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PARAMETRIC STUDY ON THE ROLLER GEAR REDUCTION DRIVE

TRW Report ER-5637

By

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J. E. Bauer

June 1965

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

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THOMPSON RAMO WOOLDRIDGE, INC.

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This report represents a part of a continuing research program for the investigation of new concepts of high-speed reducers for use as main transmissions in helicopters. The main efforts of this program are directed toward deriving a reduction unit, or units, with a reduction ratio significantly higher (40:1 and above) than those of currently used transmissions and which would be more compatible with the high rotational speeds of aircraft turbine engines. With this objective in mind, the roller gear drive investigation was undertaken.

This Command concurs with the contractor's conclusions and recommendations reported herein, and a continuing program is scheduled on this basis.

NOTE

On 1 March 1965, *after this report had been prepared,* the name of this command was changed from U.S. Army Transportation Research Command to:

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PARAMETRIC STUDY ON THE ROLLER GEAR REDUCTION DRIVE

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ABSTRACT

This report covers the parametric study of the roller gear drive to determine its applicability to current or projected Army helicopters. The basic principles used in the roller gear drive are presented. The results of a study to determine the weight, volume, and outside diameter as a function of ratio (from 20:1 to 100:1) and horsepower (from 250 to 4,000) are included. Other features, such as reliability, efficiency, life, vibrations, and cost are discussed.

PREFACE

This report covers the parametric study for the TRW roller gear reduction drive. It concludes the work under Phase 1 of a contract with the U. S. Army Transportation Research Command (USATRECOM) to develop an advanced transmission for eventual use in helicopters.

The TRW transmission is a product of the ideas conceived by Dr. A. L. Nasvytis. TRW has patents pending that cover his ideas.

The work on the parametric analysis began on receipt of the contract on 21 June 1963. The principal investigator and project manager was Dr. A. L. Nasvytis. The engineers working under his guidance were Messrs. O. E. Kee and J. E. Bauer.

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SYMBOLS

This section defines some of the newly conceived concepts presented in this report. These definitions apply to both the description and the calculations. Other definitions are presented within the text.

A	sun gear
a	pitch radius of the sun gear
c	pitch radius of the ring gear
D_p	pitch diameter of the sun gear
f. w.	face width of gears
F_t	tooth tangential bending force
G	shear strength modulus
J	polar moment of inertia about the centroidal axis
K	K-factor value, compressive load factor - $\frac{F_t}{D_p \text{ (f. w.)}} \cdot \frac{R+1}{R}$
L	length
n	number of last row planets
P_d	diametral pitch
R	ratio of larger first row pitch radius to sun gear pitch radius (x_1/a)
R_o	overall ratio
S_b	Bending stress
T	input torque
T_o	output torque
$X_{1,2,3}$	larger gear of a stepped planet gear with the subscript indicating the row number

$x_{1,2,3}$	pitch radius of the larger gear in a planet gear with the subscript indicating the row number
$Y_{1,2}$	smaller gear of a stepped planet gear with the subscript indicating the row number
$y_{1,2}$	pitch radius of the smaller gear in a planet gear with the subscript indicating the row number
y	Lewis tooth form factor
z_1	distance from the center of the sun gear to the center of the second row planet
z_2	distance from the center of the sun gear to the center of the third row planet
α_1	first toggle angle is the angle between the lines connecting the centers of the sun gear to the first row planet and the first row planet to the second row planet minus ninety degrees.
α_2	second toggle angle is the angle between the lines connecting the centers of the sun gear to the second row planet and the second row planet to the third row planet minus ninety degrees.
γ_1	complement of the first toggle angle is the angle between the lines connecting the centers of the sun gear to the second row planet and the second row planet to the first row planet
γ_2	complement of the second toggle angle is the angle between the lines connecting the centers of the sun gear to the third row planet and the third row planet to the second row planet
θ	half angle between last row planets

SUMMARY

This report concludes Phase I of a study performed by the Accessories Division of Thompson Ramo Wooldridge Inc. on the TRW roller gear reduction drive. It presents the results of a parametric analysis on this new method of power transmission to determine the parameters necessary to design and build an experimental helicopter transmission for USATRECOM under Contract DA 44-177-AMC-30(T).

