} is the blade angle. The vectorial difference of V_{whp} and V_{p1} is V_{shp}, which is called shock

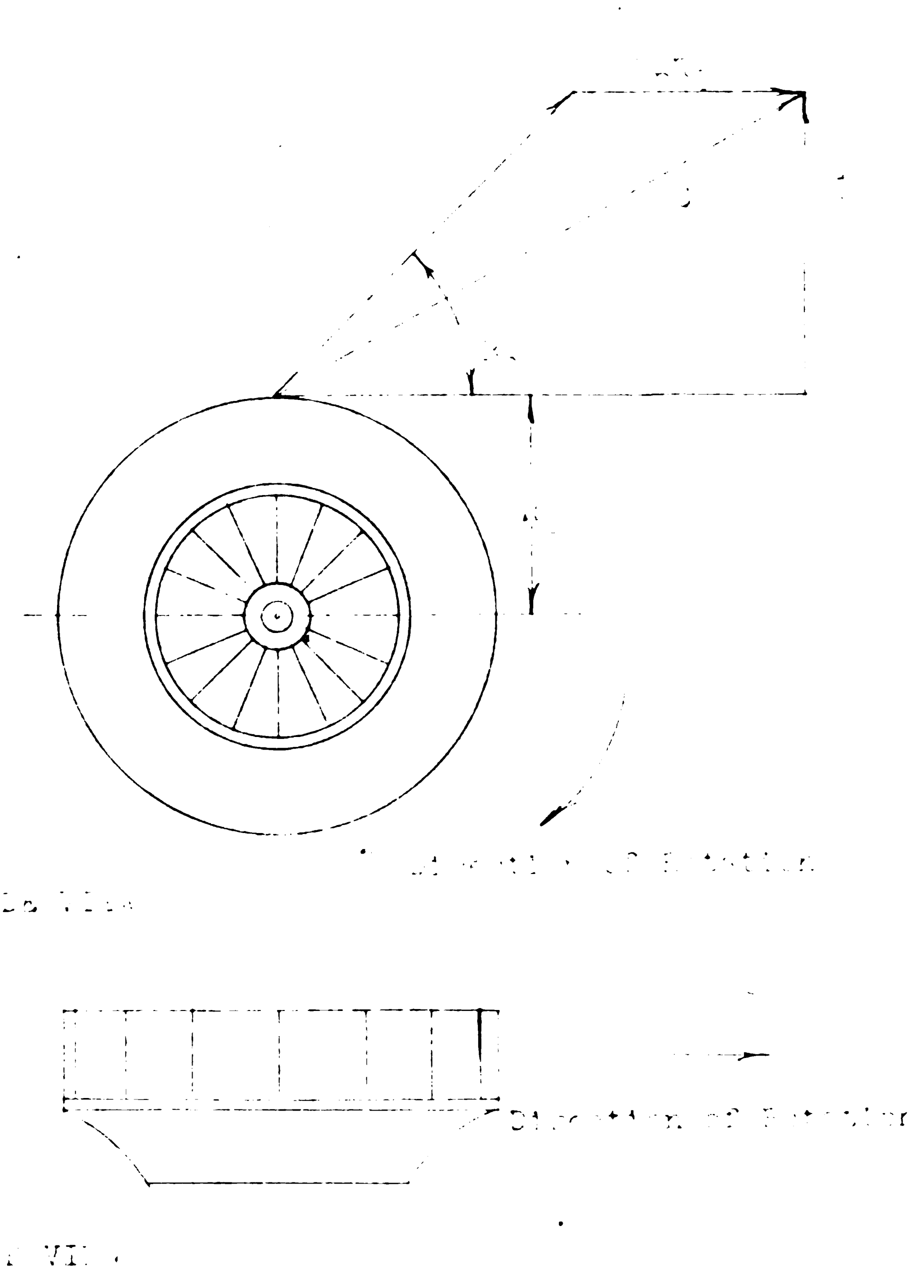


Fig. 2 Velocity Diagram at the Pump Exit.

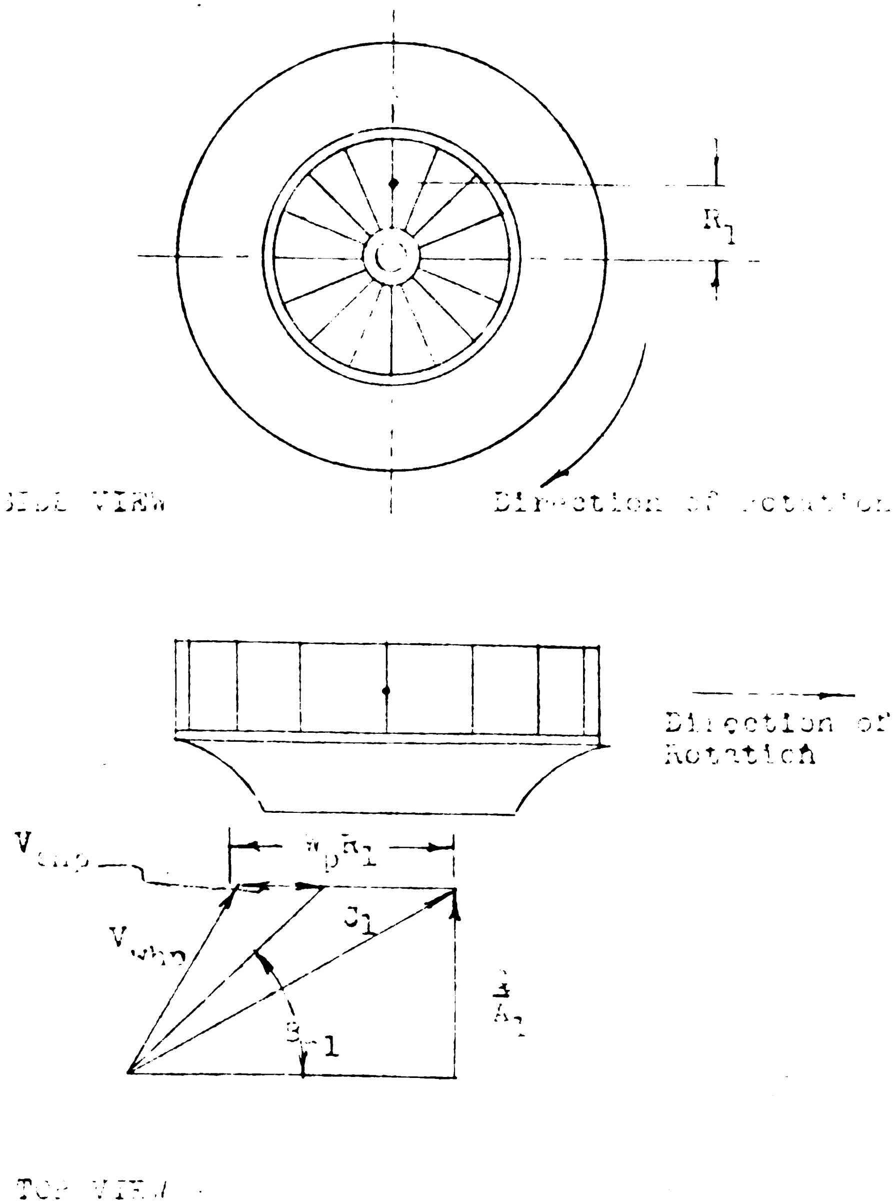


Fig. [illegible] Velocity Diagram at the Vane Entrance.

velocity. Except for one set of conditions, the blade entrance is not built to receive the fluid smoothly, and the shock velocity exists.

VELOCITY DIAGRAMS FOR THE TURBINE. Figs. 4 and 5 show the velocity diagrams at the turbine entrance and exit respectively. The following notation is given:

W_t, speed of the turbine, radians per sec.

R_3, mean radius between the turbine exit and reaction member entrance, ft.

A_3, exit area of the turbine of entrance area of the reaction member, sq. ft.

V_{t2}, relative velocity of the fluid in regard to the turbine at entrance, ft. per sec.

V_{t3}, relative velocity of the fluid in regard to the turbine at exit, ft. per sec.

C_3, absolute velocity of the fluid between the turbine and the reaction member, ft. per sec.

B_{t2}, blade angle of the turbine at entrance, deg.

B_{t3}, blade angle of the turbine at exit, deg.

(Refer to Fig. 4.) The fluid leaves the pump with an absolute velocity C_2, The velocity of the turbine at that point is W_tR_2, therefore the relative velocity of the fluid in regard to the turbine is V_{wht}. But the guiding effect of the blade changes V_{wht} to V_{p2} at an angle B_{t2}. The shock velocity is V_{sht}.

