1971

Design of a semi-automatic transmission

Yogendra Prahladray Buch

DESIGN OF A SEMI-AUTOMATIC TRANSMISSION

BY

YOGENDRA PRAHLADRAY BUCH, 1946-

A

THESIS

submitted to the faculty of

UNIVERSITY OF MISSOURI - ROLLA

in partial fulfillment of the requirements for the

Degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

Rolla, Missouri

1971

Approved by

Wm S Gatley (advisor)

Robert L. Davis

ABSTRACT

Design of a semi-automatic four-speed transmission for automotive engines with moderate horsepower is described. The basic configuration consists of main and countershafts and five pairs of constant-mesh gears. Engine torque is transmitted through an input gear pair to the countershaft, and finally to the output shaft through a selected output gear pair. Selection is accomplished by engaging one of five hydraulically operated clutches located on the countershaft. Engagement fixes the desired gear to the countershaft and engine torque is then transmitted to the mating gear, which is fixed to the output shaft. Fourth speed is obtained by coupling the input shaft directly to the output shaft through a sixth clutch. Design of clutches, gears, shafts, supply pump and control valves is analyzed in detail. Requirements for a master clutch, for use during acceleration from rest, are summarized.

ACKNOWLEDGEMENTS

The author wishes to express his appreciation to Dr. William S. Gatley for his encouragement, direction and assistance throughout this project.

The author is also thankful to Mrs. Robert Brown for her helpful suggestions in constructing the final format and to Mrs. Judy Hausman for her wonderful cooperation in typing this thesis.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

I. INTRODUCTION

From a socio-economic point of view, human wants become necessities and the need structure is ever expanding continuously. As a result, man is constantly engaged in developing ways to satisfy these wants most economically. As wants are numerous, so the approaches to satisfy them are also numerous. The objective of the present project is to try to satisfy one such major want, which may also satisfy several auxiliary wants.

When man began to employ power for terrain vehicles, he became more aware of the various possibilities for the use of this power and he looked for easier and more convenient ways to control it. His immediate need was to obtain variable torques and speeds without disturbing the engine operation. He fulfilled this need by a clutch and gearbox combination. The gearbox had one row of gears mounted on a cluster which was to slide as required. This, however, required gear shifting and operation of a clutch pedal. Gear sliding was not only lacking in versatility, but also required above average skill to avoid any noise problem.

To shift under power, attempts at using automatic clutch operations, synchronizers, epicyclic gears, etc. were made. These, however, were expensive or complex. Some required the use of a torque converter which added to the loss of the power of the engine in the form of heat.

Man could even develop such an arrangement so as to automatically control the form of power to facilitate handling by inexperienced drivers. This was done by using such transmissionsas Ford-O-Matic, Dynaflow, Torque-flite, Hydramatic, etc. In most cases, road conditions caused a hydraulic fluid to actuate, by several valves, a band or clutch to engage, which changed power flow through an epicyclic gear train. While this relieved the driver from exercising his own judgement, it proved inevitably complicated, bulky and at times heavy. Heavy transmissions reduced power to weight ratio, which decreased road performance and increased fuel consumption in most cars.

The present project appears to have succeeded in finding a combination to solve most of the above mentioned problems. It also achieves vehicle flexibility in terms of road speed and torque by employing several gear pairs of different sizes. This proposition is similar to a constant mesh transmission system. Here, power is directed to the counter shaft through a fixed input gearset. On the counter shaft, pinions of other gear pairs rotate freely. Each of them can transmit the power to the output shaft only when engaged by a separate hydraulically operated clutch provided. Expensive synchronizers are, thus eliminated. To eliminate any manual engagement effort, electric solenoids are proposed to engage clutches when desired. When the engine is operated, the pressure is immediately supplied by an engine-driven gear pump, so the driver has

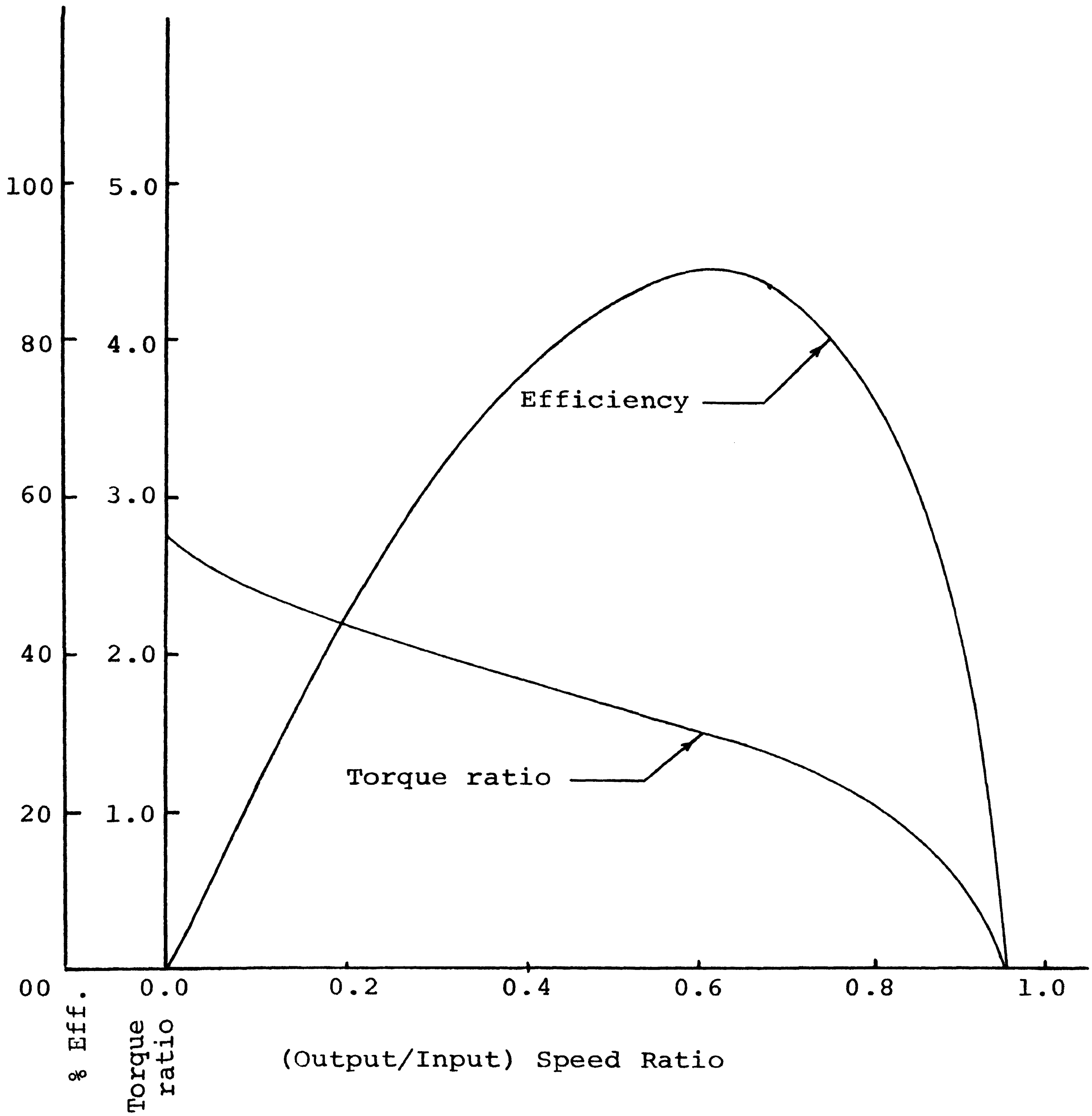


Figure 1: Performance data for a typical torque converter. |2, P. 131|

only to press a button to direct the pressure to a right clutch through the respective solenoid valve.

As the curve in Figure 1 shows, a typical torque converter has an efficiency of 80 to 85 percent. Similarly, maximum efficiency of a plain gear train is about 90 percent, so if both have to be used, about 25 percent of the engine power is lost. In the present approach, one can eliminate the use of a torque converter since the driving speeds can be more flexibly selected.

Experience has shown that fully automatic transmissions are better suited to cars having high power to weight ratio than to average, small and medium cars, [2, P. 20]*. For cars with a low power to weight ratio, a constant mesh transmission is especially attractive because of its lower weight and greater efficiency.

---

*Numbers in brackets indicate reference numbers found in the Bibliography.

II. REVIEW OF LITERATURE

The automobile transmission is a very broad subject involving design of many major mechanical components such as gears, shafts, clutches, seals, springs, bearings, etc. It also involves a study of hydraulics and fluid pumps. Therefore, this magnifies the scope of literature to be surveyed.

To develop a systematic approach, it was decided to review the overall history of automotive transmissions. Heldt [1] deserves to be mentioned first for reviewing operationally, most of the transmissions up to 1942. He includes several transmissions with magnetic clutches, with electric drives and also many hydrostatic, hydramatic, variable throw and pneumatic transmissions. He knowledgeably and in length discusses constructional details of many friction clutches and to a lesser extent of control devices. Judge [2] discusses the details of developments up to 1962. He, therefore, describes the operational and constructional features of various automatic transmissions. In a small chapter, he discusses how a semi-automatic transmission provides an effortless operation, the transmission in which the clutch pedal is dispensed with and still the choice of gear selecting is left to the driver. He goes a bit farther and summarizes the most recent developments in transmissions. The idea of the present design resembles Zahnradfabrik--Friedrichshafen, i.e. Z-F hydromedia semi-automatic transmission, described briefly there.

To complete the study of existing transmission designs, Larew's work [4] was studied which, in addition to describing planetary automatic transmissions with their control systems, also discusses the operational characteristics of Ford-O-Matic, Cruise-O-Matic, etc. transmissions.

An improved version of a constant mesh transmission was found to be the best choice for the proposed objective. This required five separate clamping devices and so to be most compact, hydraulically controlled plate clutches were selected. Hydraulic control offers advantages in simplicity of construction; smaller size and better accessibility.

Boner's work [11] was found useful in selecting the right transmission fluid. Our selection of electric solenoid control valves depended on a study of the works of Ernst [9] and Pippenger [10].

Having considered engagement from all points, we now concentrated on disengagement of respective clutch plates. Simplicity, cost, east in mounting, etc. favor the selection of Belleville spring washers whose design is described in Design Handbook [6]. Other springs for drain valves, control valves, etc., were also designed according to a procedure given in the same handbook.

The problem of transmitting hydraulic fluid force to the clutch plates in the form of sliding motion was solved with the aid of Seals book [7] and Gask-O-Seal [8] . Several other books on hydraulic equipment were consulted for determining pipe diameters, types of pumps, etc. Works of Timoshenko [12] and Shigley [13] were used in selecting materials for most parts and also for various design criteria. Dudley [5] was helpful in determining gear sizes, Refs. [14] to [18] for refreshing and understanding certain theories by illustrative examples, and Morton [19] was helpful in selecting and designing various bearings.

III. GENERAL DESCRIPTION

We know that a typical automobile requires a starting torque of about nine times the average engine torque. To be equally efficient on any road condition and at the same time, to cover a very wide range of road speeds, modern American tradition is to provide a powerful engine and thus increase the power to weight ratio. We propose instead an average six cylinder 230 cu. in. engine with 150 H.P. and 250 lb-ft. as maximum engine torque.

A. Gear Ratios

This engine is almost similar to the base engine in the 1970 Chevrolet Camaro where gear ratios are:

2.85:1 for low, 2.02:1 for second, 1.35:1 for third, 1:1 for fourth, and 2.85:1 for reverse gears.

When different gear pairs are selected to transmit power, the engine will momentarily decelerate and then accelerate. Providing ratios in geometrical series enables the engine to be accelerated in the same range of speeds while changing each gear. However, while changing to high gears, the momentary reverse load on the engine is found high enough to stall the engine, so generally, gear ratios are not selected in exact geometrical series. From this consideration and by comparison with ratios in current use, we tentatively select the following ratios which might have to be revised a bit when gears are selected:

2.8:1 for low, 1.9:1 for second, 1.4:1 for third,

1:1 for input set, and 2.8:1 for reverse gears.

Selection of four speeds this way results in saving fuel consumption, as overall speed reduction between engine and wheels is often smaller so engine revolutions per mile travelled are less.

B. Gear Sizes

One of the principal dimensions of any general transmission is the center distance between main and counter shafts. It generally varies between 0.5 $\sqrt[3]{T}$ and 1.0 $\sqrt[3]{T}$ |1, P. 137|, depending upon the form of gear tooth to be used. This and the tentative size of the clutch guides us to keep the center distance, d = 0.7 x $\sqrt[3]{T}$ in. where T is torque on the driving shaft in lb-ft. So,

$$d = 0.7 \times \sqrt[3]{250}$$
$$= 4.43\text{" or say } 4.4\text{"}.$$

We now calculate all pitch diameters. For that, we adopt the convention that all capital letters refer to the main shaft and all small letters refer to the counter shaft. Refer to Departmental Report for further details.

Low: Assumed ratio is 2.8:1. To minimize clutch capacity, the input gearset should not multiply engine torque; i.e., $r_4 = R_4$. This gives:

$$r_1 = 0.358\ R_1 \tag{1}$$

and $$r_1 + R_1 = 4.4. \tag{2}$$

From (1) and (2), $R_1 = 3.24"$ or say 3.3" and, therefore, $r_1 = 1.1"$. This gives the revised ratio as 3.3/1.1 = 3.0. i.e. 3.0:1

<u>Second</u>: Assumed ratio is 1.9:1. So,

$$r_2 = 0.527\ R_2 \tag{3}$$

and $$r_2 + R_2 = 4.4 \tag{4}$$

From (3) and (4), $R_2 = 2.88"$ or say 2.9" and therefore, $r_2 = 1.5"$. This gives a revised ratio as 1.9/1.5 = 1.935. i.e. 1.935:1

<u>Third</u>: Assumed ratio is 1.4:1. So,

$$r_3 = 0.715\ R_3 \tag{5}$$

and $$r_3 + R_3 = 4.4 \tag{6}$$

From (5) and (6), $R_3 = 2.56"$ or say 2.6" and therefore, $r_3 = 1.8"$. This gives a revised ratio as 2.6/1.8 = 1.44. i.e. 1.44:1

<u>Input set</u>: $r_4 + R_4 = 4.4"$ and so $r_4 = R_4 = 2.2"$. Ths also gives a final ratio 1:1. i.e. 1:1