Because of the broad scope of the investigation of the drive applications to helicopters, the study was restricted to the basic drive itself, omitting suspension bearings, one-directional clutches, accessory pads, tail rotor drives, etc. The drive design types were checked to ascertain that no significant obstacles existed to discourage their use in the helicopter. None were found. The basic drive was investigated for use in a helicopter over a range of from 250 to 4000 hp, from 12,000 to 30,000 rpm, and with 20:1 to 100:1 reduction ratios.

Only the planets in the last row require bearings. Thus, all other rows of planets are accurately located by the rollers' acting as bearing surfaces, and looseness in these rows due to bearing play has been eliminated.

The rollers make each planet self-aligning because they provide at least a three line contact for them. The roller gear drive appears to be superior to the two and three stage planetary drive in respect to weight, reliability, vibration, life, and efficiency.

This report contains the results of a study using the derived roller gear drive concepts to determine the best possible design for lowest weight and volume. These results are based upon commonly available materials and present design practices. No attempt has been made to anticipate possible improvement in the state of the art of materials and design practices.

CONCLUSIONS

1. No significant obstacles appear to exist to prevent the roller gear reduction drive from being developed and used in helicopters.

2. The roller gear drive has high efficiency. The actual measured losses in the TRW test drive before alteration were only 2 percent at rated speed and horsepower. After modifications to the drive, the losses were less than 1.5 percent.

3. The roller gear drive should have less vibration than the present planetary drives because of its inherent damping characteristics.

4. The roller gear drive is expected to have longer life than the present planetary drives. With development, the roller gear drive should have a life equivalent to or better than that presently experienced in gas turbines.

5. The roller gear drive shows an inherent improvement in reliability when compared to the conventional gear reduction drive. The reasons for this improvement are:

   a. Fewer bearings are required.

   b. All high-speed bearings are eliminated.

   c. Shorter shafts give greater rigidity.

   d. Inherent damping of drive causes low vibration.

   e. Housing distortion causes less misalignment of gear contacts due to mounting of the output planets in spherical bearings.

   f. Machining tolerance of rollers is an order of magnitude better than that of conventional gears, assuring line contact of gear teeth and accurate pitch location.

   g. Damaged or worn teeth affect operation of the roller gear drive less than that of the conventional planetary system.

   h. More equal distribution of load between planets permits more optimal designs which reduce wear and which improve efficiency and endurance.

6. The roller gear drive concept has significant weight advantages for the reduction ratios investigated (greater than 20:1).

7. The roller gear drive concept will readily accommodate accessory drive installations.

RECOMMENDATIONS

1. It is recommended that a program be initiated to design and fabricate a roller gear drive on the basis of the favorable results of the parametric analysis and the encouraging results of the TRW-sponsored test program which has proven the feasibility of the roller gear drive concept. Subsequent to the parametric analysis, such a program is being pursued, with an associated test program to be conducted by USATRECOM.

2. Pending favorable results of the test program to be conducted by USATRECOM on two experimental units, it is recommended that a program be initiated to design a roller gear transmission for a specific helicopter application.

INTRODUCTION

This report reflects the results gained from a basic study on the application of the TRW roller gear drive principle to helicopter transmissions. The study was a result of a need by the Army for a power transmission capable of matching the speed, size, and life of the gas turbine.

In 1961, TRW designed and built a roller friction drive. No-load spin tests completed in 1962 in the TRW laboratory verified the feasibility of using a cluster as a power transmitting device and using bearings only in the last row to carry the torque reaction.