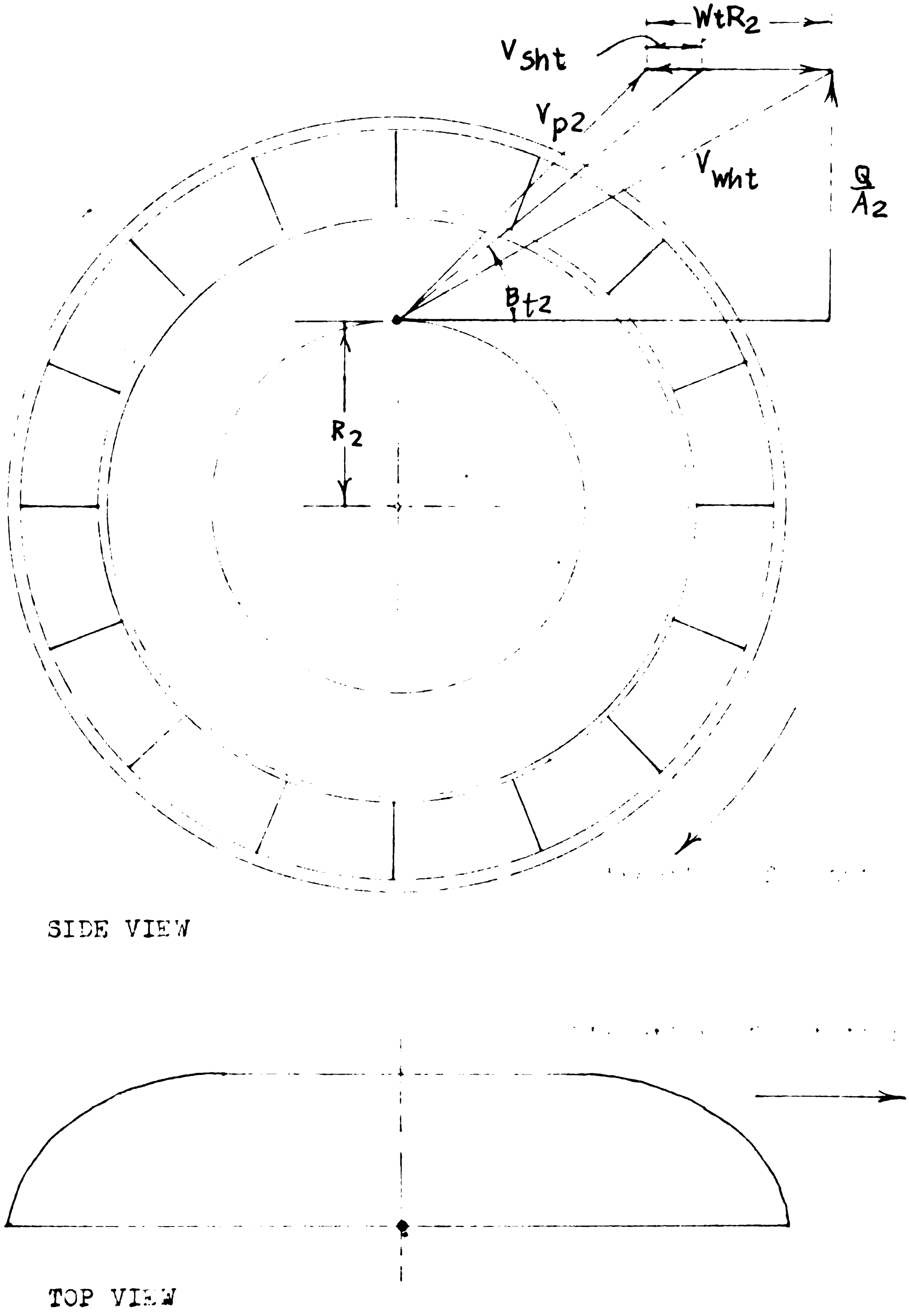


Fig. 4 Velocity Diagram at the Turbine Entrance

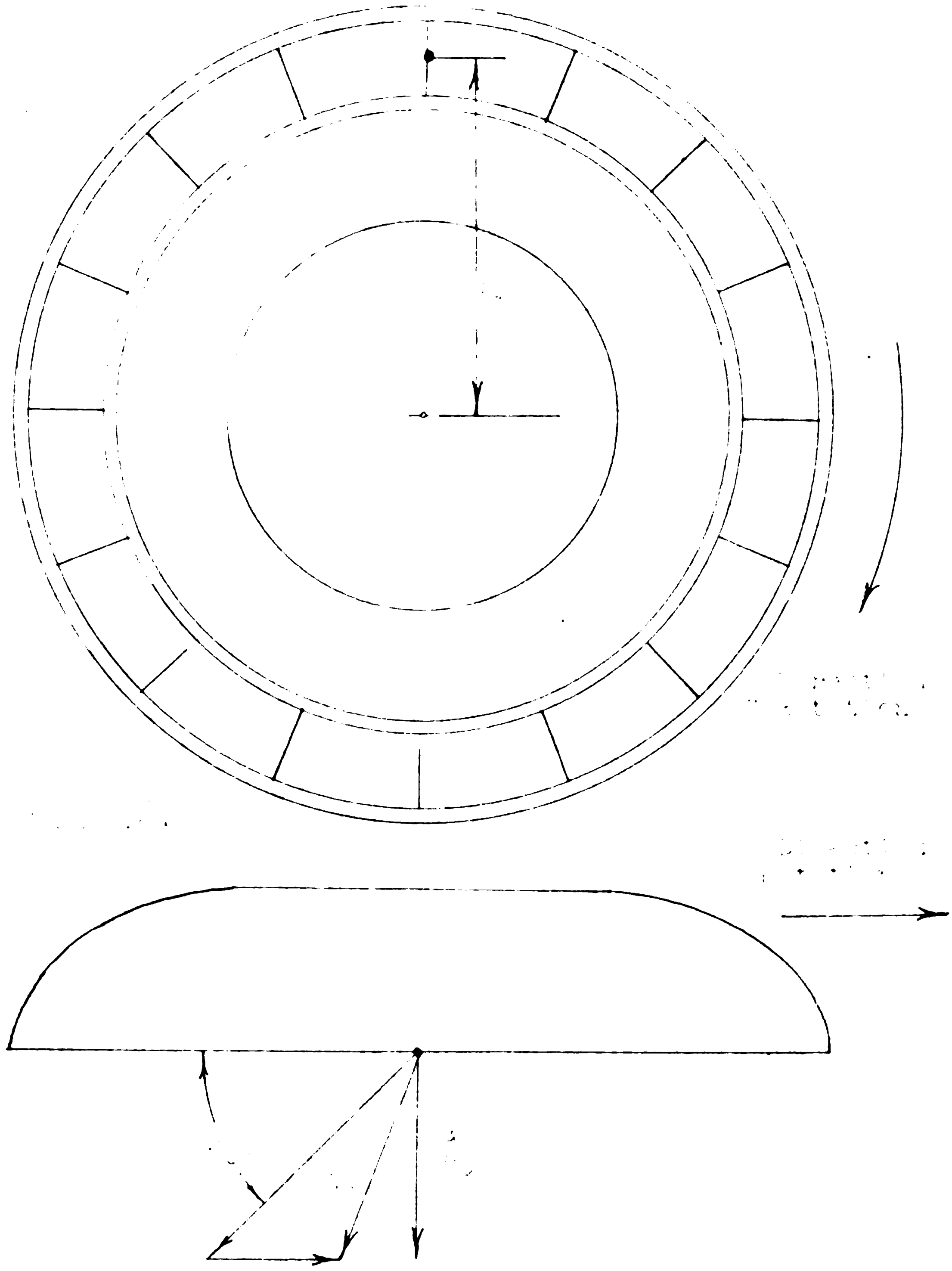


Fig. [illegible] Turbine Exit.

At the exit of the turbine (refer to Fig. 5.) the fluid leaves the turbine with a relative velocity V_{t3}. The sum of V_{t3} and the velocity of the turbine at that point W_tR_3 is C_3, the absolute velocity of the fluid. That the circumferential component of the velocity of the fluid changes direction after passing through the turbine is clearly shown.

VELOCITY DIAGRAMS FOR THE REACTION MEMBER. (Refer to Fig. 6.) For the stationary reaction member the fluid enters the wheel with an absolute velocity C_3. Since the wheel is stationary, the absolute velocity C_3 equals to V_{whr} the relative velocity of the fluid. The guiding effect of the blade changes V_{whr} to V_{shr}, the shock velocity.

(Refer to Fig. 7.) The fluid leaves the wheel with a relatively velocity V_{r1} at B_{r1}. Since the reaction is stationary, the relative velocity V_{r1} equals to C_3, the absolute velocity of the fluid between the reaction member and the pump.

The reversal of direction of the circumferential component of the velocity of the fluid is shown.

TORQUE AND ENERGY TRANSFER AT THE PUMP. Considering the interaction of the fluid with the pump, (refer to Fig. 2.) the circumferential velocity of the fluid at the pump exit is the sum of the velocity of the pump at that point $W_pR_2$plus the circumferential component of the relative

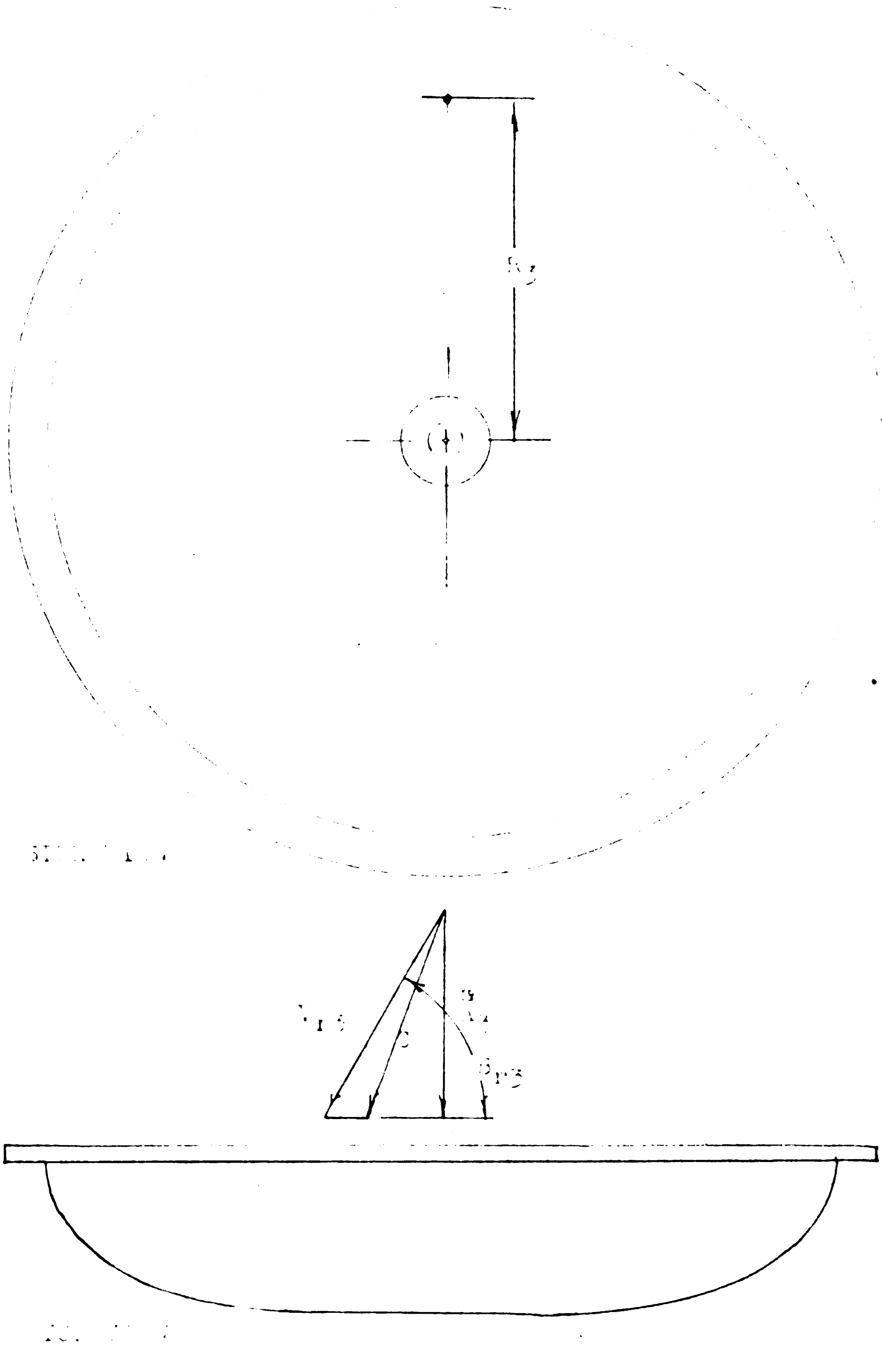


Fig. 5 Velocity Diagram at the Reaction Member Entrance.

fluid velocity $\frac{Q}{A_2}\cot B_{p2}$, since the annular velocity of the fluid relative to the pump is Q / A_2.