<u>Reverse</u>: Assumed ratio is 1.8:1. Line connecting centers of counter and Idler shafts is assumed to be at an angle of 60^0 with a vertical line connecting centers of the counter shaft and the main shaft. We also assume:

$$r_{rev} = r_1 = 1.1".$$

Now,

$$R_{rev}/R_{idler} \times R_{idler}/R_{rev} \times r_4/R_4 = 2.8.$$

Substituting;

$$R_{rev} = 3.08" \text{ or say } 3.1".$$

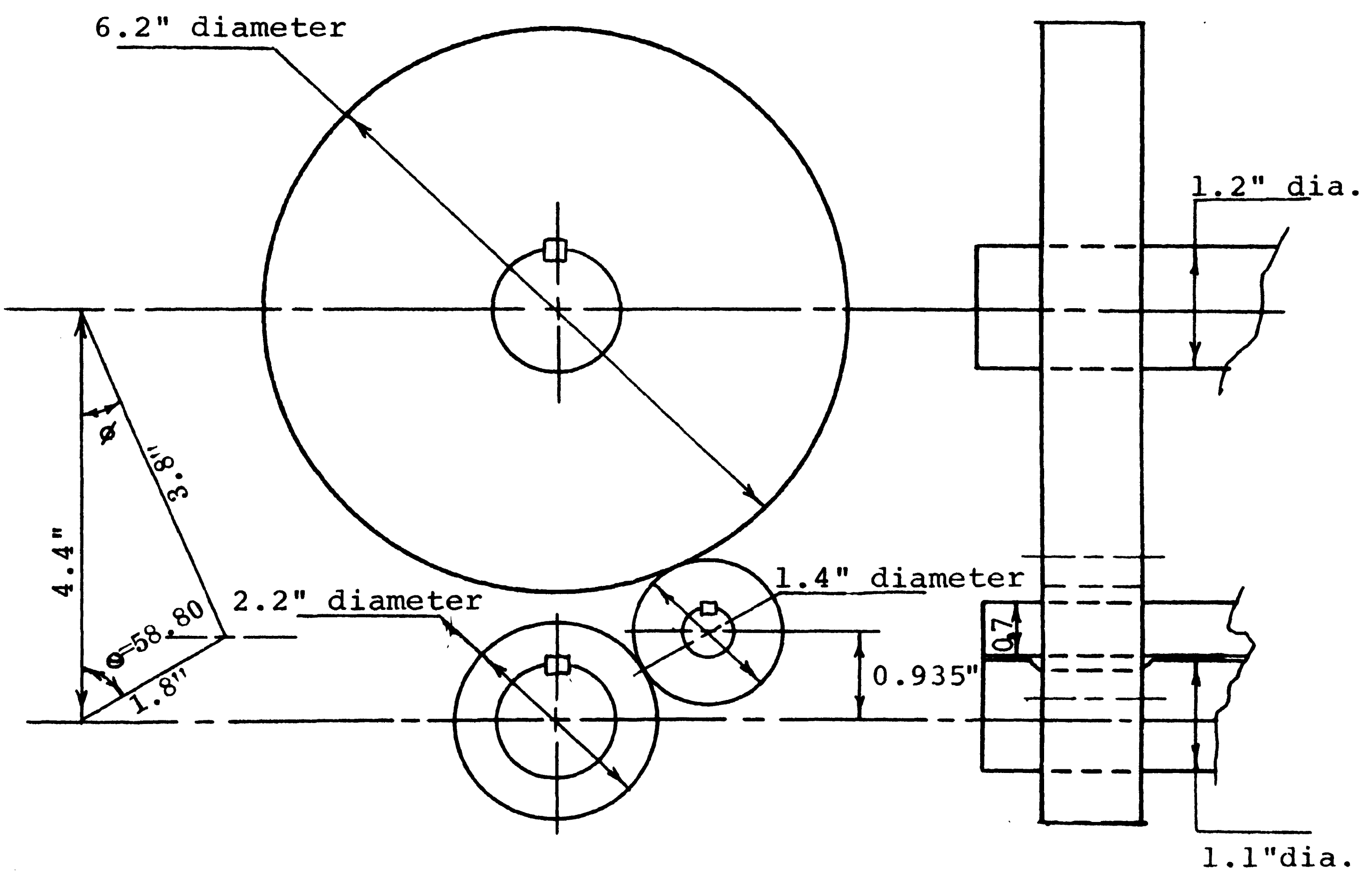


Figure 2: Schematic lay out of the reverse gear train

So, revised ratio is 3.1/1.1 = 2.82, i.e. 2.82:1.

To determine R_I, the radius of the idler gear, we refer to Figure 2 and derive:

$$(1.1 + R_I) \cos 60 + (3.1 + R_I) \cos \phi = 4.4'' \quad (7)$$

$$(1.1 + R_I) \sin 60 = (3.1 + R_I) \sin \phi \quad (8)$$

From these and the identity,

$$\sin^2\phi + \cos^2\phi = 1.0$$

we get:

$$R_I = 0.715''.$$

For keeping a whole number of teeth, we propose $R_I = 0.7''$ which changes the location of the idler shaft. If the new inclination described above is θ:

$$1.8 \cos \theta + 3.8 \cos \phi = 4.4'' \quad (9)$$

$$1.8 \sin \theta = 3.8 \sin \phi \quad (10)$$

From these and the identity,

$$\sin^2\phi + \cos^2\phi = 1.0,$$

we get:

$$\theta = 58.8^0$$

This makes the vertical distance between centers of the idler and counter shafts:

$$= (1.1 + 0.7) \cos 58.8$$

$$= 0.935''.$$

IV. GEARS

We have fixed pitch diameters of each gear and so we proceed to design tooth dimensions and calculate stresses in detail. The first step is to select the right kind of gears. As we have parallel axes and as there are no internal gears, the choice reduces to a selection between spur and helical gears. Helical gears carry very high horsepower and can operate and engage very quietly. However, as our engine horsepower is limited and as it is to be a constant mesh transmission, the final choice is spur gears which are cheaper to manufacture. They can be hobbed, shaped, milled, stamped, drawn, sintered, cast or shear cut, which means that more of these methods, suitable to this application are available to make spur gears than any other types.

A. Tooth Form

We know that the universally used tooth form is the involute form. When two such gears mesh, we can always find an imaginary circle on each gear, at the intersection of which there is no sliding velocity. Such a circle is called pitch circle and its diameter, pitch diameter.

At any point on the tooth profile, a tangent to the profile will make a constant angle with that gear's radial line through this point. This angle is called the pressure angle. To avoid excessive undercutting and also to extend the effective depth of teeth well below the pitch circle, a pressure angle of 20^0 is adopted.

Automobile gears are generally very heavily loaded; to make gear teeth stronger, the AGMA (American Gear Manufacturers Association) has developed a shortened tooth system for automotive and other similar purposes. This is called the stub tooth system. These gears are sometimes designated by double pitch numbers which means that the radial tooth dimensions are calculated on the basis of one diametral pitch and circumferential dimensions on the basis of the other. To make tooth width greater, circumferential dimensions are based on the smaller diametral pitch.

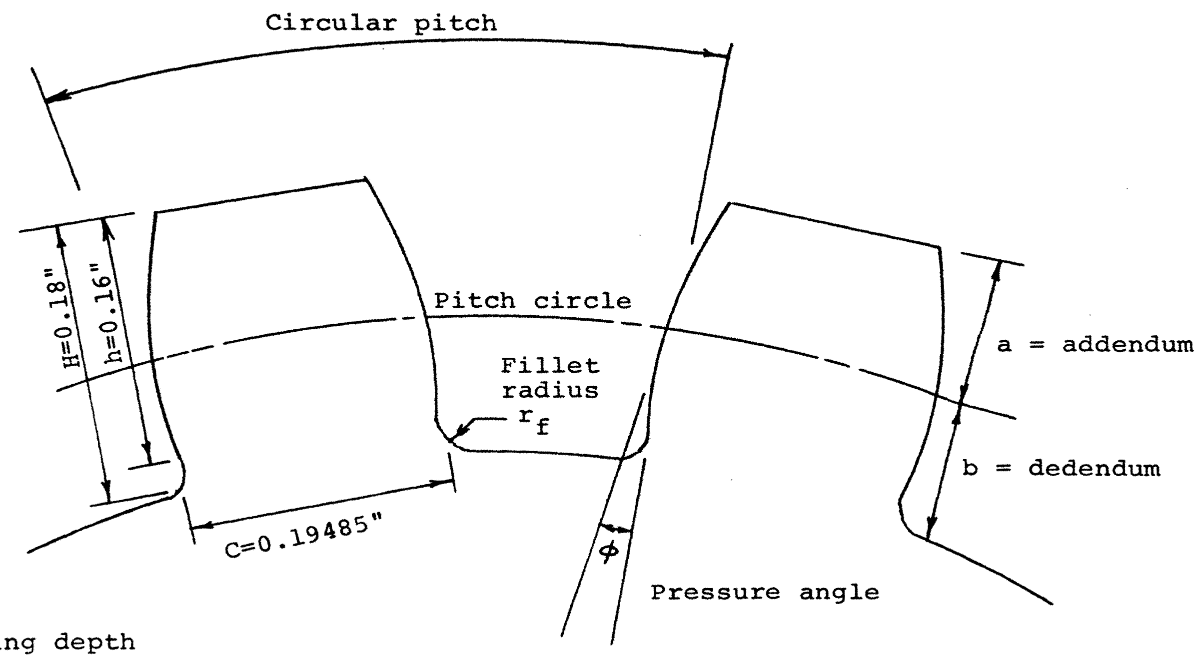
We now illustrate the procedure by calculating tooth dimensions of the input gear set.

Input set: Here both driving and driven gears are of the same 4.4" diameter so both will be identical. Pressure angle of 20^0 gives:

Base radius for gear and pinion

$$r_b = 4.4/2 \times \cos 20 = 2.07".$$

The standard measure of a spur gear tooth size is diametral pitch. As pitch diameter is fixed, its value depends only on selecting a particular number of teeth. We know that the more teeth a pinion has, the quieter it runs and the better is its resistance to wear because more teeth are in contact. A smaller number of teeth will, however, give increased tooth strength, lower cutting cost, and larger tooth dimensions. With this consideration, a



h = Working depth

H = Whole depth

C = Chordal thickness

Figure 3: Enlarged view of a gear tooth ----- P_d = Diametral pitch = 10

compromise of 44 teeth for both input gears is reached. All proportions are [1, P. 106]:

Diametral pitch $P_d = 44/4.4 = 10$,

Addendum $a = 0.8/P_d = 0.08"$

Dedendum $b = 1/P_d = 0.1"$

Working depth $= 1.6/P_d = 0.16"$

Whole depth $= 1.8/P_d = 0.18"$.

These are illustrated in Figure 3. As these gears are to be heavily loaded, it is considered desirable to determine the radius of curvature of root fillet r_f given by:

$r_f = r_T + (b - r_T)^2 / [d/2 + (b - r_T)]$ where:

r_T is the radius of the generating rack = 0.03",[5],

b is addendum = 0.08", and

d is pitch diameter of the gear concerned = 4.4".

Substituting:

$r_f = 0.03216"$.

Tooth thickness is a circumferential dimension based on $P_d = 8$, from Dudley [5], rather than on $P_d = 10$. From the geometry of a tooth, we get:

Tooth thickness $t = (p - B)/2 + \Delta_a\ (2 \tan \phi)$ where:

p = circular pitch $= \pi/P_d = \pi/8 = 0.393"$

B = backlash = difference in circumferential tooth thickness of meshing gears = 0.04", [5]

$\Delta_a = (\text{ a - working depth})/2 = 0.0$

$\phi = 20^0$.

So, $t = 0.389/2 = 0.1945"$.

As backlash varies depending on tooth cutting methods, etc., we propose that the hobbing method be employed for tooth cutting and so we keep 0.002" tolerance in tooth thickness. This thickness is generally measured by calipers so it gives corresponding chordal tooth thickness which is a little smaller measure than actual. Hence, according to Dudley [5], we apply 0.00015" correction.

B. Stresses

To determine exact stresses developed, we need to know how many tooth pairs are always in contact and thus share the load. For smooth continuous action, this should be more than one. However, we will consider the nearest whole number to be on the safe side. From [1, P. 118], total number of pairs in constant contact is given by:

$$x = 1/2\ \pi\cos\phi\ \left[\sqrt{(N+2c)^2 - (N\cos\phi)^2} + \sqrt{(n+2c)^2 - (n\cos\phi)^2} - (N+n)\sin\phi\right]$$

Here: ϕ = Pressure angle

N and n = number of teeth on gear and pinion, respectively = 44 for this input pair.

c = coefficient which is 0.8 for stub teeth.

Substituting:

$$x = 1/2\ \pi\cos 20\ \left[\sqrt{(44 + 1.6)^2 - (44\cos 20)^2} + \sqrt{(44 + 1.6)^2 - (44\cos 20)^2} - 88\sin 20\right] = 1.511\ .$$

Any gear is subjected to a bending stress and a compressive stress due to contact with mating gear tooth. Lewis, after extensive analysis considering each gear tooth as a cantilever beam with the load assumed to act at the point of contact, has derived a widely accepted result for bending stress. According to him, tensile stress due to bending:

$S_t = F/wpz$

Where: F = Tangential tooth load

p = Circular pitch, and

z = tooth form and number factor

z depends on the number of teeth and pitch of the gear concerned. From [1, P. 111], we find that for this pair, $z = 0.255$.

Also for maximum horsepower and torque of the engine,

$F = (250 \times 12)/2.2$

$= 1365$ lbs.

and,

$p = 0.314$

$w = 1.0"$

Substituting:

$S_t = 17{,}000$ psi.

The stress concentration factor in stub teeth varies from 2.05 to 2.18 in gears varying from 14 to 48 teeth. Thus, the maximum theoretical tensile bending stress in the weakest section (at the root of the tooth) is

$2.18 \times 17{,}000 = 37{,}000$ psi.

We now calculate the compressive contact stresses on gears. Prof. H. Hertz has developed a theory to calculate these contact stresses. It is accepted in fundamental procedure, but recent changes have enabled his equation to be reduced to a simpler form according to which:

Maximum compressive stress

$$S_c = 5715 \sqrt{K} \text{ psi.}$$

Here, factor K is given by:

$$K = (W/Fd) \times (m_G + 1)/m_G$$

where: W = tangential load

F = face width

d = pitch diameter of the gear concerned,

and m_G = gear ratio.

Substituting for this pair, K = 621. So,

$$S_c = 142{,}500 \text{ psi.}$$

C. Gear Sets

We now repeat this same sequence for other gear pairs and tabulate all different proportions as shown in Table 1.

Face width:

We have assumed the face width w to be 1.0" above. Exact values are calculated below, by considering the load capacity of each gear. Each gear tooth was considered to be a stationary beam and to compensate for tooth form factor and pitch line velocity, coefficient c is introduced in the equation [1]:

$$w = L \times P_d/c,$$

where L = tangential load. Coefficient c depends on the gear pair under consideration.

GEAR TRAIN	r_b BASE RADIUS		PITCH DIAMETERS		NO. OF TEETH		COMP. STRESS S_c	TENSILE STRESS S_t
	GEAR	PINION	GEAR	PINION	N	n		
Input	2.07"	2.04"	4.4"	4.4"	44	44	142,500 psi	17,000 psi
Low	3.1 "	1.03"	6.6"	2.2"	66	22	232,500 psi	75,250 psi
Second	2.72"	1.41"	5.8"	3.0"	58	30	182,000 psi	54,500 psi
Third	2.44"	1.69"	5.2"	3.6"	52	36	160,500 psi	45,350 psi
Reverse	2.91"	1.03"	6.2"	2.2"	62	22	300,000 psi	105,500 psi

N.B: Pitch diameter of reverse idler is 1.4" and it has 14 teeth.

Table I: Major gear proportions and stresses.

For low gear pear: $c = 30{,}000$

$L = 2730$ lbs.

So, $w = 0.91"$.

For second: $c = 21{,}500$

$L = 2000$ lbs.

So, $w = 0.93"$.

For third: $c = 17{,}000$

$L = 1662$ lbs.

So, $w = 0.985"$.

For input gears: $c = 13{,}000$

$L = 1370$ lbs.

So, $w = 1.0"$.

Thus for all gears, $w = 1.0"$ is a satisfactory value.

Material:

The stresses calculated are hardly true stresses, especially for contact stresses. This is because:

1. Stress concentration factor is indeterminately variable so only the maximum value is considered.
2. Distribution of tangential load over gear face width is unknown.
3. Failures such as scoring and pitting do depend on contact stress but much more so on the type of lubricant used. The lubricant, generally, contains an extreme pressure additive to protect against these failures.
4. We considered only one tooth pair in constant contact which is about 50 percent less than the actual.

5. Tensile and compressive stress are found greatest in low gears which are required to handle this full engine torque load only a small fraction of the operation period.

We still propose a material to withstand at least these stresses. We select S.A.E. steel no. 4815 which contains 8 percent Molybdenum as the chief ingredient and 15 percent carbon, or S.A.E. steel no. 2512 which is a nickel steel with 5 percent nickel and 12 percent carbon. The maximum permissible strength of these steels is 220,000 psi. in tension according to T. H. Wickenden's research [1, P. 115]. To determine the maximum permissible surface compressive stress or Hertzian contact stress, we use a formula [13, P. 422] according to which:

$\sigma H = (S_{fe} \times C_L \times C)_H / (C_T \times C_R)$ where:

S_{fe} is the surface endurance limit which equals 250,000 psi. for case carburized steel material for gears.

C_L is the life factor which equals 1.1 for a total life of 1,000,000 cycles.