The roller gear drive evolved from the roller friction drive. It became obvious that the principle of using rollers to support the torque carrying members could be used in conjunction with gear teeth to carry the torque. The advantages of this approach are increased power capability and elimination of the high radial preloads necessary in the roller friction drive.

TRW recognized the Army's need for a power transmission capable of matching the speed, size, and life of the gas turbine. We therefore initiated a company funded program to back the development of such a drive. The test drive, called the AN-1, has been designed and the prototype tested.

The TRW test program has run concurrently with this study, and its results have proven the basic principles of the roller gear drive and the validity of the analytical predictions made. The efficiency data presented in the report are based on these test results.

Analytical investigations during Phase I of the Government program indicated that the roller gear drive can be significantly lighter and more efficient than the conventional reduction drives when high ratios and high speeds are required.

Gas turbine driven helicopters appear to be a natural application for this type of drive, although its advantages could equally be employed in other applications using high-speed turbines and requiring reduction ratios greater than 10:1. The roller gear drive could be used with piston engines (low reduction ratio) but at little, if any, weight savings. The advantages in this case appear to be better efficiency and reliability.

The roller gear drive offers numerous variations in arrangement as to number of rows of planets, number of planets per row, types of output gear, staggering or nonstaggering of planets (in a row and in other rows), and various stepped planet configurations. The Phase 1 parametric study considers all these variations, as well as the limiting of combinations of numbers of teeth as functions of toggle angle. This study is a necessary prerequisite to reviewing systematically the application of this drive to a particular helicopter size and configuration.

THEORY

The evolution of the TRW roller friction and roller gear drives begins with the simple planetary friction drive shown in Figure 1. In this simple friction drive, the preload, which is necessary to develop the frictional force required to transmit the tangential load, is the same at the sun roller as at the ring roller contact. This preload is created by the elastic deformation of an undersized ring roller. The preload forces are balanced out between the roller contacts and thus do not load the bearings located in the idler planets.

The plenetary friction drive suffers from numerous limitations which limit the thermal efficiency of the device. Some typical ones are difficulty in keeping planets aligned and sensitivity to thermal expansion. Further, the maximum speed ratios which can be obtained within a single stage are limited by the number of planets, i. e., for 3, less than 12, etc. In order to achieve higher ratios, more than one stage must be used.

A demand exists for drives of higher reduction ratio than is possible with the single row of planets. Up to now higher reduction ratios has been obtained by putting two or more simple planetary drives axially in series. This method produced pronounced disadvantages for high speed and high ratios, two of these being high rotary speed in the first stage planet output and a large sun input size in the last stage planet. The roller friction drive and roller gear drive concepts eliminate these limitations.

This section presents the theory of the TRW roller friction and roller gear drives and discusses the problem areas. A theory is developed whereby the minimum weight and volume design trends can be established for roller gear drives.

TRW REDUCTION DRIVE PRINCIPLE

TRW has started to exploit another logical possibility: to obtain a high ratio by putting additional radial planet rows (stages) in one place between the sun input and the ring output. Figure 2 shows the transition to the stepped roller planetary drive. A schematic representation of the simple planetary drive is shown on the left of the figure; a stretch-out of the stepped roller improvement is shown in the center of the figure, and its schematic representation is shown on the right.

The total ratio of the TRW drive consists of the ring diameter to sun roller diameter ratio multiplied by the ratio of the larger (input) element of each row planet divided by its smaller (output) element. This way the total ratio gain is considerable. Besides the planet ratio factor, the ratio between the ring and sun rollers can be larger because the geometrical restrictions for the ratio in a simple planetary drive are eliminated.