The angular momentum of the circulating fluid across the exit section of the pump per unit time is

$$\frac{DQ}{g}\left(W_p R_2 + \frac{Q}{A_2}\cot B_{p2} \right) R_2 \text{ ft-lb. per sec.}$$

The angular momentum of the fluid per unit time entering the pump is likewise the angular momentum of the fluid per unit time across the exit section of the stationary reaction member, i. e.,

$$\frac{DQ}{g}\left(\frac{Q}{A_1}\cot B_{r1} \right) R_1 \text{ ft-lb. per sec.}$$

The torque exerted on the pump is the difference of the angular momentum of the fluid leaving and entering it, therefore

$$T = \frac{DQ}{g}\left(W_p R_2^2 + Q\,\frac{\cot B_{p2}}{A_2} R_2 - Q\,\frac{\cot B_{r1}}{A_1} \right) \text{ ft-lb} \qquad (2)$$

Or,

$$T = K_1 W_p Q + K_2 Q^2 \text{ ft-lb} \qquad (2)$$

where, $K_1 = \frac{D}{g} R_2^2$

$$K_2 = \frac{D}{g}\left(\frac{\cot B_{p2}}{A_2} R - \frac{\cot B_{r1}}{A_1} \right)$$

The power input at the pump is

$$P_p = T_pW_p = (K_1W_pQ + K_2Q^2) W_p \text{ ft-lb per sec.} \tag{3}$$

At constant flow rate Q, the torque on the pump increases directly with the speed, while the power input varies as a parabolic function of the pump speed. While at the constant pump speed W_p, both the pump torque and power input varies as a parabolic function of the flow rate.

<u>TORQUE AND ENERGY TRANSFER AT THE TURBINE</u>. The driving torque on the turbine wheel equals the difference between the entrance and exit angular momentum of the fluid per unit time. The entrance fluid angular momentum of the fluid per unit time to the turbine is the exit fluid angular momentum per unit time from the pump, namely

$$\frac{DQ}{g} (W_pR_2 + \frac{Q}{A_2} \cot B_{p2}) R_2 \text{ ft-lb per sec.}$$

The exit angular momentum of the fluid per unit time from the turbine is

$$\frac{DQ}{g} (W_tR_3 + \frac{Q}{A_3} \cot B_{t3}) R_3 \text{ ft-lb per sec.}$$

The driving torque exerted by the fluid on the turbine is therefore,

$$T_t = \frac{DQ}{g} \left(W_t R_2^2 + \frac{QR_2 \cot B_{p2}}{A_2} - \frac{QR_3 \cot B_{t3}}{A_3} - R_3^2 W_t \right) \text{ ft-lb.} \quad (4)$$

Or,

$$T_t = K_1 Q W_p + K_3 Q^2 - K_4 Q W_s \quad \text{ft-lb} \quad (4)'$$

where,

$$K_1 = \frac{D}{g} R_2^2$$

$$K_3 = \frac{D}{g} \left(\frac{R_2 \cot B_{p2}}{A_2} - \frac{R_3 \cot B_{t3}}{A_3} \right)$$

$$K_4 = \frac{D}{g} R_3^2$$

The power transfer between the fluid and the turbine is

$$P_t = T_s W_s = (K_1 Q W_p + K_3 Q^2 - K_4 Q W_t) W_t \text{ ft-lb per sec.} \quad (5)$$

If the pump speed in constant, at any given turbine speed W_t, both the turbine torque and power output varies as a parabolic function of the flow rate. For any given flow rate Q, the turbine torque decreases with increase of turbine speed, and the power output varies as a parabolic function of W_t.

To find the relation of the power output P_t and turbine speed W_t for constant pump speed W_p and constant flow rate Q, equations (4)' and (5) can be rewritten:

$$T_t = M - NW_t \quad \text{ft-lb} \tag{6}$$

$$P_t = (M - NW_t) W_t \text{ ft-lb} \tag{7}$$

where,

$$M = K_1QW_p + K_3Q^2$$

$$N = K_4Q$$

The maximum turbine torque is M when W_t is zero, or, in other works, when the turbine stalls.

To find the turbine speed at maximum output. Let the first derivative of P_t with respect to W_t to zero,

$$\frac{d}{dW_t} P_t = M - 2NW_t = 0$$

so, $$W_t = \frac{M}{2N}$$

Substituting this value of W_t into equation (6) gives T_t equals to M / 2. This means that the stalling torque is just twice the torque developed at the turbine speed when maximum power output occurs.

ENERGY LOSSES ACROSS THE PUMP. The fluid flow through various passages is subjected to different losses. These losses are divided into shock and friction losses. Shock losses are suffered from a wrong angle entrance into the blades and roughly varies with the square of the shock velocity. While the friction losses are suffered through the path of the converter assuming that the fluid enters the blade shock free, and are proportional to the square of the exit relative velocity of the fluid.

The energy losses across the pump will be discussed first. Refer to Fig. 3 and 7, the shock velocity V_{shp} is equal to

$$\frac{Q}{A_1}\cot B_{r1} - \frac{Q}{A_1}\cot B_{p1} - W_p R_1 \text{ ft-lb per sec.}$$

Therefore, the shock loss at the pump

$$L_{shp} = \frac{DQ}{2g}\left(\frac{Q}{A_1}\cot B_{r1} - \frac{Q}{A_1}\cot B_{p1} - W_p R_1\right)^2 \text{ ft-lb per sec.} \qquad (8)$$

In the equation (8), the shock coefficient is unity.

The exit relative velocity at the pump exit of the fluid is $\frac{Q}{A_2}\csc B_{p2}$ (See Fig. 2), the friction loss

$$L_{fp} = \frac{fDQ^3}{2g}\left(\frac{\csc B_{p2}}{A_2}\right)^2 \text{ ft-lb per sec} \qquad (9)$$

where, f = .2 roughly

ENERGY LOSSES ACROSS THE TURBINE. For the shock loss, refer to Fig. 2 and Fig. 4, the shock velocity at the

turbine entrance V_{sht} is

$$W_pR_2 + \frac{Q}{A_2}\cot B_{p2} - W_tR_2 - \frac{Q}{A_2}\cot B_{t2} \text{ ft. per sec.}$$

The shock loss,

$$L_{sht} = \frac{DQ}{2g}\left(W_pR_2 + \frac{Q}{A}\cot B_{p2} - W_tR_2 - \frac{Q}{A_2}\cot B_{t2} \right)^2$$

$$\text{ft-lb per sec.} \quad (10)$$

The relative exit velocity of the fluid is $\frac{Q}{A_2}\csc B_{t3}$, (See Fig. 5). The friction loss

$$L_{ft} = \frac{fDQ^3}{2g}\left(\frac{\csc B_{t3}}{A_3} \right)^2 \text{ ft-lb per sec.} \quad (11)$$

ENERGY LOSSES ACROSS THE REACTION MEMBER. Refer to Fig. 5 and 6, the shock velocity at the reaction member entrance V_{shr} is

$$W_tR_3 + \frac{Q}{A_3}\cot B_{t3} - \frac{Q}{A_3}\cot B_{r3} \text{ ft per sec.}$$

Therefore, the shock loss

$$L_{shr} - \frac{DQ}{2g}\left(W_tR_3 + \frac{Q}{A_3}\cot B_{t3} - \frac{Q}{A_3}\cot B_{r3} \right)^2$$

$$\text{ft-lb per sec.} \quad (12)$$

And from Fig. 7, the relative velocity of the fluid at the reaction member exit is $\frac{Q}{A_3}\csc B_{r1}$. The friction loss at the reaction member is

$$L_{fr} = \frac{fDQ^3}{2g} \left(\frac{\csc B_{r1}}{A_1} \right)^2 \text{ ft-lb per sec.} \qquad (13)$$

ENERGY EQUATION. According to the law of conservation of energy, the power input at the pump is equal to the sum of the power output at the turbine and the losses across different wheels. Therefore,

$$P_p = P_t + L_{shp} + L_{fp} + L_{sht} + L_{ft} + L_{shr} + L_{fr}$$

$$\text{ft-lb per sec.} \qquad (14)$$

Substituting P_p, P_t and different losses from the foregoing sections and simplifying gives:

$$\frac{W_p^2}{2g} \left(R_2^2 - R_1^2 \right) + \frac{W_t^2}{2g} \left(R_3^2 - R_2^2 \right)$$

$$= \frac{Q}{g} \quad W_p R_1 \cot B_{p1} / A_1 - W_s R_3 \cot B_{r3} / A_3$$

$$- \left(W_p - W_s \right) R_2 \cot B_{t2} / A_2$$

$$+ \frac{Q^2}{2g} \quad \frac{f \csc B_{r1}^2 + (\cot B_{r1} - \cot B_{p1})^2}{\overset{2}{A_1}}$$

$$\frac{f \csc B_{r1}^2 + (\cot B_{r1} - \cot B_{p1})^2}{\overset{2}{A_2}}$$

$$\frac{f \csc B_{t3}^2 + (\cot B_{t3} - \cot B_{r3})^2}{\overset{2}{A_3}} \qquad (15)$$

CHARACTERISTICS OF TORQUE CONVERTER. Torque ratio is defined as the ratio of the turbine torque to pump torque, or

$$\text{Torque ratio} = T_t \ / \ T_p = \frac{K_1 Q W_p + K_3 Q^2 - K_4 Q W_t}{K_1 W_p - K_2 Q^2} \tag{16}$$

where, $K_1 = \frac{D R_2^2}{g}$

$$K_2 = \frac{D}{g} \left(\frac{\cot B_{p2}}{A_2} R_2 - \frac{\cot B_{r1}}{A_1} \right)$$

$$K_3 = \frac{D}{g} \left(\frac{R_2 \cot B_{p2}}{A_2} - \frac{R_3 \cot B_{t3}}{A_3} \right)$$

$$K_4 = \frac{D}{g} R_3^2$$

The overall efficiency of the torque converter is given by the equation:

$$e = P_t \ / \ P_p = \frac{(K_1 Q W_p + K_3 Q^2 - K_4 Q W_t) W_t}{(K_1 W_p + K_2 Q^2) W_p} \tag{17}$$

Equation (15) can also be expressed as an implicit function of variable W_p, W_t, Q (and the blade angles or radii if they can by made to vary) :

$$F (W_p, W_t, Q) = 0 \tag{18}$$

Only two of the variables W_p, W_t and Q are considered to be independent. If W_p and W_t are chosen as independent variables, Q can be calculated by equation (18). By substituting the values of W_p, W_t and Q into the torque and energy or power equations derived in the previous sections, the characteristics of a torque converter of certain design can be found.

If the pump speed W_p is kept constant, then the circulation rate Q varies with the turbine speed W_t according to the equation (18) or (15). The turbine torque T_t modified from a linear relation with the turbine speed W_t. And the pump torque T_p cannot be constant throughout the range of the turbine speeds, and must vary with the turbine speed.

If the pump torque T_p is kept constant, then

$$T_p = K_1 W_p Q + K_2 Q^2 = \text{constant}$$

or,

$$Q = \frac{- K_1 W_p + (K_1^2 W_p^2 + 4K_2 T_p)^{1/2}}{2K_2}$$

On substituting in the equation (18) or (15), W_p is directly related with W_t. At constant pump torque, the pump speed increases nearly linearly with the turbine speed.