C_H is the hardness ratio factor and from available information, it is suggested to be unity.

C_T is a temperature factor which also is unity for temperatures up to 250° F. And,

C_R is a reliability factor which equals 0.8 for 100 percent reliability requirements.

Substituting:

σH = 324,000 psi. which is more than actual contact stress calculated, so the material selected is safe.

V. CLAMPING DEVICE

As a constant mesh transmission is proposed, we need a separate clamping device for each pair of gears. For compactness, a multiple disc clutch proves the best choice but while selecting the exact type, we keep in mind that any manual sliding controls are to be avoided for an effortless operation. First consideration was that of using a conical or spring type clutch. In addition to them having to slide for engagement, the driven member in both of them carries a heavy moment of inertia so engagement is not smooth and quiet. Other possibilities considered for this were to provide a vacuum operated, pneumatic, centrifugal or magnetic clutch. They were rejected because they either failed to be entirely independent of engine operation or they proved to be too complex and expensive.

Compactness also calls for clutches to be located at the point of minimum torque. Except for input gear train, where location of the clutch makes no difference in the clutch size, the rest of the clutches should, therefore, be located on the counter shaft. Pitch diameters of the input gearset are identical. Therefore, all clutches transmit only the engine torque and are similar in design. This not only simplifies the design, but is also desirable for interchangeability.

An exploded view of the clutch assembly is shown in

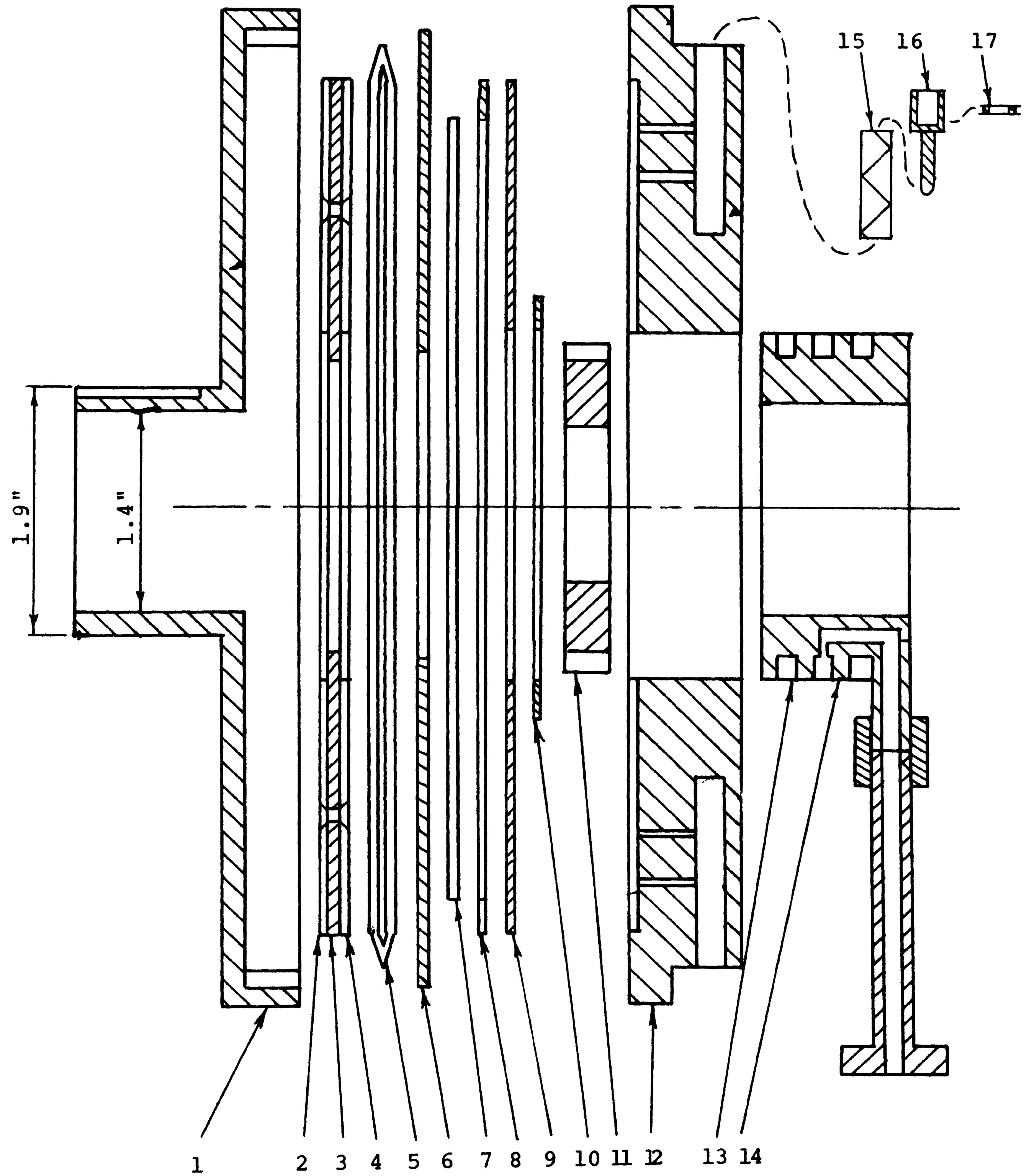


Figure 4: Exploded view of the clamping device.

Figure 4

KEY

Number	Part Name	Dimension
1	Drum	OD = 7.0", ID = 1.4"
2	Friction disc 1	OD = 6.0", ID = 2.5"
3	Driving plate	OD = 6.0", ID = 2.2"
4	Friction disc 2	10 pieces, 1/16" thick
5	Belleville washer	t = 3/32", h = 1/64"
6	Driven plate	OD = 6.5", ID = 2.3"
7	Insulating plate	1/16" thick of Asbestos
8	Outer ring	1/32" thick
9	Diaphragm	Active OD = 5.5", ID = 3.0"
10	Inner ring	1/32" thick
11	Hub	OD = 2.2"
12	Cover plate	-
13	Stationary boss seal	-
14	Stationary boss	OD = 1.5", ID = 1.5"
15	Drain valve spring	-
16	Drain valve piston	-
17	Valve piston seal	-

Figure 4. To engage the clutch, oil is pumped into a chamber which has one wall as a diaphragm to which is fixed a heat insulating plate. The diaphragm is secured to the cover plate at its outer and inner peripheries by two rings. The fluid pressure in the chamber extends the diaphragm and this stroke moves the plate, bringing into contact the friction facings with the driven plates. Torque is thus transmitted through the gear set to the output shaft of transmission.

We now determine the principal dimensions of the clamping device before considering each part in detail.

We know that torque to be transmitted by all clutches is the maximum engine torque which is 250 lb-ft. Several attempts of being compact finally give:

Outer diameter of friction facing = 6.0"

Inner diameter of friction facing = 2.5" .

As described a little later, the friction material with friction coefficient as 0.3, covers 300^0 arcwise or 0.813 of the plate area. This is done for a smooth "feel" of contacting plates and to avoid chatter. This therefore, gives:

$$\text{Friction surface area} = 0.813 \times \pi/4 \times (36 - 6.25)$$
$$= 19 \text{ in.}^2.$$

Selection of friction material limits the allowable unit pressure. To be well on the safe side, we select 120 psi. which is about 30 psi less than the maximum allowable pressure for the proposed material. This gives:

Net normal force per plate surface = P_n

$= 19 \times 120 = 2280$ lbs.

Hence:

Frictional force per plate surface $= P_n \times \mu = 685$ lbs. This force is distributed over the radial area. Average moment arm r_g is, therefore, selected to be the radius of gyration of the annular section around its axis of rotation given by:

$r_g = (1/3) \times (D^3-d^3)/(D^2-d^2)$ where D and d are OD and ID,

$= (1/3) \times (200.3/29.75) = 2.24"$.

If n surfaces are assumed to be driven, the torque capacity of the clutch:

$T_c = 685 \times 2.24 \times n = 1535$ n lb-in.

This has to be equal to the maximum engine torque. So,

$n = 250 \times 12/1535 = 1.995$.

The nearest whole number, n = 2, is preferred and thus, just one plate is needed. With n = 2, only 2200 lbs of axial force is found sufficient to transmit the required torque. The remainder of the axial force compresses the spring washers provided for disengagement.

A. Drum

The specified dimensions of the driving plates and also the geometry of the other parts, fix almost all dimensions of the drum. Wall thickness of boss and web are kept at 1/8". This and the geometry gives:

Outer diameter of the drum = 7.00".

Inner diameter of the drum = 1.4".

On the outer side is formed a rim in which teeth of driving plates slide. On the inner side is formed a boss on which gears are mounted. The boss has:

Inside diameter = 1.4"

Outside diameter = 1.65".

The minimum drum diameter is thus kept greater than the shaft diameter. This gap is necessary in order to let the respective drum and hence the gear mounted on it freely rotates when not engaged. To take up any possible rubbing wear, a sleeve bearing is provided in this gap.

We now check every section for a shear stress developed due to the torque transmitted. We will also design the splines in the drum rim in which driven plates will slide. Any section of the drum will have to resist engine torque of 250 lb-ft. which will pass through it unmultiplied. Due to this torque, shear stress developed is:

$S = T \times C/J$

where: T = Maximum engine torque x reserve factor which compensates for any decrease in clutch holding power.

C = Distance of the outer most fiber from center of rotation.

and J = Polar moment of inertia = $\pi/32 \times (D^4 - d^4)$ where D and d are outer and inner diameters of the part concerned.

Rim: Here,

$D = 7.0"$ and $d = 6.50"$;

so, $J = 63$ in."

$C = D/2 = 3.5"$

and $T = 250 \times 12 \times 1.25 = 3750$ lb-in.

The section where abrupt change takes place is weakest and there will be a concentration of stress there. With the stress concentration factor there as 1.5, we check that section [Moore, P. 67]:

$S = TC/J \times 1.5 = (3750 \times 3.5/63) \times 1.5 = 314$ psi.

Wall: Here,

$D = 7.0"$ and $d = 1.6"$;

so $J = \pi/32 \times (2390 - 6.5) = 234.4\ in^4$

$C = D/2 = 7/2 = 3.5"$

$T = 3750$ in-lb as in rim.

Again, checking the weakest section:

$S = (3750 \times 3.5/234.5) \times 1.5 = 84$ psi.

Boss: Here,

$D = 1.65"$ and $d = 1.4"$;

so, $J = \pi/32 \times (7.2 - 3.9) = 0.324\ in^4$

$C = D/2 = 1.65/2 = 0.825"$

$T = 3750$ in-lb. as in rim.

Again, checking the weakest section:

$S = (3750 \times 0.825/0.324) \times 1.5 = 14{,}300$ psi.

The drum material is selected to be grey cast iron whose shear strength, 36,000 psi. to 60,000 psi. |13, P. 605|, is above all these stresses calculated, so the drum is quite safe.

Splines: Teeth in the driven plates will slide in these splines in the rim. Thickness of the driven plates is proposed to be 1/16" so we propose teeth of the height and width, 1/8" and thickness 1/16". The area of 1/16 x 1/8 = 0.078 in^2 for each spline shall carry all the bearing load. We propose eight splines so:

$$S_t = (250 \times 12/3.25) \times [1.5/(78 + 8)] = 18{,}500 \text{ psi.}$$

This is below the stress permitted in the drum material, so it is safe. We can also check the bending stress on the splines. As there are eight splines, a load, $F = (3000/18 \times 3.25)$ = 115 lbs. acts on each spline. This load will create a moment $M = 115 \times 1/16 = 7.18$ lb-in. The spline section (1/16") x (1/8") has $I = (1/12) \times (1/16) \times (1/8)^3$, so bending stress $\sigma_t = Mc/I$ where c = maximum distance from the point of action of the load = 1/16". Substituting, $\sigma_t = 44{,}000$ psi. which is also within the limit.

The axial fluid force of 2280 lbs. tries to deform the drum on one side and the coverplate on the other. We can check the proposed wall thickness of 1/8" against this deformation. For it, we consider the drum wall as a hollow circular plate of OD = 6.5" and ID = 1.65" with uniformly distributed load, q = 120 psi. Since the rim and boss are proposed to have double the wall thickness and since no relative radial motion is possible, it is assumed that both inner and outer edges are clamped. The equation for deflection is:

$$w_{max} = k_1 \times q \times a^4/E \times h^3,$$

where k_1 = coefficient which for the assumed end condition

= 0.179 [13, P. 328].

a = OD/2 = 3.25"

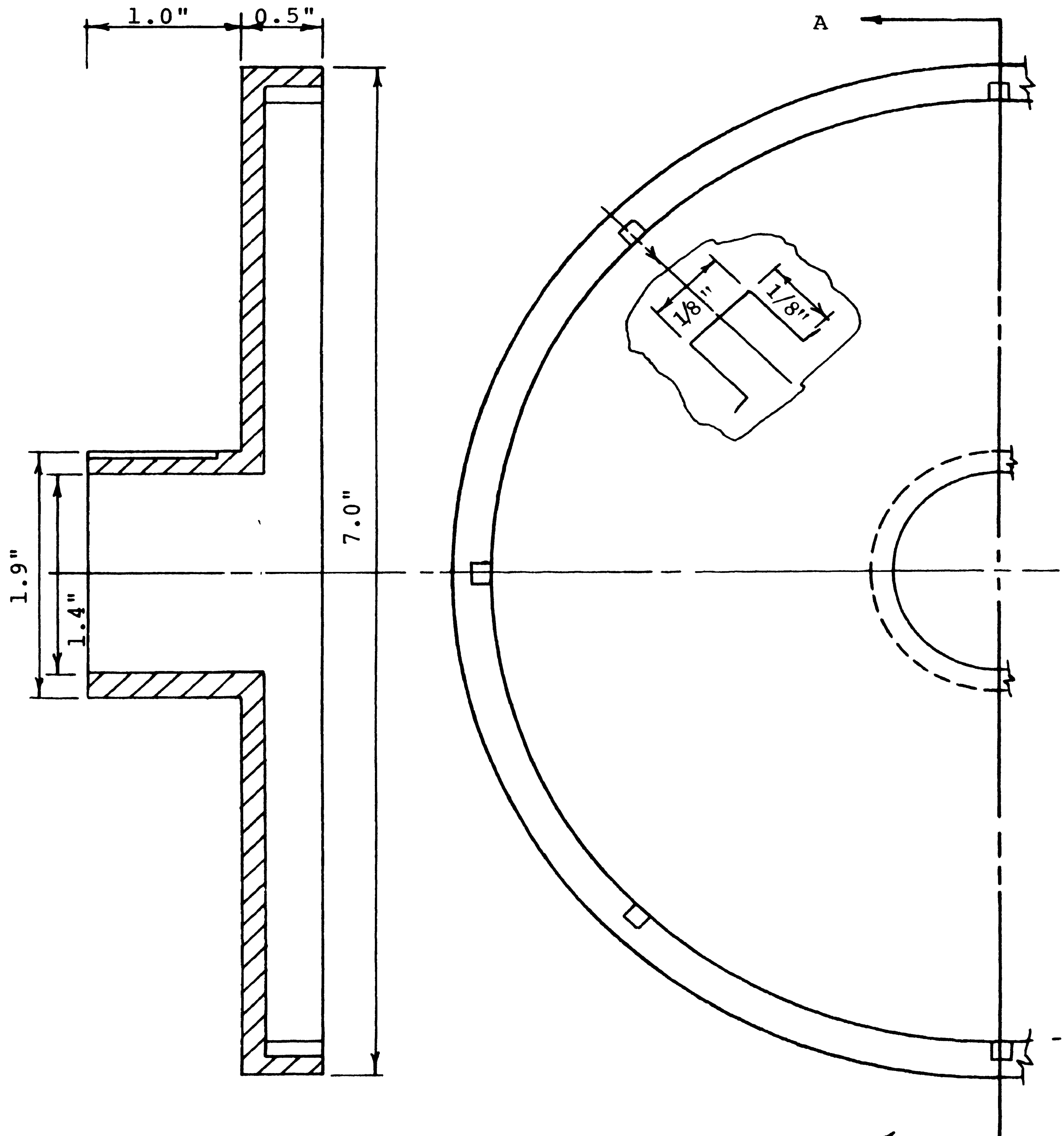


Figure 5: Drum for the clamping device.