However, if the pump speed W_p is constant, the characteristics of the torque converter are not greatly modified by assuming a constant flow rate Q for a wide range of turbine speed W_t. Since the pump speed W_p and flow rate Q are assumed to be constant, this implies that the power input P_p and pump torque T_p are constant. The efficiency of the torque converter therefore will be

$$e = \frac{P_t}{P_p} = \frac{(M - NW_t)\, W_t}{T_p W_p} \tag{19}$$

where, $M = K_1 Q W_p + K_3 Q^2$

$N = K_4 Q$

The efficiency, assuming a constant flow rate, as shown in equation (19) reaches a maximum when W_t equals to $M / 2N$ or the rated speed, and falls to zero when W_t equals to zero and M / N.

One difficulty with a direct application of the torque converter is the rapid drop of efficiency with increasing overspeed relative to its rated speed.

Fig. 8 shows the torque and efficiency curves at constant pump speed W_p for a wide range of turbine speed. The flow rate Q increases as the turbine speed either decreases below or increases above the rated speed, thereby the corresponding output torque and efficiency are higher than that of assuming a constant flow rate.

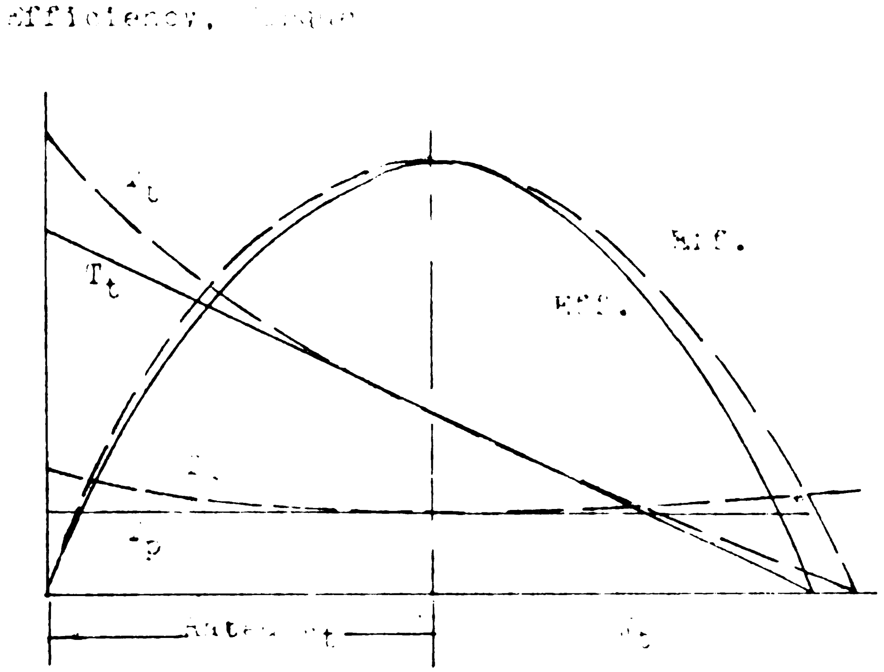


Fig. 3 [illegible] Efficiency [illegible]
[illegible]

[illegible]

[illegible] Flow Rate

However, the increase of torque ratio and efficiency due to the increase of flow rate is checked by the increase of friction loss and shock loss at both the lower and higher speeds. Therefore, a thorough understanding of the nature of these losses is indispensable in order to decrease these losses and thereby increase the performance of the torque converter.

DESIGN FACTORS FOR REDUCING FRICTION LOSSES. Equations (9), (10) and (11) express the friction losses at different wheels as follows:

$$L_{fp} = \frac{fDQ^3}{2g} \left(\frac{\csc B_{p2}}{A_2} \right)^2$$

$$L_{ft} = \frac{fDQ^3}{2g} \left(\frac{\csc B_{t3}}{A_3} \right)^2$$

$$L_{fr} = \frac{fDQ^3}{2g} \left(\frac{\csc B_{r1}}{A_1} \right)^2$$

The friction coefficient f depends on the blade and shell wall friction, curvature of the blade, and the viscosity of the circulating fluid. Under most favorable conditions: smooth surface finish on the blades and shell wall, uniform rate of change of curvature of the blades, and minimum viscosity of the fluid, the friction coefficient can be made as low as 0.08. In most cases, the friction coefficient lies between 0.15 to as high as 0.25.

The relation of the exit blade angle to the friction losses is clearly shown in the equations. In order to reduce the friction losses, the angle should be small. The annular cross-sectional area also affects the friction losses, the larger the area, the smaller the losses.

The most important single factor affecting the amount of friction losses is the rate of circulation of the fluid Q. When the turbine speed is low, the flow rate is extremely high. Therefore the highest friction losses are suffered when the turbine speed is low, especially when the turbine stalls.

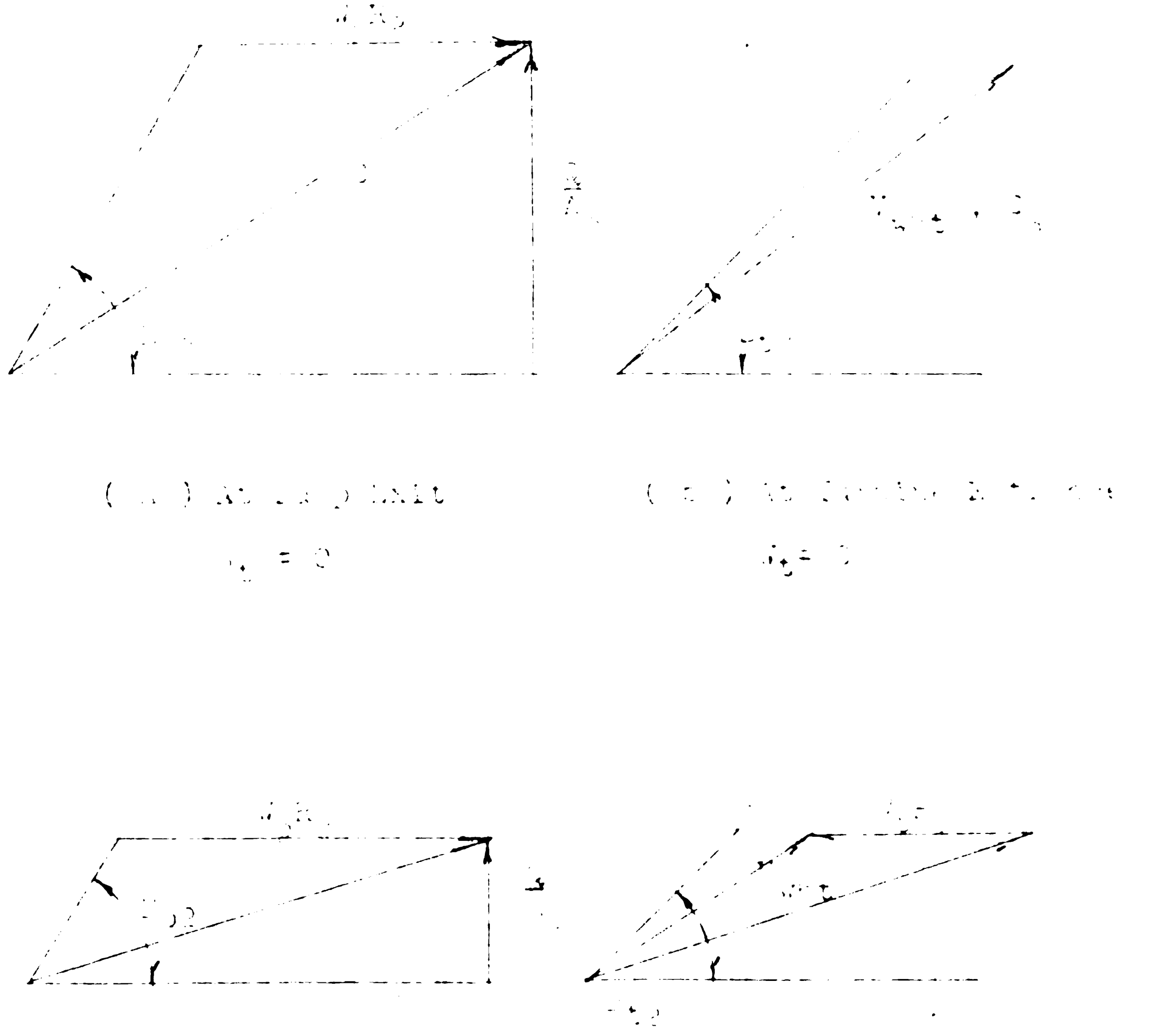
The fluid density also affects the friction losses as indicated by the equations. However, its effect is so small, that the fluid with maximum density and minimum viscosity is used in order to carry more energy per unit volume of the fluid. In other words, the converter can be made smaller with heavier fluid.

DESIGN FACTORS FOR REDUCING SHOCK LOSSES. Shock losses are responsible for the rapid decrease of the converter efficiency at low and high turbine speeds. In turbines with the wheels arranged in the pump, turbine, reaction member order, the highest shock loss is at the reaction member entrance, less at the pump entrance and negligible at the turbine entrance.

Assuming the pump torque being constant, the flow rate Q is highest when the turbine speed is zero. As the turbine speeds up, the flow rate decreases until it becomes minimum at the rated turbine speed, and the flow rate increases as the turbine overspeeds. The pump speed, however, increases linearly at a slow rate with the increase of the turbine speed. But when the turbine speed is over the rated value, the pump speed increases rapidly.

The nature of the entrance relative velocities V_{whp}, V_{wht} and V_{whr} relative to the turbine speed will be discussed separately.

The entrance condition at the turbine will be discussed first. Fig. 9a is the velocity diagram at the pump exit when the turbine stalls, and Fig. 9b is the velocity diagram at the turbine entrance. As the turbine speed increases, the velocity diagram at the pump exit is changed, as shown in Fig. 9c. The flow rate Q reduced due to the increase of turbine speed, and the fluid velocity C_2 rotates clockwise as compared to C_2 in Fig. 9b. However, at the turbine entrance, the effect of change of direction of the fluid velocity is compensated for by the motion of the turbine which tends to make the entrance relative velocity V_{wht} in the same direction as V_{wht} in Fig. 9b. Consequently, the shock losses at the turbine is small and can be neglected.



(a) At Pump Exit

$\omega_t = 0$

(b) At Turbine Entrance

$\omega_t = 0$

(c) At Pump Exit

With ω_t

(d) At Turbine Entrance

With ω_t

Fig. [illegible] Velocity Diagrams at the Pump Exit and the Turbine Entrance

But the situations at the pump and reaction member entrance are different. In the following section, the shock loss at the pump entrance and the method of reducing it will be discussed.

<u>DESIGN FEATURE FOR REDUCING SHOCK LOSS AT THE PUMP</u>. Fig. 10a is the velocity diagram at the reaction member exit, and Fig. 10b is the velocity diagram at the pump entrance when the turbine stalls. As the turbine speeds up, the velocity diagrams at the reaction member exit and pump entrance are shown in Fig. 10c and Fig. 10d respectively. The smaller flow rate and higher pump speed cause the relative fluid entrance velocity V_{whp} shift to the left side as compared with V_{whp} in Fig. 10b. Fig. 10e shows the change of direction of V_{whp} the relative entrance velocity at low and high turbine speeds. At low turbine speed, especially when the turbine stalls, the value of rate of fluid flow is high, and the shock loss is at its maximum.