E = modulus of elasticity = 28.5 x 10^6 psi.

and h = plate thickness = 1/8".

Substituting:

ω_{max} = 0.0425" which should be satisfactory.*

The drum is to be bolted to the cover plate. We propose six bolts of 1/8" diameter which have blind ends in a 1/4" thick drum rim. The tensile load on these bolts is 2280 lbs. and the resisting area is 0.295 in^2. So, the tensile stress developed is:

S_t = 2280/0.294 = 7,725 psi. which is safe.

B. Plates

1. Friction plates are formed by rivetting separate pieces of friction material on the driving plates.** As mentioned earlier, to avoid chatter and for a smoother "feel" of the contact plate, friction plates are formed of segments of material covering 300^0 of the plate surface. To facilitate the replacement of parts in case of wear, we propose ten pieces of 30^0 arc width with 6^0 gap in between. The thickness of each plate is selected arbitrarily as 1/16". The friction material is selected to be powdered copper asbestos compound and as the clutch is to operate in the dry condition, its coefficient of friction is to

---

*Assuming the extreme possibility of simply supported ends, k_1 = 0.492 [Marks Handbook, P. 5-69] and ω_{max} = 0.117". The actual deflection will probably lie between these values and a thicker section may be required.

**Alternate method: bond friction linings to the driving plates.

be 0.3, [1]. This material is selected as allowable contact pressure in it can go up to 150 psi. These segments are rivetted to the plate so the next step is to determine the rivet diameter.

To prevent any twisting of the segments on driving plates, we propose two rivets, ½" from the top and bottom of each segment. We fasten two segments on both sides of the driving plate. For 20 segments, therefore, we need to provide 20 rivets. By equating shearing and crushing strength and substituting values of bearing and rubbing failure in single shear, rivet diameter d = 2.28 nt for steel rivets where:

n = number of rivets, and

t = plate thickness = 1/16".

So, d = 2.28 x 2 x 1/16 = 0.071".

We keep a standard 3/32" diameter of rivets. As driven plates have to press on friction surfaces, we propose counter sunk heads 3/16" in diameter.

2. Driven plate material is to be open hearth steel with:

tensile strength = 89,000 psi. and

shear strength = 60,000 psi.

Since these plates slide in the drum rim, eight teeth of size 1/16" thickness, 1/8" width and 1/8" height are provided to slide in corresponding grooves in the drum.

3. Driving plates: Material is the same as for the driven plates. Again spline teeth dimensions are to

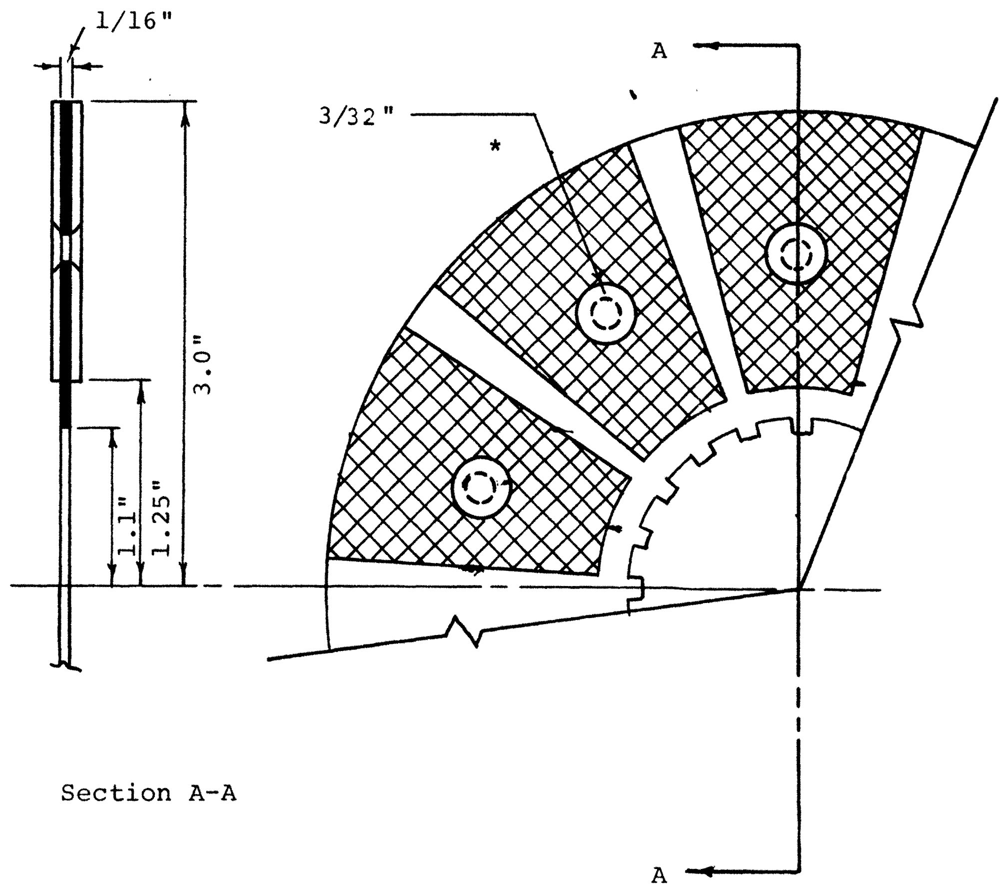


Figure 6: Friction plate and driving plate assembly

*Two 3/32" rivets per segment are required.

be 1/16" thickness, x 1/8" height, x 1/8" width. The number of teeth depends upon the weaker hub material which is proposed to be grea cast iron. We propose sixteen such teeth and calculate the shear stress developed due to torque:

Shear stress x area x mean radius x 16 = torque transmitted.

$S \times (1/16 \times 1/8) \times 2.2/2 \times 16 = 3750$

So, S = 27,250 psi. which is safe.

4. Hub: Material is again grey cast iron. The purpose of this hub is to connect the axially sliding driven disc to the shaft. Hence, the tentative inner diameter is to be 1.2". As the friction facing has an inside diameter of 2.5", we need to keep the outside diameter of this hub as 2.3" to accomodate the teeth of the driving plate. Since the number of teeth in the driving disc is calculated to be sixteen, there will also be sixteen splines of the size 1/16" x 1/8" x 1/8" on this outside diameter. This might weaken the section so we assume outer diameter as being 2.2" for checking shear stresses. This gives:

$J = \pi/32 \times (d^4 - d^4)$

$= \pi/32 \times (22.9 - 2.06) = 2.04 \text{ in}^4$

$C = 1.1"$

$T = 3750 \times$ stress concentration factor (1.5)

So, $S = 3750 \times 1.5 \times 1.1/2.04 = 3080$ psi which is permissible.

C. Disengagement Springs

Severe space limitations exist in this area. The deflections have to be just 1/64" per plate. Moreover, friction surfaces on both sides of the driving plate have to be disengaged. After several tries on helical compression springs, the decision was made to use Belleville spring washers which are compact and also simpler to mount.

Material: High carbon steel strip, AISI 1075, is selected with:

Modulus of elasticity	$E = 30 \times 10^6$ psi,
Modulus of rigidity	$G = 11.5 \times 10^6$ psi,
Density	$\rho = 0.284$ lbs/in^3.

This material also has good fatigue resistance, high relative strength and can work efficiently even at 250^0F temperature. Driven plates on both sides of the driving plates are each 1/64" apart. Thus, when in contact, total deflection of the washer has to be 1/32" due to force of the fluid and its reaction in the opposite directions. We have a calculated axial force due to fluid pressure as 80 lbs. on the washer. To allow for manufacturing variations, we assume:

$$P_1 = K \times P_{min}.$$

where K is a factor by which the axial force is increased |6|. Here, $K = 1.1$ and P_{min} = the minimum axial force = 80 lbs, so

$$P_1 = 1.1 \times 80 = 88 \text{ lbs}.$$

We also have:

outside diameter = 6.5"

and inside diameter = 6.0" for each washer.

These give:

R = outside diameter/inside diameter

= 6.5/6.0 = 1.083.

We propose the washer to be a very loose fit between two rotating parts. So, we keep 0.06" tolerances on both diameters and hence:

outside diameter = 6.44"

and inside diameter = 6.06".

This now gives:

$R = 6.44/6.06 = 1.06.$

Since the deflections have to be just 1/64" or 0.0156" per plate for an axial load of 88 lbs, we select 1/64" thick washers. We also know that:

$2(h + t) = 7/32"$

so we get,

$h = 3/32"$.

We know that the washer is in the form of a hollow frustum of a cone. The maximum compressive stress generated by the fluid pressure is at the upper inner edge of it and is given by:

$$S_c = [Ef/(1-v^2)Ma^2] \times [c_1(h-f/2) + c_2t]$$

Here,

v = Poisson's ratio = 0.3,

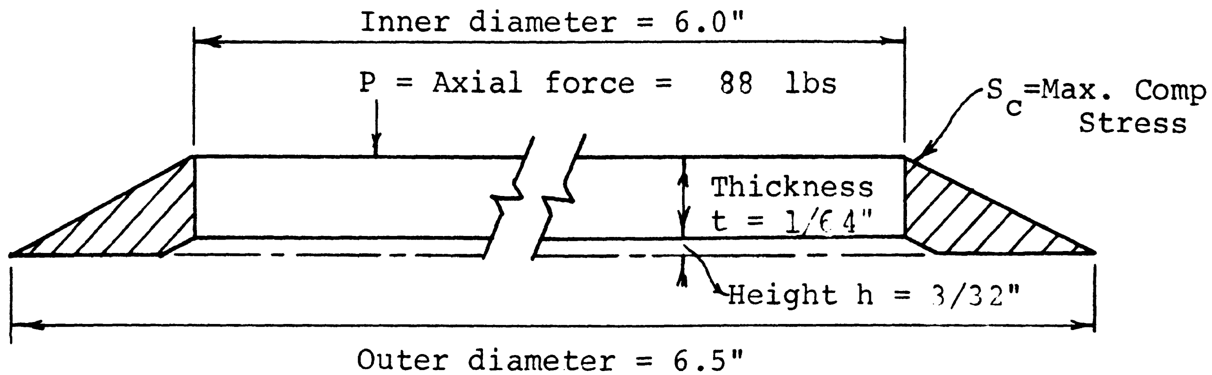


Figure 7: Belleville spring washer (end view).

c_1, c_2, M and a are constants calculated as follows:

$$c_1 = [6/(\pi \ln R)] \times [(R-1)/\ln R - 1]$$

$$= (6/\pi \times 0.05827) \times [(0.06)/(0.05827 - 1)]$$

$$= 0.60.$$

$$c_2 = [t/(\pi \ln R)] \times [(r-1)/2]$$

$$= (6/\pi 0.05827) \times (0.06/2)$$

$$= 0.60.$$

$$M = (6/\pi \ln R) \times [(r - 1)/R]^2$$

$$= (6/\pi \times 0.05827) \times (0.06/1.06)^2$$

$$= 0.0638.$$

$$a = 6.5/2 = 3.25$$

So, $a^2 = 10.5.$

Substituting:

$$S_c = (30 \times 10^6/64 \times 0.91 \times 0.0638 \times 10.5)$$

$$\times [(0.6 \times 11/128) + 0.6/64]$$

$$= (0.0768 \times 10^6) \times (0.0609)$$

$$= 4700 \text{ psi.}$$

From Table 3, [6, P. 48], we get the design stress of the washer material for a minimum life of 10^4 cycles in torsion to be about 45 percent of its tensile strength. If hardness of washer material is taken as Rockwell 48, from Figure 24, [6, P. 42], we find:

Tensile strength = 240,000 psi.

So,

design stress = 0.45 x 240,000

= 108,000 psi.

We see that this is far greater than the value calculated, so our assumptions are proved satisfactory.

D. Diaphragm

The purpose of the diaphragm is to transmit the hydraulic force into the form of motion to the plates. It is generally made of a flexible material, usually a fabric impregnated with an elastomer. The stroke required is 1/32" maximum. Our geometry fixes the outer diaphragm = 5.5" and the inner diameter = 3.0" for the diaphragm. So,

$$\text{diaphragm area} = \pi/4 \times (30.2 - 9.0) = 16.5 \text{ in}^2.$$

For it to exert a contact pressure of 120 psi:

$$\text{Total hydraulic pressure} = 120 \times 19.0/16.5$$
$$= 138 \text{ psi}.$$

As the outer diameter of the diaphragm is large enough, we propose a completely flat diaphragm instead of a convoluted one. This makes it impossible to avoid undesirable stretching of the diaphragm under fluid pressure. The fabric might be excessively stretched in addition to being flexed. We can be ensured against this stretching hazard by calculating the stroke as a percentage of the outside diameter. This should be less than 7 percent to 9 percent, [7]. Here it is,

$$100/(5.5 \times 32) = 0.57\%,$$

so a flat diaphragm is satisfactory. The fabric here can be selected as being a nylon fabric. It has a good resilience and a high tensile strength. As the stroke length is just 1/32", we propose the diaphragm thickness as being 1/32".

This diaphragm is connected to the cover plate at the outer and inner peripheries. This is done by letting the portions of the diaphragm there work as gaskets between the cover plate and two other supporting rings specially provided for that purpose and screwing all three together.

At these contact surfaces, the diaphragm has to work as a tight seal. This requires that screw spacing be about 3.5 times the screw diameter. With space limitations in mind, we propose a screw diameter to be 1/4". This makes screw spacing = 3.5d = 3.5/4 = 0.875". This criterion causes us to select 20 screws on the outer ring and 10 screws on the inner ring. There is a total tensile load on these screws of 2280 lbs. which will be shared by 30 threaded screws developing tensile stress:

$T = 2280 \times 4/(\pi \times 30/16) = 1550$ psi. so it is safe. Since outer and inner rings act only as flanges to confine the diaphragm, we can empirically keep them 1/16" thick and they can be made of SAE 1040 steel.

When plates are engaged to the friction surfaces due to friction, heat will be generated. To keep this heat away from the diaphragm, a heat insulating plate, 1/16" thick to maintain uniformity, is welded to the driven plate.

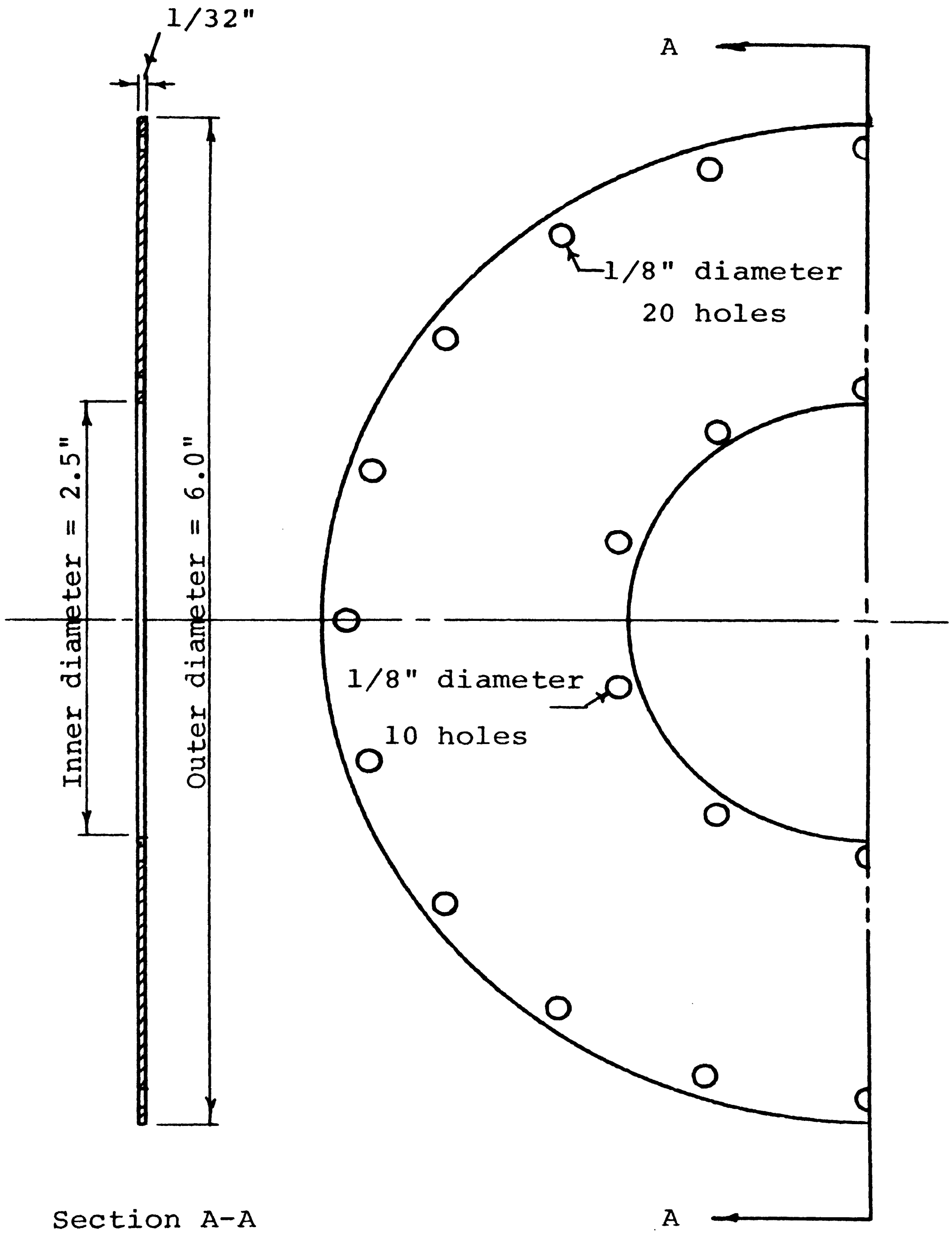


Figure 8: Diaphragm for the clamping device.

This plate is of asbestos, the best heat insulating material. We now calculate the maximum amount of heat that can be generated. This and the temperature rise of the plates after engagement were derived from an energy consideration, [13, P. 509]. We find out the relative velocity between driving and driven plates just before engagement. We assume that the speed the engine when developing maximum torque, and, hence, that of the driving plate = 2800 rpm. The minimum speed of the driven plate in other clutches is when low gear clutch is engaged. It is 1900 rpm considering a shift from low to second gear. When there is a uniform wear, we can calculate the linear velocity in ft/sec. at a mean radius (i.e. at 1.5" from the center line). For the driving plate:

$$V = (2\pi\ 1.5/12) \times (2800/60) = 36.6 \text{ ft/sec.}$$

For the driven plate:

$$V = (2\pi\ 1.5/12) \times (1900/60) = 24.8 \text{ ft/sec.}$$

Hence, the maximum relative velocity = 36.6 - 24.8 = 11.8 ft/sec. After the clutch is engaged, it must be zero. This means that the driven plate has to be accelerated. We have proposed this to happen in 0.4 seconds so the angular acceleration of the driven plate has to be:

$$2\pi n/60t$$

where t = engagement time in seconds and n in rpm.

Here:

$$\text{angular acceleration} = 236 \text{ rad./sec.}^2$$

with n, rpm for 11.8 ft/sec. velocity $= 11.8 \times 12 \times 60/3\pi$

$= 900.$

Now, according to Newton's law, maximum torque, T, required is:

$T = \pi \times$ Angular acceleration.

Here, the mass moment of inertia $J = (1/2)\ Mr^2$. To account for the effect of engine inertia during clutch engagement, we assume the total J to be equal to that of the steel disc, 2" thick and 15" in diameter. With its density,

$\rho = 0.284\ lbs/in^3$,

Mass of the disc $M = W/g = [(\pi/4)(15)^2(2)(0.285)]/(32.2)$

$= 1.1$ slug.

$J = 1/2 \times (1.1) \times (15/2)^2 \times 1/144 = 0.194$ slug-ft^2

So, $T = 1240 - 194 \times 236.0 = 550$ lbs-ft.

We now see that the energy required for a single clutching operation is:

$E_k = \pi^2 n^2 J/1800 = 853$ ft-lbs.

This much energy gets converted into heat energy, or heat generated in BTU is:

$H = 853/778 = 1.1$ BTU.

The temperature of the clutch plates at the end of a single clutching operation is:

$t_2 = H/cw + t_1$

where: t_1 = temperature before engagement = say 180^0F,

c = specific heat of the clutch plate material

= 0.12 (average value),

w = weight of the clutch plate

$= 1/16 \times \pi/4 \times 19.75 \times$ density $= 1.45 \times$ density.

If density is $0.284\ lbs/in^3$, substituting, $t_2 = 203^0$ F.

This will be only momentarily true and then if the heat generated is assumed not to be dissipated at all, it will also raise the temperature of the contacting plate till all of the plates reach an equilibrium temperature. We can now calculate that temperature from Gagne's formula, |13, P. 509|:

$$c(t_{ave} - t_1) \times [Nt/3600 + 1.5\ (1 - Nt/3600)] = HN/A$$

where: c = heat transfer coefficient = 2.5 for rough black surface of linear velocity of about 30 ft/sec.

t_{ave} = average equilibrium temperature

t_1 = initial temperature = say 100°F.

t = time of single clutching operations per hour = 10.

Substituting, t_{ave} = 182°F.

This is much below that of the lining material, copper asbestos compound, so it is safe. The heat generated is also gradually dissipated to the circulating fluid so 663° F. average temperature is only theoretically reached.

E. Combination Supply and Drain Valve

The 138 psi. hydraulic pressure calculated above can come to the diaphragm chamber through combination supply and drain valves. Two of them are provided to speed the draining and also to reduce the inlet fluid velocity during disengaging and engaging respectively. The pressure acts on pistons within these valves. Before designing the pistons, springs, etc. in detail, overall valve operation will be described.

When fluid acts on the piston, its force tries to push the piston down. This causes the spring below the piston (which stands on a stem integral with the piston body) to deflect a certain amount depending upon its spring index. When the spring deflects, the piston moves down. In moving down, the hole in the piston body also moves down and connects the inside of the piston body (and hence the inlet from the pump) to the diaphragm chamber through a port V as shown in Figure 4. At the same time, the piston body also covers the other port T, from the diaphragm chamber. The fluid which comes in to the diaphragm chamber has no way out to relieve the pressure, so it pressurizes the diaphragm, causes it to extend, which in turn moves the plates and thus engages the clutch.

As long as there is a pressure in the fluid, the spring remains deflected, port T remains covered and the clutch remains engaged. When the driver selects to deenergize the solenoid valve (section ix), the piston moves up, thereby uncovering the passage from the chamber to the transmission box. Through this passage, the pressure is relieved and fluid is drained. The Belleville washers assume their function: they disengage the plates, move the diaphragm to its original position and complete the disengagement operation.

1. Valve piston: As the main function of the valve is to control the drain port, we propose a solid piston located between two ports. Comparing it to standards of hollow pipes [6], we decide to keep the bore diameter of the

valve housing as 0.225" and, hence, the area on which the fluid pressure acts is given by:

$$A = \pi/4 \times (0.225)^2 = 0.0395 \text{ in}^2.$$

The axial force is, therefore, equal to:

$$0.0395 \times 138 = 5.45 \text{ lbs.}$$

2. Valve spring: The mean diameter of the spring is kept at 0.2", reaching a compromise between it and the bore diameter. Since buckling of the spring is not desired, we have to select a very low spring index. It, however, should not be low enough to cause coil interference. We select 0.05" wire diameter which gives the spring index:

$$C = 0.2/0.05 = 4.0.$$

From Dr. Wahl's formula, we can now check this spring for shear stress developed. It should be noted that the force of 5.45 lbs. will be distributed along the whole coil cross-section. However, to be on the safe side, it is assumed that the load acts at the center of the coil cross-section. This then gives:

$$\text{Moment, } M_t = 5.45 \times 0.2/2 = 0.545 \text{ lb-in.}$$

Dr. Wahl's formula is:

$$S_s = 16M_t/\pi d^3 \left[(4C - 1)/(4C - 4)\right] = 28{,}000 \text{ psi.}$$

For a spring, this stress is quite low, so our dimensions are justified.

Material: After considering fatigue strength, corrosion resistance, service temperature and cost, our choice is high carbon steel wire, ASTM A230, which is available in 0.05" diameter wire size, has excellent fatique resistence, can work under 300^0 F. temperature, and has high relative strength:

$E = 30 \times 10^6$ psi.

$G = 11.5 \times 10^6$ psi. and

density = 0.284 lbs/in^3.

To check this 28,000 psi. shear stress developed, we see from Table 3, [6, P. 46] that yield point stress is 45 percent of the tensile strength. From Figure 24, [6, P.42], we find the tensile strength to be 240,000 psi. assuming a hardness of RC 48. Hence, yield point stress without residual stresses = 108,000 psi. Again from Table 5 [6, P. 46] we see that fatigue due to cyclic service reduces this stress. Using the figure for maximum 10^7 cycles, we see that only 30 percent of the yield point stress should be taken as a design stress (i.e. maximum permissible shear stress is 32,400 psi. which is more than the actual stress developed.)

We can now calculate some miscellaneous details of the spring and its deflection to locate valve ports. From geometry, we select 1.3" as the free length of the spring and following common practice, we keep the coil pitch angle of the spring as 10^0.

If x = vertical distance between two consecutive coils and D = mean coil diameter, then:

$x = D \sin 10 = 0.3472.$

So, the number of active coils:

$$n = 1.3/(0.03472\ n + 0.05\ n)$$

which gives:

$$n = 3.93 \text{ or say } 4.0.$$

We propose the ends squared and ground [6, P. 5], so we see that the total number of coils should be kept as

$$4 + 2 = 6.$$

To locate the ports, we now calculate the axial deflection δ. For that we use the formula:

$$\delta = [(p)(n)/(G)(d)] \times (8C^3 + 4.92\ C)$$

where: P = axial load

C = spring index,

and d = wire diameter.

Substituting:

$$= 0.0203''.$$

This determines the location of the drain port below the piston body. The bore is closed by a cap screwed into it.

3. Piston seal: The seal has to prevent leakage, if any, from the small clearance between the bore and the piston diameter. The reciprocating motion of the piston and, hence, the seal is too small to cause any fatique problem. We propose an O-ring seal of 001 size which has a cross-section of 0.04" diameter. This makes groove length = 0.052" [7, P. 73] and diametral squeeze of 0.0045". This is selected as it has:

1. Low initial cost,

2. Adaptability to limited,
3. High efficiency,
4. Tolerance to wide ranges of pressure, temperature and fluids,

and 5. Relatively low friction.

Material: This ring is made of the synthetic rubber, Hexafluoropropylene and vinylidene Fluoride Copolymers. These materials show excellent resistance to a great number of fluids and chemicals. Their service temperature range varies from - 40^0 F. to 450^0 F.

F. Stationary Boss

It is through this boss that the pressurized transmission fluid from the stationary pump is passed to the rotating drain valve housing integral with the cover plate. To achieve this, we need to subdivide the whole problem in the following parts and then consider them one by one.

1. An arrangement to locate and keep the boss stationary.
2. An arrangement to provide some device to prevent leakage from the passage between the control valve and the boss and also from the passage between the boss and cover plate.

For the first requirement, a possibility is that of providing supporting webs bolted to the fixed transmission box. However, our geometry tempts us to locate the controlling valves at the bottom of the transmission box for easy accessibility. The pipe which can connect each valve to its respective boss can very conveniently be arranged to transmit the fluid as well as to act as support for the boss. All but one of the clutches are located on the counter shaft, so we need only 3" long pipes to connect the bosses to the bottom of the box. These pipes can have flanges at the bottom end at which they can be bolted to the lower transmission box wall, and hence, be connected to the appropriate valves. At the top end, the pipe is to be connected to the boss. This is done by providing a swivel nut which can fasten both pipe end and the extension from the boss together. This is provided to facilitate manufacturing of the boss.

To determine the dimensions of these connections, we need first to determine the inner and outer diameters of the pipe connection to the boss. The first thing that immediately comes to attention is the question as to how much fluid is to flow and with what velocity. This requires us to calculate the space behind the diaphragm when it is on full stroke. We know that the outer diameter of the diaphragm is 5.5", the inner diameter is 3.0" and the total stroke is 1/32". When the diaphragm extends, it forms a paraboloid. However, for safety, we shall assume that a parallelopiped of 1/32" height is formed. We also assume that the axial thickness of the diaphragm chamber is 1/16" so we get the volume behind the diaphragm as:

$$V = \pi/4 \left[(5.5)^2 - (3)^2\right] \times 3/32 = 1.55 \text{ in}^3.$$

In addition to this, fluid from the pipe will also drain off. Presently, we keep a tentative inside diameter of the pipe piece as 1/8" which gives the volume of fluid in that piece as $\pi/4 \times (1/64) \times 2.0 = 0.0246$ in^3. This makes the total space to be filled in as 1.5746, or approximately 1.58 in^3. Actually, it will be slightly less than this as the whole volume will not be emptied after disengagement. This will compensate for the space above the piston of the drain valve, which is not calculated.

We propose that this much liquid has to be filled in 0.4 seconds. This requires that the pump supply capacity be at least:

$$(1.58 \times 60)/(0.4 \times 231) = 1.025 \text{ G.P.M.}$$

Now we can calculate the flow velocity in this pipe. Pippenger [10], informs that the flow velocity is related to the pipe diameter and flow volume. It is:

$$Q = 2.44 \times d^2 \times v$$

where Q is in GPM, v in ft/sec. and d is in inches. This, then gives:

$$v = 1.025 \times 64/2.44$$

$$= 26.8 \text{ ft/sec.}$$

Too high a velocity can result in an excessive pressure drop in the system so we determine if v = 26.8 ft/sec. is permissible. It is a matter of experience to use suitable velocity in hydraulic practice. Ernst [9] has, however, noticed several satisfactory applications and has recommended certain velocities. Accordingly, a velocity

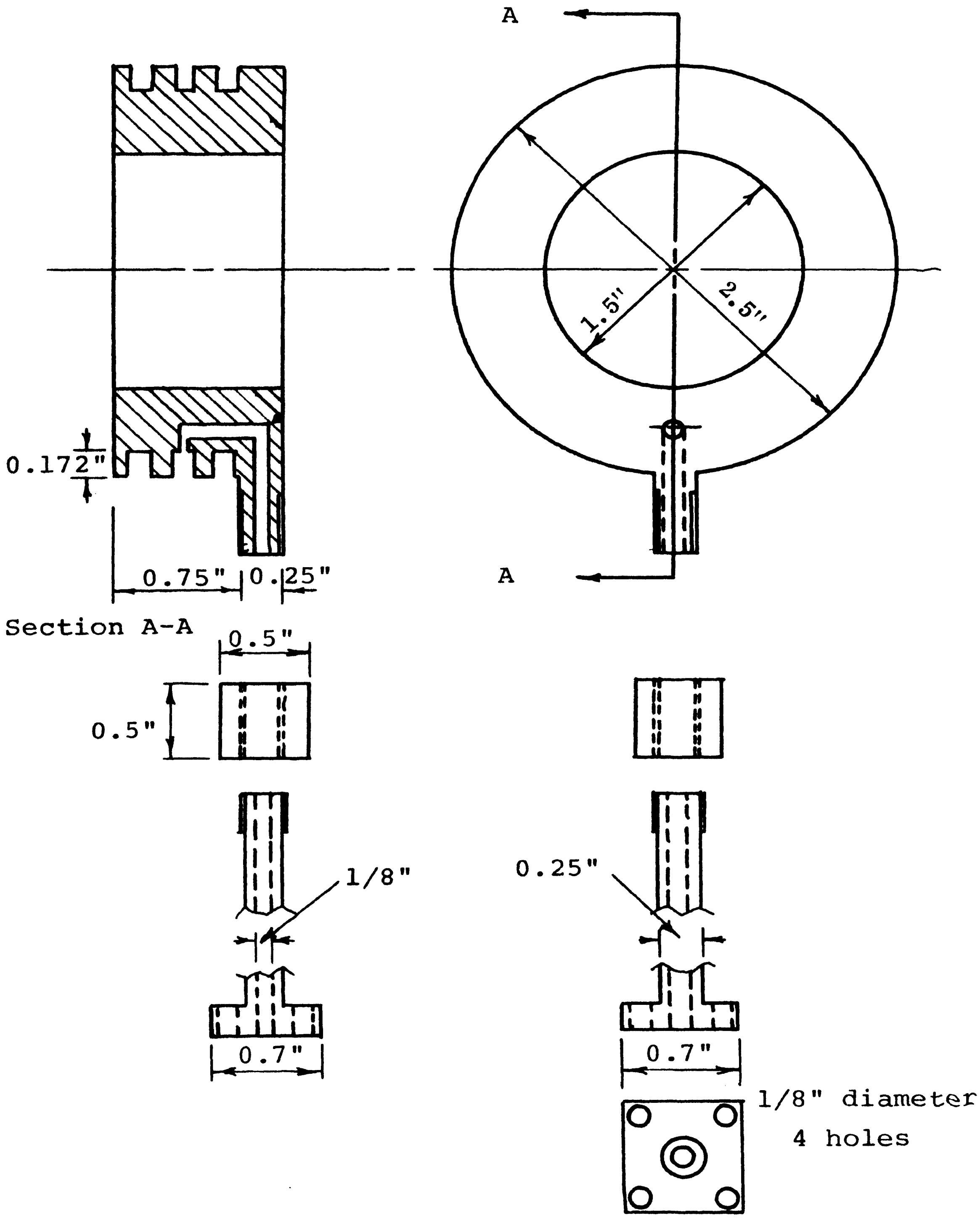


Figure 9 : Stationary boss and its flange.

of 25-30 ft./sec. through the control valves and other short restrictions is quite satisfactory. So, we can now specify the inner diameter of the pipe as being 1/8".