In order to reduce the shock loss at the pump entrance at low turbine speed, the pump is divided into two pieces. Refer to Fig. 11a, the secondary pump S is attached to the primary pump P by roller clutches. S is permitted to turn faster than P in the direction of motion of P, but never lags behind due to the one way action of the roller clutches.

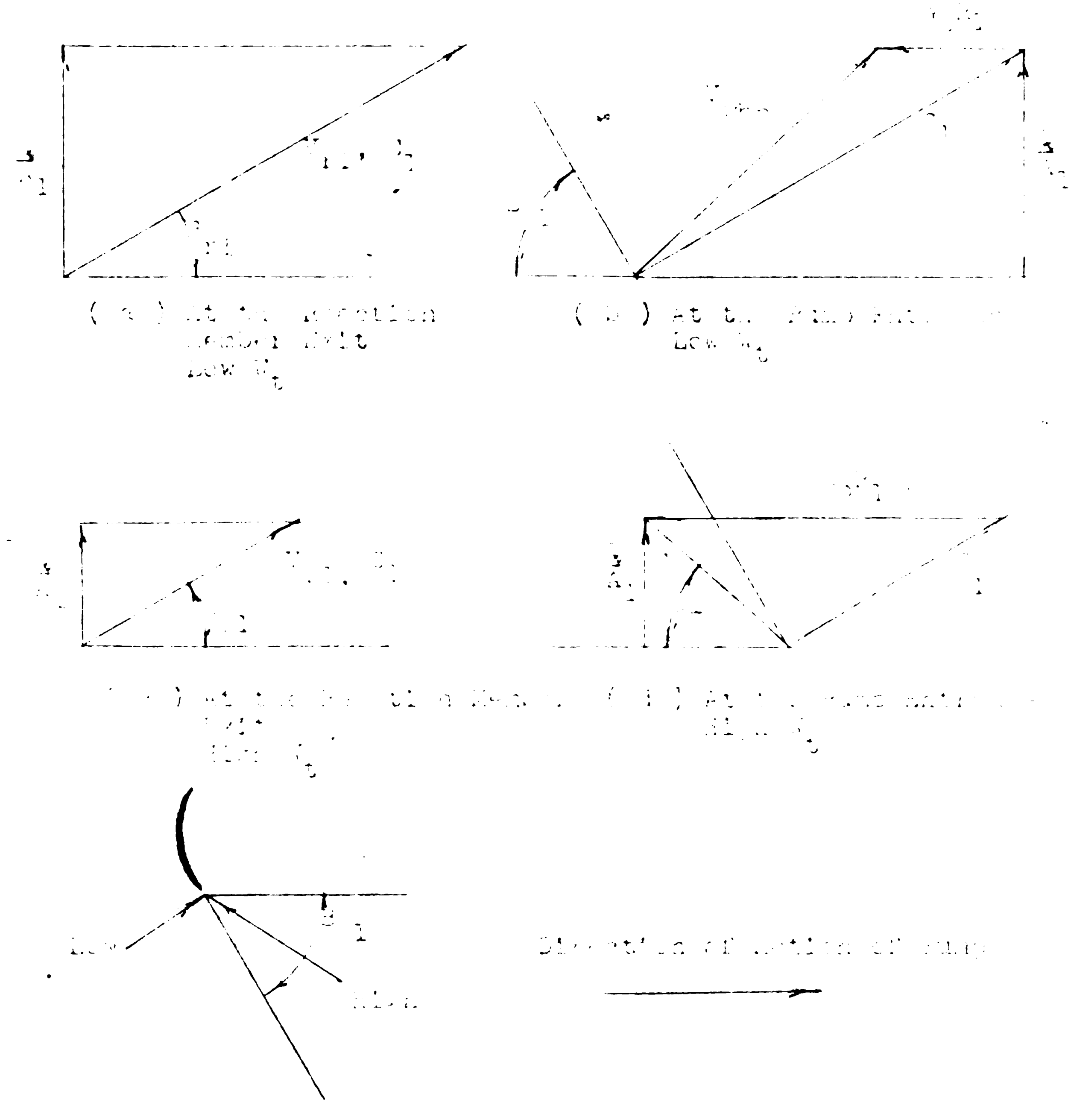


Fig. 10 Velocity Diagrams at the Reaction Member Exit and Pump Entrance

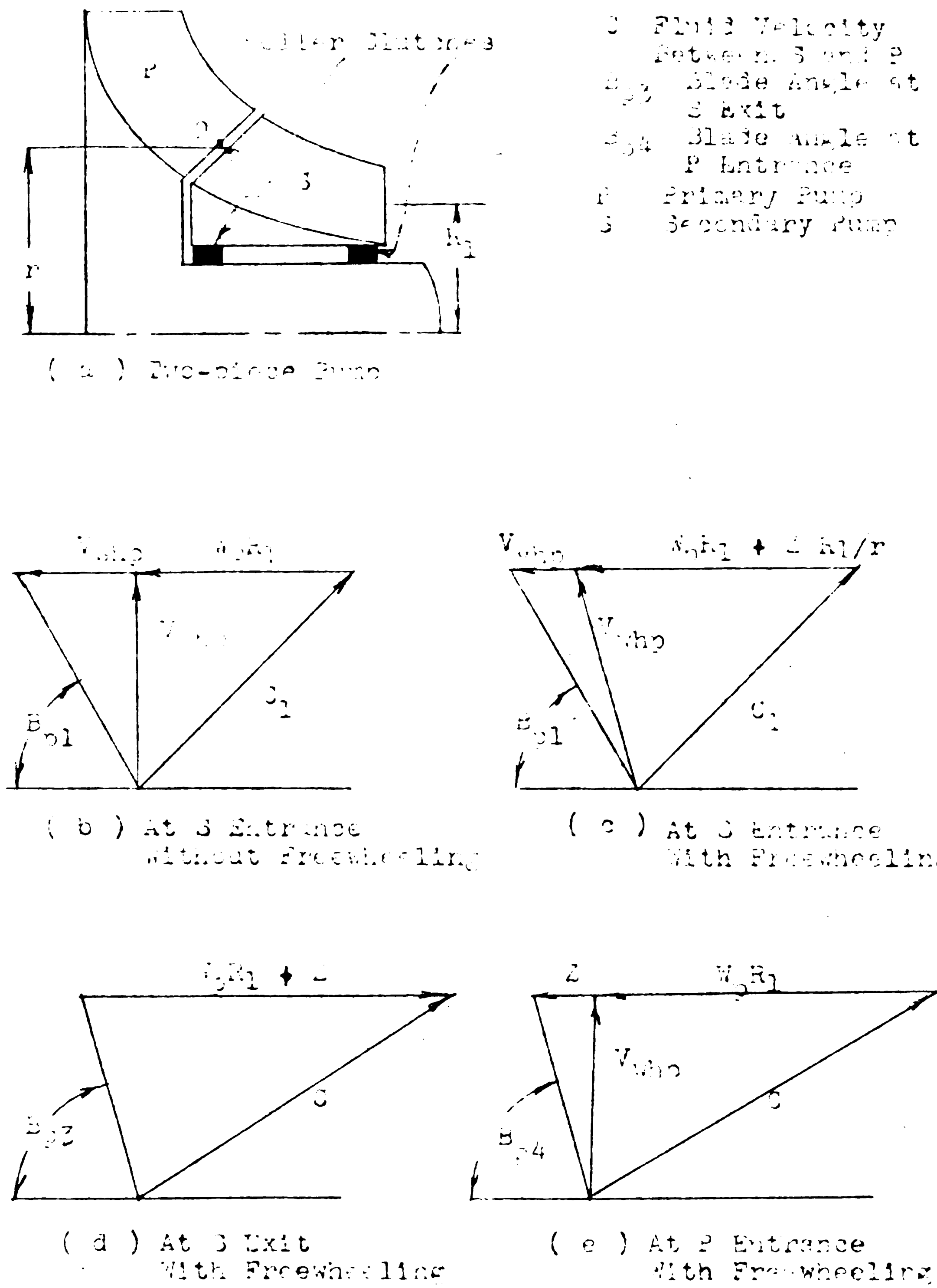


Fig. 11 Velocity Diagrams at the Primary and Secondary Pumps with and without Freewheeling

At low turbine speed, the fluid tends to throw a positive torque on the pump, and as a result the secondary pump freewheels ahead of the primary pump. When the turbine speeds up, the direction of the flow velocity of the fluid relative to the pump changes, and the secondary pump is locked to the primary pump as the flow direction passes the blade angle.

Fig. 11b, c, d, and e show the velocity diagrams at the pump entrance without and with freewheeling units. Z being the difference of circumferential velocities of points p and s on P and S respectively.

The shock loss at S without freewheeling is

$$L_{shp} = \frac{V_{shp}^2}{2g}$$

With freewheeling unit, the shock loss at piece S is (See Fig. 11c)

$$\frac{\left(V_{shp} - Z \frac{R_1}{r} \right)^2}{2g}$$

And the shock loss at piece P is

$$\frac{Z^2}{2g}$$

The total shock loss with freewheeling unit is, then

$$L'_{shp} = \frac{\left(V_{shp} - \frac{ZR_1}{r} \right)^2 + Z^2}{2g}$$

The percentage in shock loss saving with freewheeling unit at a velocity differential Z is

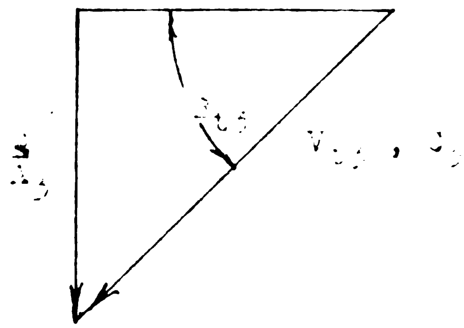
$$\frac{L_{shp} - L'_{shp}}{L_{shp}} 100 = \frac{2\, V_{shp}\, Z\, \frac{R_1}{r} - Z^2 \quad 1 + \left(\frac{R_1}{r} \right)^2}{V_{shp}^2} \quad 100 \qquad (20$$

Equation (20) can also be expressed as

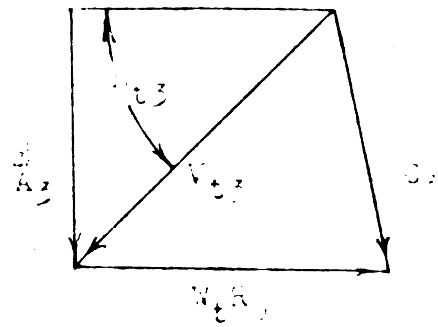
$$\frac{L_{shp} - L'_{shp}}{L_{shp}} = 2\, \frac{R_1}{r}\, \frac{Z}{V_{shp}} - \left(\frac{Z}{V_{shp}} \right)^2 \quad 1 - \left(\frac{R_1}{r} \right)^2 \qquad (20)^1$$

The saving is a function of Z/V_{shp}, and the radius ratio R_1/r. It is highest when R_1/r equals to 1, i.e., when the entrance radius at P is equal to the entrance radius at P'.