For determining the thickness of the pipe, we use Barlow's formula given by Mr. Holzbock. It is:

$t = PD/2S$

where P is pressure and d is the outer diameter which has to be tentatively taken as 3/8". We again select pipe material identical to the boss material, viz. SAE 1010, a low carbon steel which has a tensile strength, S, of 55,000 psi. The JIC (Joint Industry Conference) has established certain hydraulic standards and it suggests that we should at least have a factor of safety of 8:1 for such joints. We, thus, have:

$t = (138 \times 3 \times 8)/(8 \times 2 \times 55{,}000) = 0.00374"$.

This is to our advantage and is much less than our tentative assumption. If the wall thickness is kept at 1/16", then it is just a little more than 1/32". However, since the piece has to act also as a web, it is considered safe to keep 1/16" thick pipe.

Our geometry and other restrictions require us to connect this pipe to the bottom wall and, hence, to the control valve by a flange. Ernst has very rightly pointed out that no generally adopted standard is available for such flanges. Even ASME - Piping Standards Book, fails to give any standard for this small a pipe. Therefore, by comparing to similar drawings, we arbitrarily determine the dimensions of a four bolt flange. The dimensions are shown in Figure 9. It should be noted that the four bolts of the flange fit into blind holes which end at about half way in the transmission box wall.

As pressurized fluid has to pass through this pipe, we need to keep one gasket here to prevent any leakage. After studying various seals, we finally selected the Mark-II seal, described in [8]. It will be a one-sided, double crown seal. It is described as having super reliability, and it completely prevents leakage. As noted in the diagram, the inner diameter of this seal groove is to be only 0.45". This is under 6" so 0.172" can be adopted as the optimum recommended groove width with ±0.005" as tolerance. The edge distance can be kept 0.05" as bolts are to be only 0.125" in diameter. The thickness of the flange is also recommended to be 0.1". It is also recommended that the bolt torque be just sufficient to compress the gasket by 25 to 30 pounds per lineal inch. While selecting the material for the seal, we have to keep in mind corrosion, temperature resistance, and also its compatibility with transmission fluid. Fluoro-Silicon material seems to be most compatible and cheapest so it is selected.

G. Seals

Pressurized fluid which comes from the pump is to be transmitted to the diaphragm chamber through a cover plate that rotates on the boss. Thus, at the point in the passage where the fluid passes from the stationary to the rotating part, precautions against leakage through the clearance (which has to be of the order of 0.001) must be taken. This requires installing two seals on both sides fo the passage. The following points require consideration:

1. Installation and removal of the sealing device should be easy.
2. The wear of the sealing material due to rubbing should be minimized.
3. Provision should be made to dissipate heat generated at the rubbing surface.

The outer diameter of the stationary boss is a fixed parameter, viz 2.5". The center line of each seal is proposed to be at 0.15" on opposite sides of the center line of the passage.

For ease in mounting, we propose cut rings. This will make it a floating type of seal which can also minimize the rubbing wear due to adjustment by floating. The split ring can, however, create the possibility of gap leakage. For it we compromise by selecting a three piece ring. To keep the rings stationary, they should be loosely locked to the stationary boss. From [8, P. 245], we can adopt the groove width and depth to be 0.172".

We can now consider leakage prevention in detail. For this type of seal, there are two possibilities for leakage.

Primary leakage can take place at the surface where relative rotation takes place. But when the fluid pressure acts on the inner surface of the seal, it sets the seal tight across the clearance and, thus, prevents primary leakage. The secondary leakage path exists at the radial surface of the groove in the boss, against which the seal ring is seated axially. The force due to fluid pressure compresses the seal to this seat and prevents secondary leakage. These axial and radial forces, however, should be balanced so that there is no frictional "hanging" taking place and no extra wear. These sealing forces should overcome not only the friction of the respective side of the sealing surfaces, but must also be large enough to move the mass of the ring fast enough to be effective. If b and d are breadth and depth of the overall seal and μ, the coefficient of friction, then:

$$F_r = \rho bd \times a_r + \mu F_a$$

and axially:

$$F_a = \rho bd \times a_a + \mu F_r$$

where a_r and a_a are radial and axial accelerations. To minimize leakage, F_r should be equal to F_a, which is true since here:

$$b = d = 0.172".$$

From these equations, it is also evident that a material with lower density would be more responsive to fluid pressure.

We also need to consider any wear due to surface contact. To minimize wear, we have to select the most

Ring A:

0.086" x 0.086"

OD = 2.328"

ID = 2.156"

Ring B:

0.086" x 0.086"

OD = 2.5"

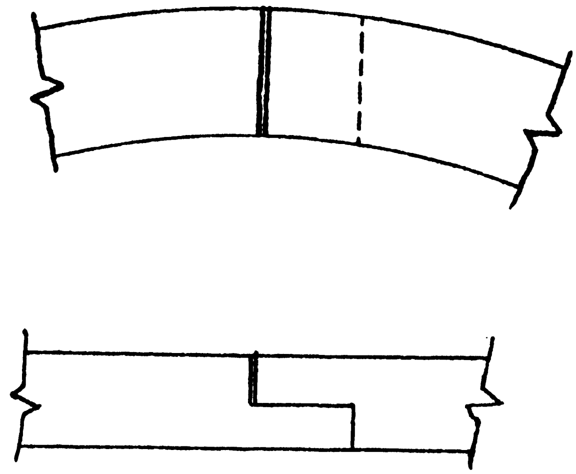
ID = 2.238"

Ring C:

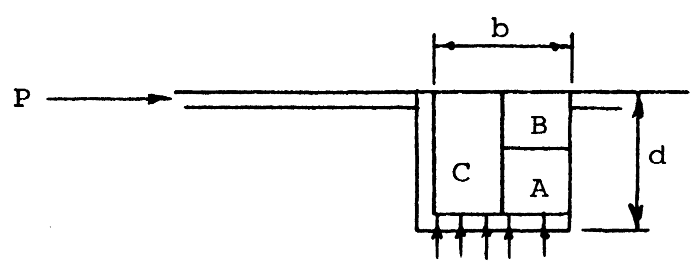
0.172" x 0.086"

OD = 2.5"

ID = 2.156"



Enlarged view of a step joint



Cross-sectional view when assembled

Figure 10: Multiring seal for stationary boss.

suitable material. Carbon graphite seems to be a suitable material. The transmission fluid which acts as a lubricating fluid, lubricates the seal at the rubbing surface. Because of this, the friction coefficient cannot be more than 0.04 so very negligible wear takes place. Any wear can also be compensated for by the fluid pressure at the inner diameter of the seal which expands the ring. This expansion can widen gaps, but as there are three rings, each gap will be overlapped by the solid portion of a ring. The coefficient of thermal expansion is as low as 0.15×10^{-6} in/in/^{0}F, so it takes care of the wear that might occur from eccentricity due to the heat generated.

The heat generated due to rubbing is at the surface of the cover plate (which is in constant contact with circulating transmission fluid on the outside), so heat conduction is also fast.

Locations of these rings are shown in Figure 10.

VI. SHAFTS

The main shaft will rest in a spigot bearing in the input shaft on one side and in a bearing in the housing on the other side. The spigot bearing will permit relative rotary motion between main and input shafts. Its reaction will, however, be transmitted to a stationary member through the input shaft bearing in the housing. Thus, for calculation of bending moments, the main shaft is assumed continuous. The countershaft will be supported by two bearings in the transmission housing and the reverse shaft will rest in one bearing in the housing in the form of a cantilever beam.

The design so far enables us to determine the lengths of each shaft. For countershaft, it will be 11.8" between bearings; for the main shaft, it will be 10.8" between the spigot bearing and other bearing in the housing wall; and for the reverse shaft, it will be 1.5" outside the housing walls.

After several trials on combined stresses, we select S.A.E. 4340 which is a Cr-Ni-Mo steel, for the shaft material. Its properites are:

Ultimate tensile stress S_{ut} = 260,000 psi.

Modulus of elasticity E = 28.5×10^6 psi.

We now determine diameters of each shaft separately. We know that each shaft will be subjected to combined torsional and bending stresses. The formula developed with this consideration is [13, P. 478]:

$$d = \left[5.1/T_d \times \{(C_m M)^2 + (C_t T)^2\}^{1/2}\right]^{1/3}$$

where: d = shaft diameter

T_d = design shear stress

= $0.18 \times S_{ut}$ = 46,700 psi. [13, P. 477]

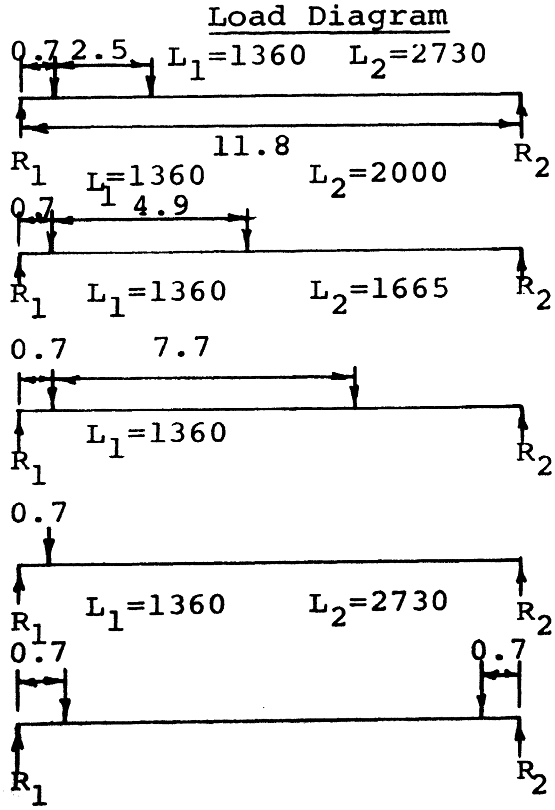
C_m = 1.5

and C_t = 1.0 are bending moment and torsion factors, respectively, for any shock loading;

M and T are bending moment and torque, respectively.

Counter shaft: Two gear pairs at different locations are employed to transmit the engine power in required form in each gear train. Assuming the engine to operate on full power and torque, we shall tabulate M and T for each individual gear train. Here R_1 and R_2 are bearing reactions and L_1 and L_2 are calculated gear loads for both pairs.

Load Diagram



Gear	M	T
Low	7050 lb-in.	3000 lb-in.
Second	5900 lb-in.	3000 lb-in.
Third	4190 lb-in.	3000 lb-in.
Input	895 lb-in.	3000 lb-in.
Reverse	1800 lb-in.	3000 lb-in.

From the above, it is clear that the quantity $(M^2 + T^2)$ is greatest in low gear, so the shaft diameter need only be calculated on that basis. Here:

$$(C_m M)^2 = 1.12 \times 10^8$$

and $(C_t T)^2 = 0.09 \times 10^8$.

So, $\left[(C_m M)^2 + (C_t T)^2\right]^{1/2} = 1.104 \times 10^4$.

So,

$$\text{shaft diameter, } d = \left[(5.1/4.67 \times 10^4) \times (1.104 \times 10^4)\right]^{1/3}$$
$$= 1.06''.$$

Key ways and grooves weaken the shaft so, to be safe, we propose a 1.1" diameter counter shaft.

Reverse shaft: We have load F = 3000/1.1 = 2730 lbs tangential to the idler gear due to the engine torque to be transmitted so at 0.7" from the support on shaft:

$$M = 2 \times 2730 \times 0.7 = 3820 \text{ lb-in.}$$

If shaft diameter is d, bending stress $\sigma_t = M(d/2)/I$. We know that yield point of the shaft material is 156,000 psi. Using a safety factor 2, we get $\sigma_t = 78{,}000$ psi. Substituting:

$$d = 0.795'' \simeq 0.8''.$$

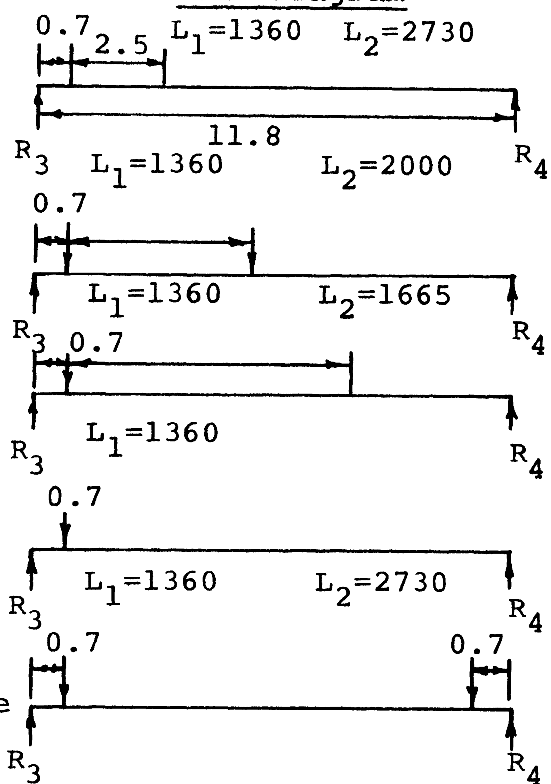
Maximum shearing stress $\tau_{max} = 4V/3A$ [13, P. 33], where:

V = shearing load = 5460 lbs.

A = area of cross-section = $\pi/4 \times 0.64$.

So, $\tau_{max} = 14{,}500$ psi.

Main shaft: Not considering the spigot bearing, bending moments will be the same as those on the counter shaft and torques will be multiplied differently in each pair. R_3 and R_4 are bearing reactions and L_1 and L_2 are gear loads. Tabulating:



Gear	M	T
Low	7050 lb-in.	9000 lb-in.
Second	5900 lb-in.	5700 lb-in.
Third	4190 lb-in.	4330 lb-in.
Input	895 lb-in.	3000 lb-in.
Reverse	1800 lb-in.	8740 lb-in.

Again, shaft diameter is calculated on the basis of low gear M and T. Substituting:

$$\left[(MC_m)^2 + (C_t T)^2\right]^{\frac{1}{2}}$$

in the equation for shaft diameter, we get:

$$d = 1.15''.$$

To compensate for key ways and grooves, we propose 1.2" diameter main shaft.

Deflections: These shafts are subjected to heavy loads and, therefore, to prevent them from lateral deformations, we need to provide an extra pair of bearings.

Lateral deformations depend directly upon the bending moment the respective shaft has to carry. Since M for both shafts is the same, obtaining a satisfactory deflection for the smaller countershaft is sufficient. When the low reduction pair is in operation, concentrated loads act at two points on the counter shaft.

Treating counter shaft as a simply supported beam, we know that:

$M/EI = d^2y/dx^2$

where M = bending moment,

I = moment of inertia

y = lateral deflection

and x = variable along length of the beam.

Substituting for M in terms of x and integrating twice, we get:

$EIy = Pb/6l \left[l/b(x-a)^3 + (l^2-b^2)x - x^3 \right]$ for $a < x < l$.

Here a, b, l and P are as shown in adjoining sketch.

Now it is clear from the loading diagram that maximum deflection due to both loads occurs in the range $a < x < l$.

For the first load:

a = 0.7",

b = 11.1",

and P = 1360 lbs.

For the second load:

a = 3.2",

b = 8.6",

l = 11.8",

and P = 2730 lbs.

Superimposing deflections in general terms with respective a, b, l and P, and then equating its differential to zero, we get maximum deflection at:

$x = 4.05"$ from the left support.

Substituting:

$x = 4.05$ in the general equation for deflection, we get:

Absolute maximum deflection = 0.0033".

The decision as to whether this deflection is satisfactory or not depends upon the pitch of the gear teeth and also upon the accuracy to be maintained in mating action. For our purpose, 0.0033" deflection is satisfactory so there is no need to provide extra bearing. This is a conservative approach, but it was made to be on the safe side. Actually, due to the tooth profile, deflections due to both concentrated loads make a 40^0 angle with each other so their resultant should be an algebraic sum. Our arithmetic sum gives a little larger deflection which is also found to be satisfactory.

VII. BEARINGS

Calculating deflections of shafts, it was seen that for each shaft, a bearing in each wall was sufficient. In addition, there is a spigot bearing in the input shaft supporting the main shaft. Besides providing support, these bearings should have minimum friction. Therefore, we propose the use of roller bearings since they have starting friction only slightly greater than running friction. Ball bearings are preferred since there is no thrust load and they generate less heat. Their friction coefficient varies from 0.001 to 0.003.

Before determining the sizes of bearings, we need to know the load that each bearing will have to carry. This can be easily determined from the requirements that:

1. Every section of the shaft is in static equilibrium.
2. Bending moments at the free ends are always zero for simply supported beams.

Numbering bearings on the counter shaft as 1 and 2 and those on the main shaft as 3 and 4, we can tabulate all reactions as under:

Gear	Bearings 1 and 3 Lbs.	Bearings 2 and 4 Lbs.
Low	3270	820
Second	2330	1030
Third	1760	1265
Input	1280	80
Reverse	1443	2647

From the above table, it is seen that the greatest bearing loads are occuring on all bearing in either low gears or in reverse gear pair operation, the combinations which are used least. Now the limiting load that a bearing can carry depends on how long that load must be carried. It is found that calculated maximum loads have to be carried only a small percentage of the time and that also is calculated on the basis of full engine power which happens infrequently. For these reasons, it is found practical to determine bearing sizes on the basis of loads calculated in the secon gear pair [1, P. 191].

These bearings are mounted on the transmission housing walls and are selected to be of the single row, deep groove type. Mainly they carry radial load, but they are also capable of carrying some thrust load. Geometry and the speed of the shaft in the bore of the bearing may change the equivalent radial load a bearing can carry. AFBMA (Anti-Friction Bearing Manufacturer's Association) suggests the following equation:

$$R_{eq} = x\ v\ L\ F_r$$

where x = radial factor

v = rotation factor

L = load application factor

and F_r = applied radial load.

For radial contact bearings of this type, with a rotating inner ring, all x, v, and L are taken unity so the bearing capacity $= F_r$.

Bearing 1: Its capacity should be 2330 lbs. From S.K.F. Industries Inc. [13, P. 322], if we select 03 series of single row, deep groove ball bearings with 2330 lbs. load rating, we find the dimensions of these bearings as follows:

Outside diameter = OD = 1.8504"

Inside diameter = ID = 0.6693"

Width = 0.5512"

and Fillet radius = 0.039"

Bearings 2, 3, and 4: These are required to carry a smaller load than bearing 1, but for uniformity and according to the minimum bore diameter, we propose these bearings of the same size and type.

Spigot bearing: These bearings in the input shaft are proposed to be a needle bearing with:

OD = 1.0"

ID = 0.5"

Width = 0.75"

and fillet radius = 0.025".

From shear force diagram, we see that it has to carry

3270 - 1280 = 1990 lbs.

We now calculate the load capacity of this bearing which should be more than this 1990 lbs. load to be safe. Following a procedure in [14, P. 172] we find:

Bearing pitch diameter = OD + ID/2 = 0.75".

The roller diameter is selected to be about 10 percent of p. We propose the roller diameter to be:

$d = 0.8"$.

This gives a value

$k = 0.94 = p/d$ [14, P. 172].

The nearest standard k value is now selected as 10.223. This gives:

Exact $p = kd + N(C.C.)/\pi$

where N = number of rollers, generally suggested as 33,

C.C. = circular clearance = 0.0001".

Substituting:

$p = 1.22605"$.

The load capacity is now given by:

$W = p\ L\ c/(rpm)^{1/3}$

where c is a constant = 32,000,

$p = 1.226$,

$rpm = 3160$

width $L = 0.75"$.

Substituting:

$W = 2000$ lbs.

This is slightly more than the theoretical maximum load on a spigot bearing so it is safe.

Material: Balls and needles can be made of Chromium alloy steels S.A.E. 52110, 51100, or 50100. These steels have to be quenched for maximum hardness and tempered for maximum fatigue strength. The cage in each bearing keeps balls or needles apart. They are pressed out of sheet steel, are light in weight, and maintain a free passage of lubricant into the bearing.

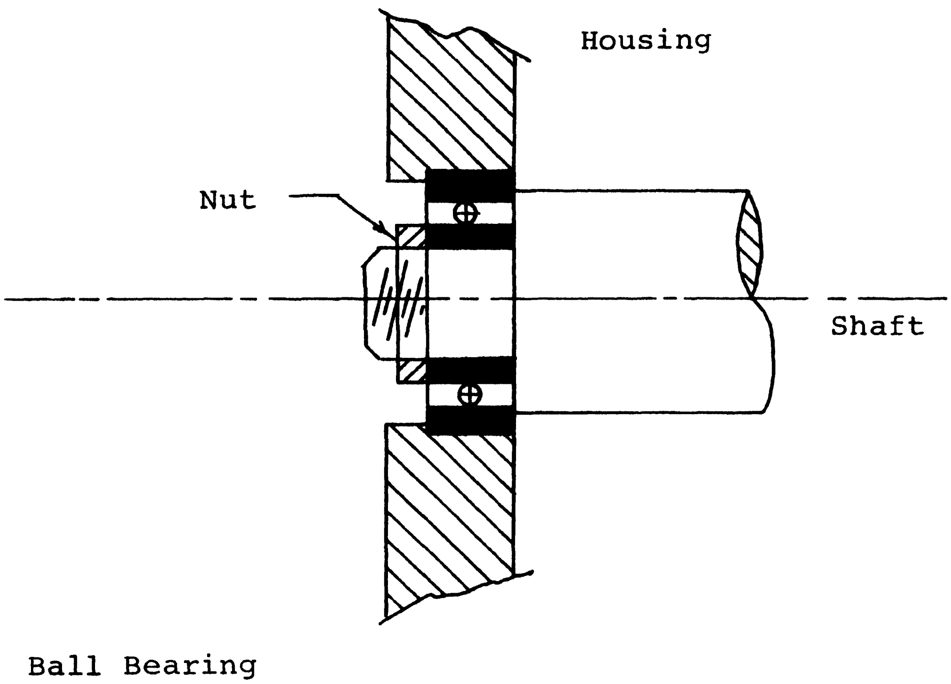
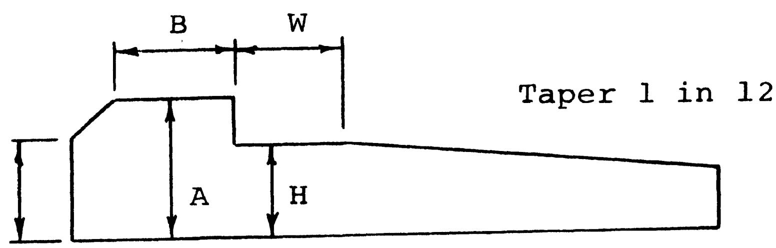


Figure 11: Bearing

Mounting: Races of bearings in housings are backed up against the shaft shoulder and a housing shoulder. The outer race is held in position and prevented from any axial movement by a round nut as shown in Figure 11.

VIII. FASTENERS

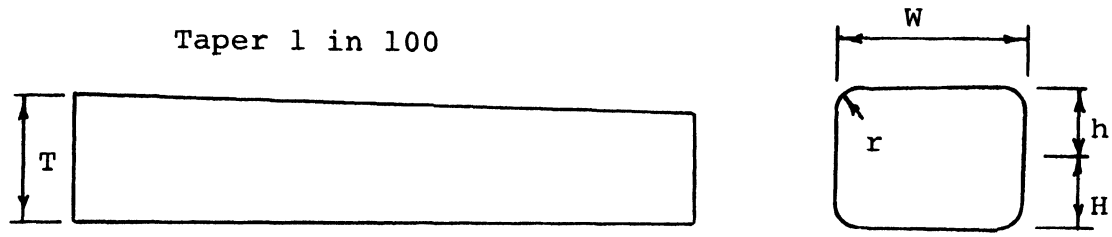
Keys: Gears on the counter shaft are fastened to their respective drums and other gears and hubs are fastened directly to their respective shaft. For easy removal and assembly, a gib head square key is selected. Its taper, when firmly driven in, acts to prevent any axial motion, and its head makes removal easier without access to the other end. Material for these keys is proposed to be a cold drawn or cold rolled steel. Every key is standardized according to an A.S.M.E. code, depending upon the diameter of the shaft on which it fits. Figure 12 shows dimensions of a gib head key for a shaft diameter from 15/16" to 1 1/4".



W	H	A	B	C	Tolerances in W, H, and gib head taper
1/4"	1/4"	7/16"	11/32"	11/32"	0.002" each

Figure 12: Gib Head Taper Key

The load due to gears on the countershaft is also radial. Due to axial fluid force in drums, however, there will be a tendency for axial motion of the drums and, hence, gears. To prevent this motion of the freely rotating drums, retaining rings are provided which axially fix the respective drum. In the presence of these rings, a rectangular taper key for gears is quite sufficient. Its dimensions as found in the A.S.M.E. code are as shown in Figure 13.



W		T		h	H	Keyway
max.	min.	max.	min.	in hub	in shaft	Radius r
0.502"	0.5"	0.316"	0.311"	0.114"	0.19"	0.02"

Figure 13: Rectangular Taper Key

The drums of the clutches have to freely rotate when their respective clutches are not engaged. This requires a clearance between the drums and the respective shaft. Bolting this drum on the outer end to the cover plate, however, keeps the drum concentric with the shaft. Considering the extreme possibility of the drum being eccentric, we provide a sleeve bearing in this gap to take up any rubbing wear possible. The axial load is taken by the retaining rings and the radial load due to gear is transmitted through the bolts on the outer periphery to the cover plate and then to the stationary boss. Theoretically, this reduces to nil the load acting on the sleeve bearing. Any indeterminate load on this bearing will be carried by the resisting oil, always present in the provided gap. This load, however, is negligible small, and we empirically fix the sleeve bearing dimensions:

Outside diameter of the sleeve = 1.4"

Inside diameter of the sleeve = 1.15"

Gap between shaft and sleeve = 0.025

and length of the sleeve = 1.0".

The bronze sleeve or bushing is press fitted into the inside diameter of the drum boss.

We provide 1/32" thick external retaining whose outer and inner diameters fixed by our geometry are:

Inner diameter= 1.0",

Outer diameter = 1.25".

The inner diameter of the retaining ring will fit into a groove cut in the 1.1" diameter shaft. The outer diameter is greater than the inside sleeve diameter (press fitted with the drum) so it will resist any axial load.

Similarly the reverse idler gear is to rotate freely on a 1/8" thick sleeve. To prevent any axial motion, proper retainings are provided on the reverse shaft to hold the gear in position.

IX. CONTROL VALVES

The required 138 psi, hydraulic pressure is to be supplied by an engine-operated pump. The pressure is maintained constant by providing a relief valve in the grain line near the pump. This constant pressure will have to be directed to the appropriate clutch. For a semi-automatic transmission, this should be left to the driver. We propose six solenoid operated valves for this control.

When the driver operates the gear selecter lever or presses a button, he simply makes the circuit of the solenoid concerned. This energizes the solenoid located at the bottom of the transmission box. When current flows through it, the plunger of the control valve is activated. This motion connects the pressure line from the pump to the inlet of the clutch through the stationary boss. The clutch is then engaged.

When the driver changes the lever selection, he breaks the circuit and thus deenergizes the original solenoid. A spring within the valve then pushes the plunger back to its original position. The main pressure line is therefore disconnected and a passage to the transmission box is opened to the inlet of the clutch. Through this opening, the fluid from the pipe drains causing a lack of pressure above the piston of the drain valve. The piston uncovers a port T, drains fluid behind the diaphragm, and thus disengages the clutch. Simultaneously, the newly energized solenoid engages the other clutch.

When the driver shifts to neutral, no solenoid is energized. This keeps all clutches disengaged and the pressurized fluid just circulates from the outlet to the inlet of the pump through the transmission box.

In Figure 14, we can see that the plunger is shown in "neutral" condition with the solenoid deenergized. The arrangement is such that the draining port is completely closed when the pressure port starts to open. The port openings are proposed to be 1/8" wide and overlap of the piston over the port is also kept 1/8". This requires the plunger to be moved 1/4". For this purpose, we propose to use an electric CR 9503, size N solenoid. When the solenoid is de-energized, the spring takes effect. This requires that the solenoid rated force be greater than the spring force.

For designing control valve spring, we follow the procedure described in the Design Handbook [6]. We assume a mean coil diameter of 1/2". It is also assumed that initial compression is 5/16" and free length of the spring is 1.3". This gives:

L_1 = length before compression = 1.3 - 5/16 = 0.997"

and L_2 = length after compression = 0.997 - 0.25 = 0.727".

We desire a spring which exerts 10 lbs at L_1 and 25 lbs. at L_2. Now we proceed as follows:

1. We have assumed D = 1/2" and we also assume maximum shear stress $S_2 = 10^5$ psi. as recommended.
2. Wire diameter $d = (2.55 \times P_2 D)/S_2$ where P_2 = 25 lbs.