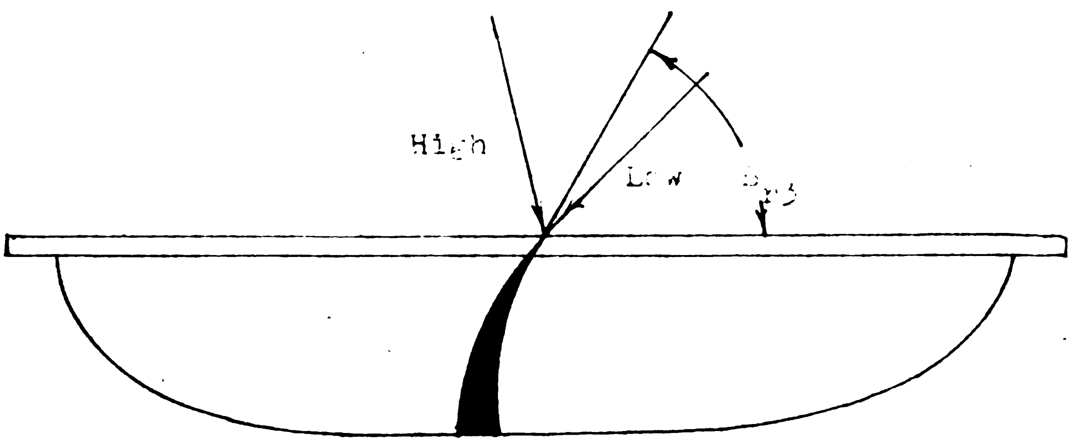
<u>DESIGN FEATURE FOR REDUCING SHOCK LOSS AT THE REACTION MEMBER</u>. The shock loss at the reaction member is the highest compared with those at the turbine and at the pump. Fig. 12a shows the velocity diagram at the turbine exit when the turbine stalls. The fluid rotates in a negative direction, i.e., in the opposite direction compared to the motion of the pump. But as the turbine speed increases, the reduction in the fluid rate Q together with the increase of pump speed change the direction



(a) At Turbine Exit
$W_t = 0$



(b) At Turbine Exit
High W_t



(c) Relative Entrance Velocities at the Reaction Member Entrance at Low and High Turbine Speeds

Fig. 1[illegible] Velocity Diagram at the Turbine Exit and the Reaction Member Entrance.

of rotation of the fluid to that of the pump as shown in Fig. 12b. The fluid also throws a positive torque on the reaction member at high turbine speed which tends to turn the reaction in the direction of the pump motion. (See Fig. 12c.)

In order to reduce the shock loss at the reaction member entrance at high turbine speeds, the reaction member is mounted on the converter housing with roller clutches. As soon as the fluid begins to throw a positive torque on the reaction member, the roller clutches allow the reaction member to rotate freely in the direction of the motion of the pump. As the reaction member freewheels, the torque converter has a similar characteristics of a hydraulic fluid coupling, which has high efficiency at high turbine speed.

The efficiency of the torque converter at intermediate turbine speed can be increased by dividing the reaction member into two parts. Both parts are mounted independently on the converter housing by roller clutches.

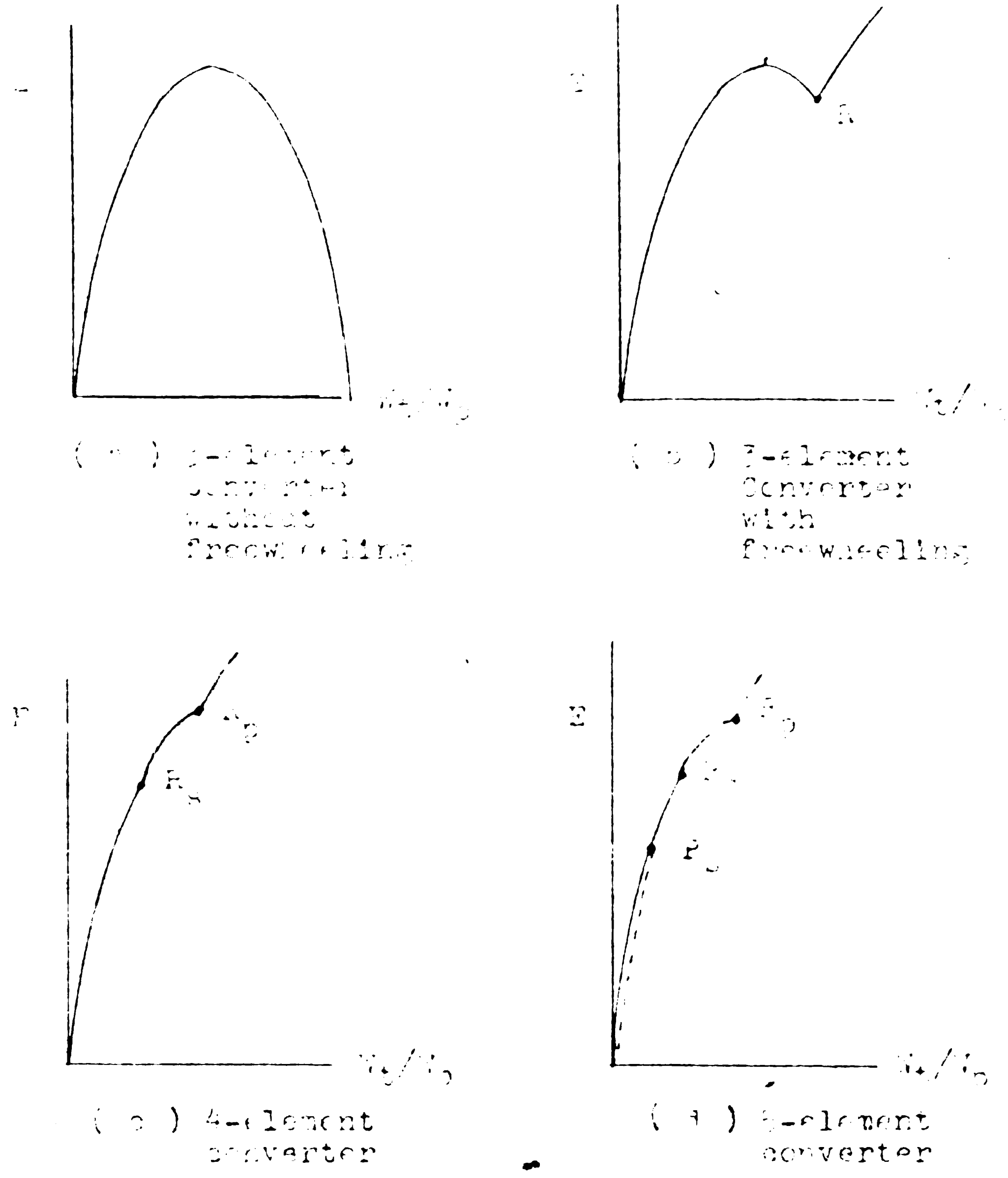
Both reaction members are locked to the housing when the turbine stalls or at low turbine speed. But at some designed point in the intermediate turbine speed range the secondary reaction member which meets the fluid first will start to freewheel. As the turbine runs at high speed, the primary reaction member also start to freewheel at some designed point in the direction of rotation of the pump.

IMPROVEMENT OF CONVERTER PERFORMANCE BY FREEWHEELING UNITS. The torque converter efficiency curve of a three-element converter without freewheeling unit is shown in Fig. 13a. The efficiency rises as the turbine speeds up until the designed point is reached after that point the efficiency falls off rapidly as a result of high shock losses at different wheels.

With a freewheeling unit as the reaction member the efficiency curve goes up at the clutch point R, as shown in Fig. 13b, when the reaction member starts freewheeling. A remarkable increase in efficiency at high turbine speed is indicated.

The torque converter efficiency at the intermediate speed range can be increased by using two reaction members. The secondary and the primary reaction members freewheel at intermediate and high turbine speeds respectively. The efficiency curve is shown in Fig. 13c, where R_s and R_p are the clutch points for the secondary and primary reaction members respectively. This kind of converter is called four element torque converter.

When a two piece pump takes place of the one piece pump in the four-element torque converter, it is called a five-element converter. In a five-element converter, the



(a) 3-element converter without freewheeling

(b) 3-element Converter with freewheeling

(c) 4-element converter

(d) 5-element converter

Fig. 13 Efficiency vs. Speed Ration Curve for the Three, Four, Five Element Converters.

secondary pump is locked at a low turbine speed. The increase of efficiency of the five-element converter over that of the four-element at low turbine speed is shown in Fig. 13d.

PART 2

TYPES OF HYDRODYNAMIC TORQUE CONVERTERS

TYPES OF HYDRODYNAMIC TORQUE CONVERTERS

The discussion of types of hydrodynamic torque converters in this thesis is confined to those used in the automobiles. Generally speaking, all of them can be classified as either a three or five element torque converters. They produce maximum stalling torque ratios at a little over two to one, and they are cooled either by air or by engine cooling water with heat exchangers.

The torque converters used in the different makes of automobiles will be discussed in the following order: (1) Chevrolet, (2) Buick, (3) Ford and Mercury, (4) Packard and (5) Studebaker.

TORGUE CONVERTER IN CHEVROLET POWERGLIDE TRANSMISSION. (Refer to Fig. 14) The torque converter used in the Chevrolet Powerglide transmission is a five-element, polyphase type. It consists of the following five major elements:

1. Primary pump with 29 vanes.
2. Secondary pump with 31 vanes.
3. Turbine with 31 vanes.
4. Primary stator with 25 vanes.
5. Secondary stator with 23 vanes.

Each of these elements is a vaned wheel made from steel stampings which are assembled in place, spot welded and copper-brazed into an integral unit.