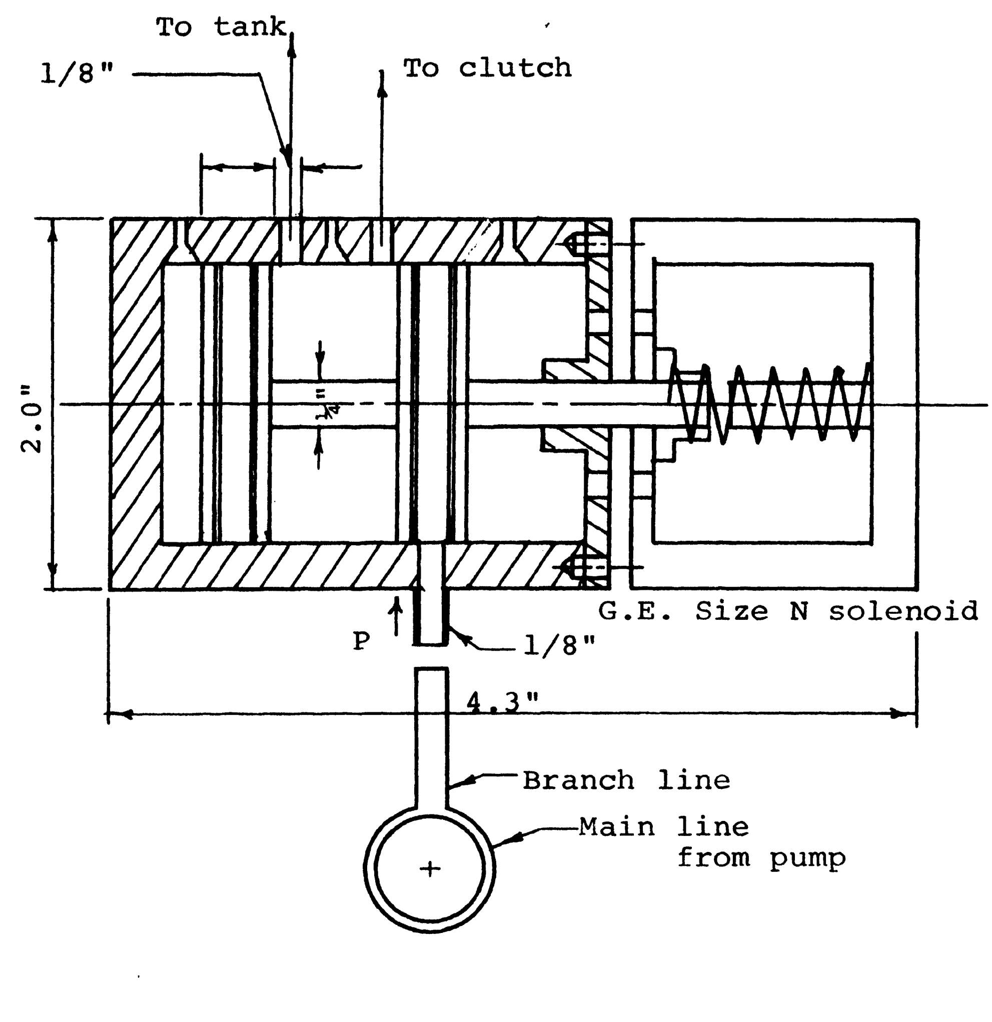


Figure 14: Plunger type, Solenoid operated control valve.

$= (2.55 \times 25 \times 0.5)/10^5$

$= 0.0685"$.

3. Now, the outer diameter of the spring

   $= 0.5 + 0.0685 = 0.5685"$.

4. Factor $K = (P_2 - P_1)/(L_1 - L_2) = 15/0.25 = 60$.

5. If the modulus of rigidity, $G = 11.5 \times 10^6$ psi. then the number of active coils,

   $n = Gd^4/8D^3K = 4.1$ or say 5 coils.

6. Keeping two extra coils for the ends, makes the solid length $L_s = 7 \times 0.0685 = 0.4795"$.

   Free length $L_f = 25/60 + 0.727 = 1.144"$.

   This gives: $f_2 = L_f - L_2 = 0.417"$.

   Clearance between consecutive coils

   $= L_2 - L_s = 0.2475"$.

7. Deflection when solid $= f_s = L_f - L_s = 0.7645"$; so, $P_s = K \times f_s = 45.7"$.

   So, stress at solid $S_s = (2.55 \times D \times P_s \times K_{w1})d^3$.

   Here, Wahl factor $K_{w1} = 1.2$ at $C = 0.5/0.7645 = 7.3$;

   So, $S_s = 150{,}000$ psi.

8. We have $0.1\ f_2 = 0.0417"$ which is less than the clearance calculated.

9. From [6, P. 46], we confirm that S_s is quite less than the yield point stress in tension.

10. From [6, P. 42], we have tensile stress T_s $= 240{,}000$ psi. and, hence, $0.3\ T_s = 72{,}000$ psi., which is less than S_s calculated so the design is complete and safe.

X. PUMP

The pump will be driven from the engine and its purpose will be to supply pressurized fluid to the clutch chamber. Our geometry fixes the outside and inside diameters of each diaphragm to be 6.0" and 3.0" so, if the diaphragm chamber is assumed 1/16" thick, its volume is:

$$V = 1/16 \times (\pi/4) \times (36 - 9) = 1.32 \text{ in}^3.$$

Maximum gear changing period is to be only 0.4 seconds so in order to fill that much volume in 0.4 seconds, the pump capacity should be at least 0.86 GPM. In addition to the diaphragm chamber, each time a clutch is re-engaged, pipe pieces connecting the control valve to the chamber, as well as some volume in the control valve, will have to be filled up so we propose a 1.0 GPM capacity pump. When a clutch is engaged, after 0.4 seconds, the entire volume mentioned above is filled. It then offers resistance to any incoming fluid which, in turn, increases the pressure immediately. As soon as this pressure reaches 138 psi. mark, the relief valve provided in the line opens, bypasses extra fluid and does not let pressure rise above 138 psi. See Figure 15.

We, thus, have to look for a low capacity, rotating type of pump. We follow the current automotive trend in oil pumps and select a gear pump for this which is similar in size, design and capacity to the automotive pump. As the capacity required is low, we select gears with the

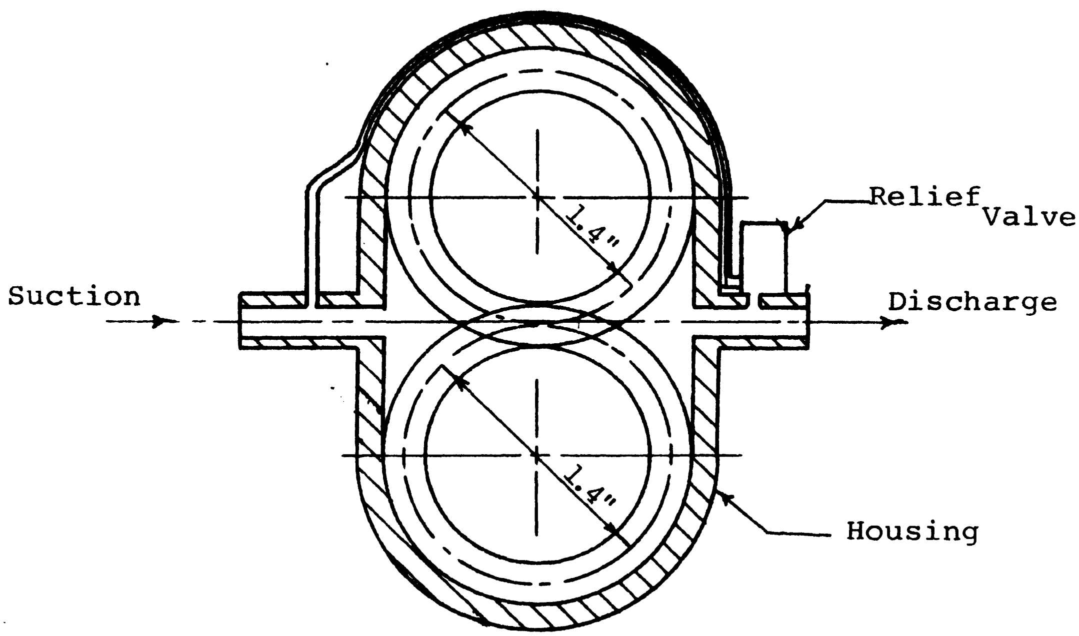


Figure 15: Schematic diagram of a gear pump

minimum 14 teeth to facilitate smoother and quieter operation due to less number of overall engagement changes. To determine various tooth proportions, we need to select a standard diametral pitch which is again preferred to be 10. Here, we select full depth teeth with only 0.4" face width. Standard proportions are:

Addendum = $1/P_d$ = o.1"

Dedendum = $1.157/P_d$ = 0.1157"

Whole depth = $2/P_d$ = 0.2"

Whole depth = 2.157 P_d = 0.2157".

This gives a clearance between the root circle of one gear and the top of the mating gear to be 0.0157". We also propose the pressure angle $\phi = 20^0$ and backlash B = 0.003; so:

Arc tooth thickness = $(p-B)/2 + \Delta_a(2\tan\phi)$ = 0.155".

For obtaining a minimum volume between teeth, we take 0.14" space between consecutive teeth and assume a rectangular cross-section of size 0.14 x 0.2157 sq. in. in the plane of the gear. This will give the total volume where fluid will be trapped in both gears:

$$V = 13 \times 0.14 \times 0.2157 \times 0.4 \times 400/231$$

$$= 2.72 \text{ GPM.}$$

This more than doubles the required capacity; but as we have assumed the minimum pump rpm to be as high as 400 rpm, we need to revise it as in actual practice, it may be less. If it is assumed to rotate at 200 rpm, the pump capacity will be 1.36 GPM which is just little more than required, so this size is quite satisfactory.

Pitch diameters of each gear will be 14/10 = 1.4".

So, the overall pump size is:

Outer diameter of gears + housing clearance

+ housing thickness

= 2.8 + 0.2157 + clearance + thickness

= 3.0157" + clearance + thickness.

XI. TRANSMISSION FLUID

We know that the transmission fluid not only has to act as a power transfer medium, but also has to lubricate gears and bearings. This causes some of the requirements to conflict and, hence, a suitable compromise is needed. For instance, a lubricant has to reduce wear which directly depends on oil viscosity in addition to the force between and the velocity of the contact surfaces. The larger the viscosity, the more the load the oil film at contact surface will carry. On the other hand, a low viscosity oil has the advantage that it can dissipate heat very readily. Therefore, for power transfer purposes, a low viscosity oil is attractive.

A power transfer medium should also have the following ingredients to achieve the described characteristics:

1. The oil has to be compatible with materials and temperatures, and to achieve this compatibility, some extreme pressure additives, which have limited solubility in high viscosity oils, are blended with Napthenic oils.
2. Additives like Phenates or Sulphonates are important in freeing the gear oil from separation and imparting detergent qualities to oil.
3. The oil should also be capable of operating between -30^{0}F. and $+300^{0}$F. always and for that, the addition of Napthenic or paraffinic oils should be preferred.

4. The oil should be resistant to oxidation due to longer exposure to oxygen at high temperatures and should not form sludge or varnish.
5. The oil should be rust preventive and should have an anti-corrosion agent which will not adversely affect any of the metals, seals or gaskets.
6. Certain additives to act as an anti-foam agent, a viscosity index improver, an odor preventing agent, etc. are a must.

Thus, a wide variety of additives also have to be considered before making the final selection. The best choice it seems is ATF, TYPE A, Suffix A transmission fluid which is available to fulfill the following requirements. It has:

1. Excellent miscibility without separation or color change.
2. Viscosity of 49 SUS minimum.
3. Flash and fire points at 320^{0}F and 350^{0}F respectively, minimum.
4. Good fluidity at low temperature, good anti-foaming properties, anti-corrosion properties and it forms no sludge or deposits.
5. A very small effect on the volume and hardness of the seal and has anti-odor, non-chatter, non-toxic and anti-oxidation properties.
6. A high durability and its performance is equal to or better than that of the reference fluid.

XII. SUMMARY

It appears that the present approach has achieved its main objective of designing an effortless and cheap transmission with a high efficiency for small cars which have low power to weight ratio. This is done by obtaining such a combination as to avoid the application of a torque converter. While increasing the efficiency, the present approach also obtains a compact and flexible transmission. To avoid any expensive synchronizers or sliding devices, a constant mesh transmission was proposed. The problem of making it an effortless transmission was solved by employing simple plate clutches where fluid pressure supplied by an engine driven pump substituted for manual effort. To keep the choice of gear selecting at the driver's disposal, however, an electric solenoid valve per pair is introduced.

For this combination to be feasible, however, a device which enables the transmission to take up and transmit the engine power gradually in the required form to the output shaft must accompany it. This is because 0.4 seconds of clutch engagement period was set considering the automobile in a running condition, operating on a horizontal track. While starting from a dead stop condition, however, the relative motion between the driving plate and respective drum is less as the input speed is less. This needs even less time for engagement and it is quite possible that due to external torque loads, the engine gaining the power gradually might stall. Even if not, the instant shock to the vehicle in this short period is quite inadequate. The same thing is true while the automobile is descending a steep hill. It is, therefore, suggested that between the engine output and transmission input shaft, a master clutch be provided. This clutch may also be hydraulically operated by the fluid pressure supplied from the gear pump, but a pressure regulator, which is sensitive to the engine speed should be installed in the line from pump to this master clutch. The pressure regulator will control the fluid pressure till the engine gains power enough to take any instant torque load. The master clutch till then will slip and, thus, prevent any instantaneous engagement. After the engine gains full power, the regulator will always supply the required pressure to the master clutch and, thus, keep it engaged.

The advantages of this combination are more evident if a device to make control valve operation automatic is found. It is suggested that a torque switch and a small generater be employed which will supply a current, depending upon the road speed to a control box which will actuate a right control valve.

BIBLIOGRAPHY

1. Heldt, P.M. Torque Converters or Transmissions. New York; P.M. Heldt, 1942, 1-25, 100-140, 175-180.

2. Judge, Arthur W. Modern Transmission Systems, Motor Manual 5. London: Chapman & Hall Ltd, 1962.

3. Newton, K. and Steeds, W. The Motor Vehicles. London: ILIFFE & Sons, LTD, 1950.

4. Larew, Walter B. Automatic Transmissions. New York: Chilton Company, 1966.

5. Dudley, Darle W. Practical Gear Design (Ed. 1). New York: McGraw-Hill Book Company, Inc., 1954, 7-10, 18-20, 40-47, 76-90, 152-170.

6. Design Handbook. Springs, Custom Metal Parts. Bristol: Associated Spring Corporation, 1967.

7. Machine Design. The Seals Handbook. Cleveland: Penton Publishing Co, 1961, 24-30, 48-50, 77-83.

8. Gask-O-Seal Handbook. Cleveland: Parker Seal Co, 1963.

9. Ernst, Walter. Oil Hydraulic Power and Its Industrial Applications. New York: McGraw-Hill Book Co., 1949, 11-15, 66-90, 206, 221-233.

10. Pippenger, John J. Fluid Power Controls. New York: McGraw-Hill Book Co., Inc.,1959, 98-132.

11. Boner, C.J. Gear and Transmissions Lubricants. New York: Reinhold Publising Corporation, 1964, 1-18, 34,38, 50-53, 180-183.

12. Timoshenko, S. and Young, D.H. Elements of Strength of Materials. New York: D. Van Nostrand Company, Inc., 1962.

13. Shigley, Joseph Edward. Mechanical Engineering Design. New York: McGraw-Hill Book Company, 1963.

14. Slocum, S.E. The Theory and Practice of Mechanics. New York: Henry Holt and Company, 1913.

15. Myatt, Donald J. Machine Design Problems. New York: McGraw-Hill Book Co., 1959.

16. Stephenson, Reginald J. Mechanics and Properties of Matter. New York: John Wiley and Sons., Inc.

17. Hesse, Herman C. and Rushton, Henry J. Process Equipment Design. New York; D. Van Nostrand Co., Inc.

18. S.A.E. Handbooks, 1956, 1957, 1969, etc. New York: Society of Automotive Engineers.

19. Morton, Hudson T. Antifriction Bearings. Michigan: Annarbor Press, 1965.

VITA

Yogendra Prahladray Buch was born on June 14, 1946, at Baroda, India. His father, Mr. Prahladray H. Buch, was a judge and was transferred every three years so he had his primary education in seven different high schools. He graduated from Kadiwala High School, Surat, India, in 1962. He then joined St. Xavier's College, Ahmedabad, India, for his two year pre-engineering education and received his B.E. in Mechanical Engineering from L.D. College of ENGG., Ahmedabad, India, in 1967.

In June, 1968, he started his masters studies at the University of Missouri-Rolla and had several breaks due to a car wreck soon after that.