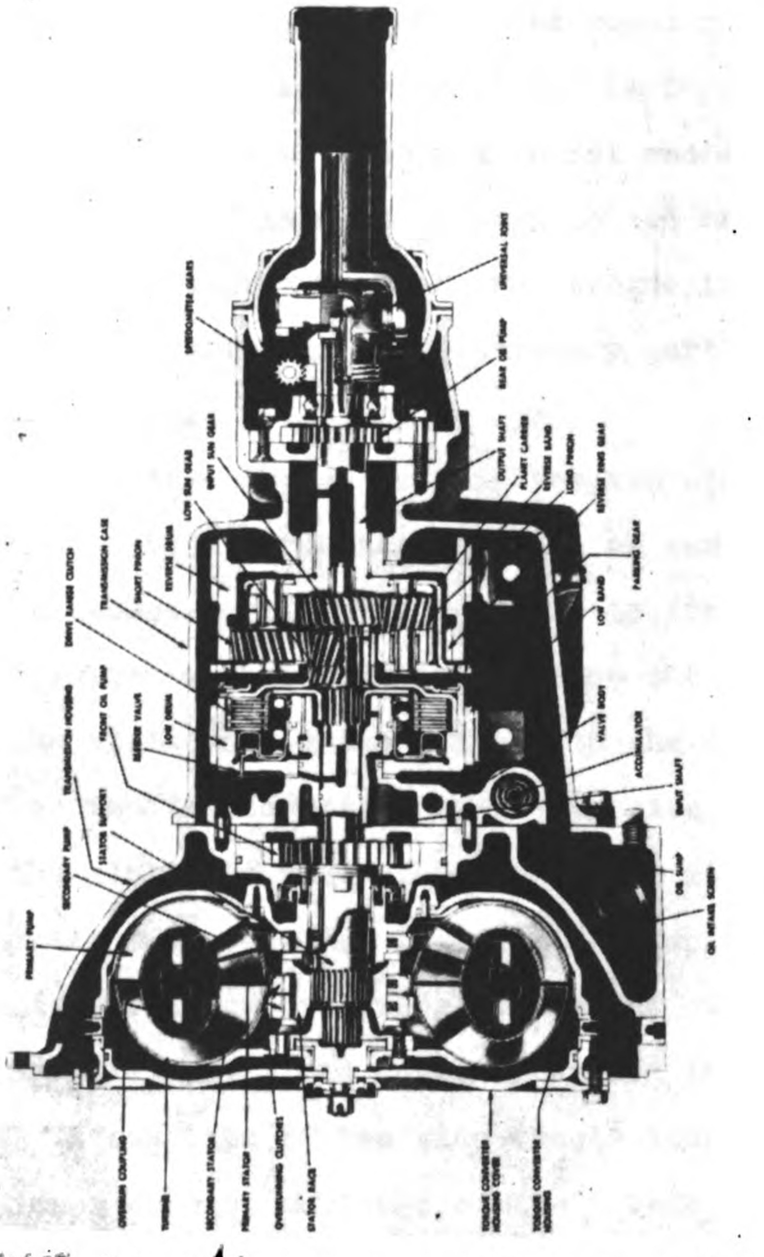


Fig. 14 Longitudinal Section of the Powerglide Unit

The primary pump is spot-welded to the torque converter housing which, in turn, is bolted to the flywheel member. The turbine is attached to the housing cover at the front by a special machined bolt but is permitted to turn freely on a bronze bushing and thrust washer. The input shaft is splined into the flange bolted back of the turbine hub; while to the rear of the turbine it runs through the stator support which is the stationary part of the transmission housing.

The secondary pump as well as the two stator members are mounted on overrunning clutches of cam and roller type. The clutches permit either stator to free-wheel in the same direction as the pump and turbine but lock the stator to the stationary stator race with the development of any forces tending to rotate it in opposite direction. Similarly the clutch on the secondary allows it to turn faster, but never slower than the primary pump.

A unique feature of Powerglide is the overrunning coupling, an auxiliary fluid coupling within the torque converter. It consists of two vane wheels installed between the inner concave surfaces of the primary pump and the turbine, one being attached to the pump and the other to the turbine. The coupline element provides additional engine braking and makes possible the extremely low speed push start.

Stalling torque ratio is of the order of 2.2 to 1. At that point both primary and secondary stators are fully locked by their clutches. At the same time the secondary pump is overrunning the primary pump to reduce shock loss. As the torque converter picks up speed and the demand for torque multiplication begins to lessen the secondary pump slows down and locks to the primary pump hub, thus integrating both pump elements into a single unit giving maximum effect from now on through the fluid coupling phase. The secondary stator then unlocks and begins to free wheel. Then as engine speed reaches the minimum value at which engine torque alone can carry the load, the primary stator unlocks and join the secondary stator in free-wheeling phase. At this point the torque converter begins to enter the fluid coupling phase.

At full throttle acceleration on a level road, the secondary pump stops overrunning the primary at about 30 mph.

The torque converter oil is cooled by a water cooler.

TORQUE CONVERTER IN BUICK DYNAFLOW TRANSMISSION. The torque converter used in the Buick Dynaflow transmission is similar to Chevrolet converter in appearance except the omission of the overrunning coupling. It consists of five elements: primary and secondary pump, turbine, primary and secondary reaction members. All of them are made from die-cast aluminum alloy.

The primary pump is driven from the engine shaft through a flexible disk, which takes the place of a fly-wheel. This flexible disk reduces the mechanical vibration transfer from the engine to the transmission and also lessens the harmful effect due to misalignment of the engine and the transmission shafts.

Before the fluid reaches the primary pump it passes through a secondary pump, which is supported on the hub of the primary pump through a roller clutch. From the primary pump the fluid passes into the turbine, in which it flows toward the axis of rotation. The turbine discharges the fluid into the two part reaction member. Both are mounted on roller clutches, of which the stationary ring of race is secured to the transmission housing. These clutches allow the reaction members to turn freely in the direction of pump rotation whenever the direction of flow at their entrances tends to turn them in that direction, but prevent them from turning in the opposite direction.

The maximum stalling torque ratio is 2.25 to 1 when both primary and secondary reaction members are locked by their clutches, and when the secondary pump is freewheeling ahead of the primary pump. As the torque converter picks up speed, the secondary pump is locked to the primary pump hub, and both pumps become an integral

unit. Then as the torque converter speeds up further, the secondary reaction member starts to freewheel. Finally the primary reaction member freewheels as soon as the engine torque alone can carry the load.

At full throttle acceleration on a level road, the secondary pump stops overrunning the primary pump at about 37 mph. The torque converter is oil cooled by using an heat exchanger.

<u>TORQUE CONVERTER IN MERCURY-FORD TRANSMISSION</u>. (Refer to Fig. 15.) The torque converter used in the Mercury-Ford transmission is composed of three elements: pump turbine and stator (reaction member). It is driven from the engine through a flexible plate for minimum vibration transfer from the engine to the transmission. Fins cast on the pump cover provide added cooling surface and serve as vanes for pumping air. Air for cooling is drawn in at the lower left side, passes through the converter housing by centrifugal action, and is exhausted at the lower right side. This method of cooling the converter and transmission oil eliminates the need for plumbing and is said to be completely effective under all operating conditions.

Slots are provided in the inside of the die cast pump cover to locate 31 stamped steel blades, each of which is held in place by four tabs while a retaining

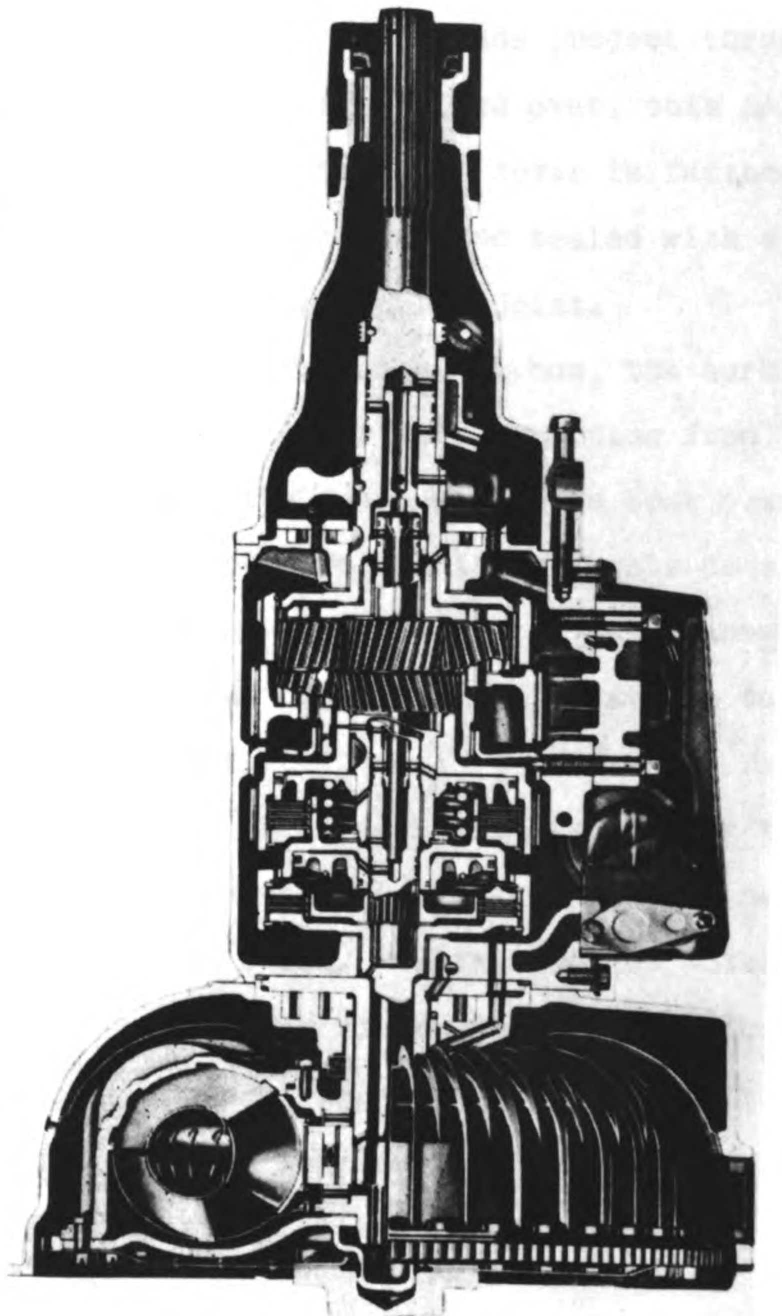


Fig. 15 Longitudinal Section of the Ford-Mercury Automatic Transmission

ring assists in holding all of the blades. An additional pair of tabs on each blade project through slots in the torus ring and are rolled over, this holding the torus ring in place. The pump cover is fastened to a cast iron hub by cab screws, and sealed with a synthetic ring to prevent leakage at the joint.

Except for a forged steel hub, the turbine is made of steel stampings. Four tabs extending from each of the 33 blades through the outer shell are bent over to hold the blades in place. Two additional tabs on each blade hold the torus ring in place in the same manner as in the pump assembly. The outer shell is fastened to the splined steel hub by rivets.

The stator is an aluminum die casting to which a formed, split steel shroud is assembled and held in place by welding at the split. Splines on the outer race of the sprag type overrunning clutch prevent rotation in the stator hub, and snap rings in conjunction with two supports hold the race in place.

The maximum stalling torque ratio is in the order of 2.1 to 1.

TORQUE CONVERTER IN PACKARD ULTRAMATIC TRANSMISSION. The torque converter as shown in Fig. 16 in Packard Ultramatic transmission appears to be a four element torque converter. Actually it is a three-element type with two

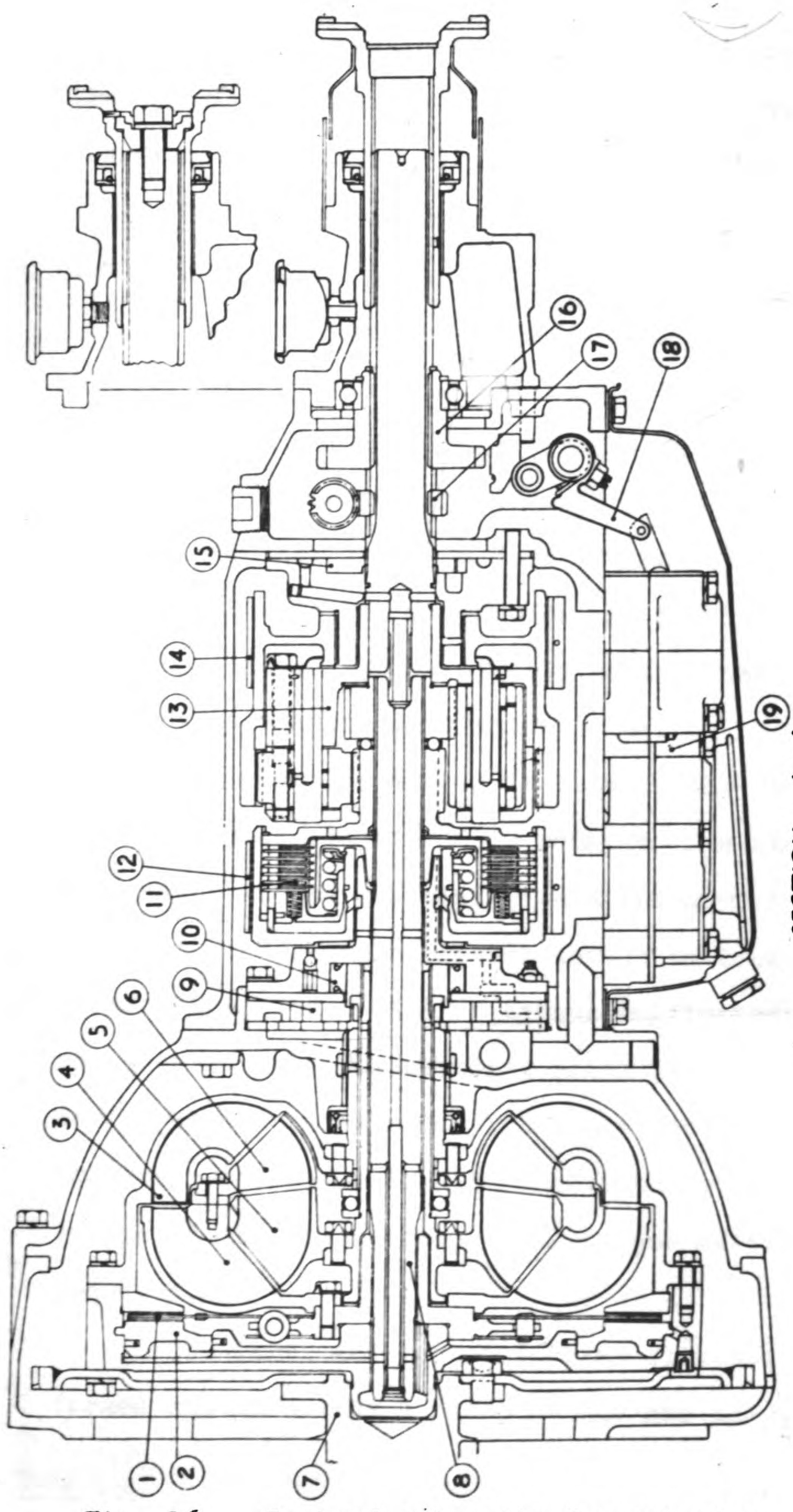


Fig. 16 Packard Automatic Transmission

turbines bolted together. The first turbine is nearest to the engine end and has been bolted to it the smaller diameter second turbine stage which is seen at the rear end of the torque converter. Being bolted together they act as a unit.

The pump, which also serves as the housing for the assembly, is at the rear and is carried forward to bolt to the flexible disk on the crankshaft. The latter replaces the conventional flywheel, since the entire mass of the converter is now a part of the flywheel system. The disk also serves to provide the flexibility required in a complicated system of this nature, making up for any tendency to misalignment at any time.

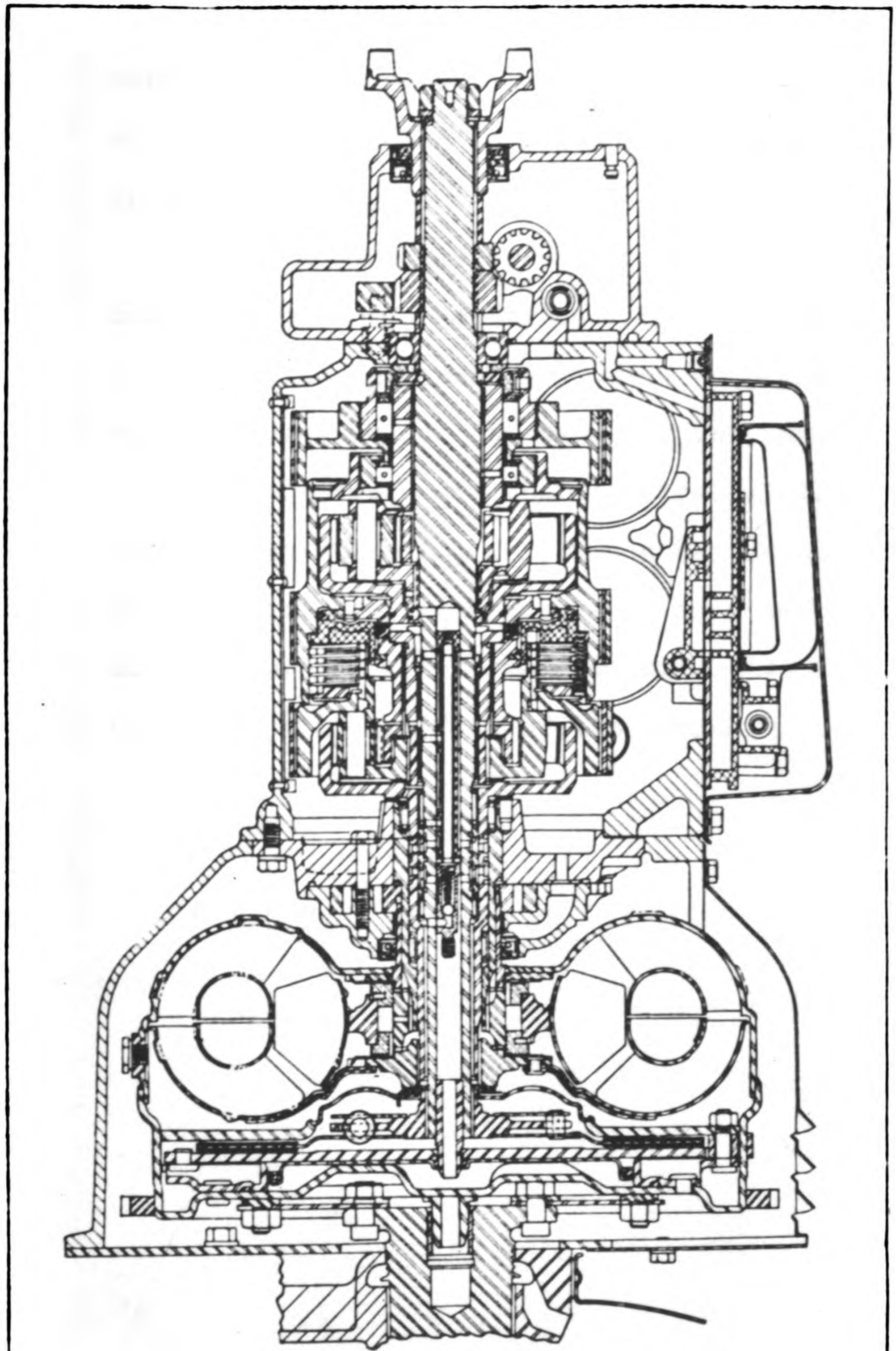
The reaction member is between the first and second turbine stages and is mounted on a sleeve which terminates at the bulkhead with a one-way clutch of sprag type. The overrunning clutch serves to lock the reaction member during the phase of torque multiplication, then releases it when the coupling point is reached. The converter then acts as a coupling until the direct clutch engages. Consequently, the reaction member is free to rotate at all times except when torque multiplication is required.

The first and second turbine elements are made with formed blades having a rounded nose and sharp exit. The pump blades, too, have a round nose and sharp exit but incorporate a constant section for the major portion of the length of the blade measuring from the nose. The second turbine element is so formed and positioned as to control the flow into the pump to achieve a rising input speed curve, starting with an engine speed of around 1600 rpm. At the same time it has the effect of extending the clutch point for smooth engagement. Through the action of modulator valving the torque converter contains an oil pressure of about 30 psi.

Although the converter is a three element type with but one reactor stage for torque multiplication, yet it is able to attain a stalling torque ratio in the order of 2.4 to 1.

The converter, as well as those used in the Chevrolet Powerglide and Buick Dynaflow transmissions, is cooled by a water-cooled oil cooler.

TORQUE CONVERTER IN STUDEBAKER TURBOMATIC TRANSMISSION. The torque converter used in the Studebaker Turbomatic transmission, as shown in Fig. 17, is of the three element type having the pump member connected to the crankshaft. The turbine member, at the left, connected to the output shaft; and the stator member, for torque multiplication and reverse of fluid flow, at the center. The latter is mounted on a sprag type free-wheel unit



Longitudinal section of Studebaker's fully automatic transmission showing details of the torque converter, single plate clutch for direct drive, two planetary gear sets, etc.

Fig. 17

connected to the case to permit only one direction of rotation. Maximum torque multiplication, at stall, is of the order of 2.15 to one and this occurs at an engine speed of 15 rpm.

The torque converter is made up of fabricated elements and does not employ castings for the wheels. In essence it is an assembly of stamped pieces with individual stampings for the vanes of the three elements.

The torque converter is air-cooled. This is effected by attaching a formed sheet metal shell or baffle fitted with eight vanes to serve as a centrifugal pump, thus giving the air stream positive direction and forcing it to scrub the surface of the torque converter.

BIBLIOGRAPHY

Eksergian, Rupen, "On the Reaction of Fluids and Fluid Jets", Franklin Institute Journal, Vol. 237, No. 5, pp. 385-410, (May, 1944).

Eksergian, Rupen, "The Torque Converter and Fluid Coupling", Franklin Institute Journal, Vol. 235, No. 5, pp. 441-478, May, 1943).

Heldt, P. M., Torque Converters, P. M. Heldt, Nyack, N. Y., 1947.

Kelly, O. K., "How the Torque Converter Works", SAE Journal, Vol. 57, No. 2, pp. 22-25, (Feb., 1949).

Kelly, O. K., "Making Torque Converters More Efficient", SAE Journal, Vol. 57, No. 4, pp. 20-23, (April, 1949).

Modern Automatic Transmission, Chilton Company, Philadelphia, Penna., 1950.

Rouse, Hunter, Elementary Mechanics of Fluids, John Wiley & Sons, N. Y., 1945.

Spannhake, E. W., "Hydrodynamics of the Hydraulic Fluid Coupling", SAE Journal, Vol. 57, No. 8, pp. 59-66, (August, 1949).