Roller Friction Drive Concept

This type of drive became practical after the discovery that the preload of all planet contacts can be almost ideally matched to that needed to transmit the torque. The preload forces must subsequently be larger on each stepped planet in the rows of planets

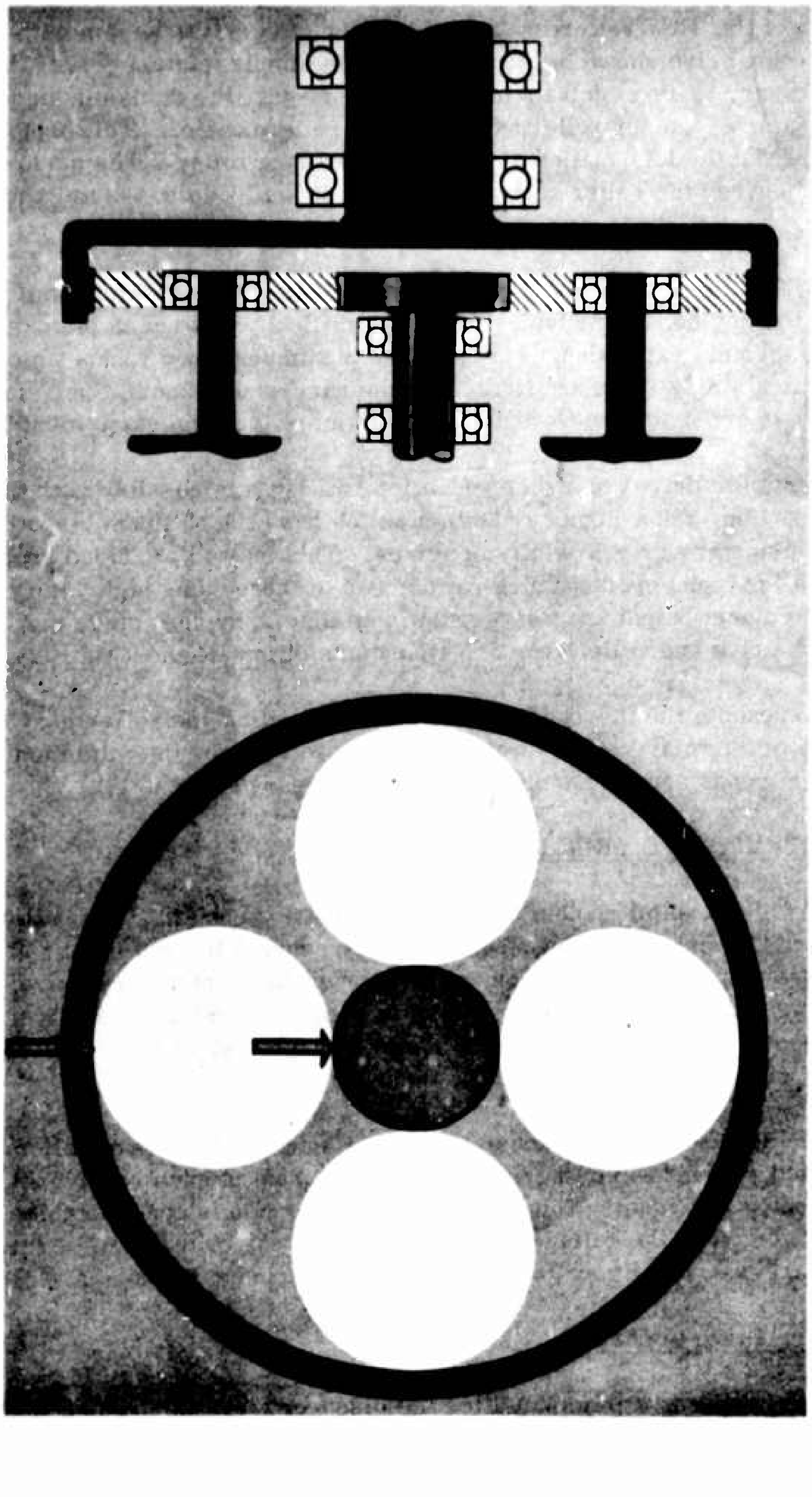


Figure 1. Simple Planetary Roller Drive.

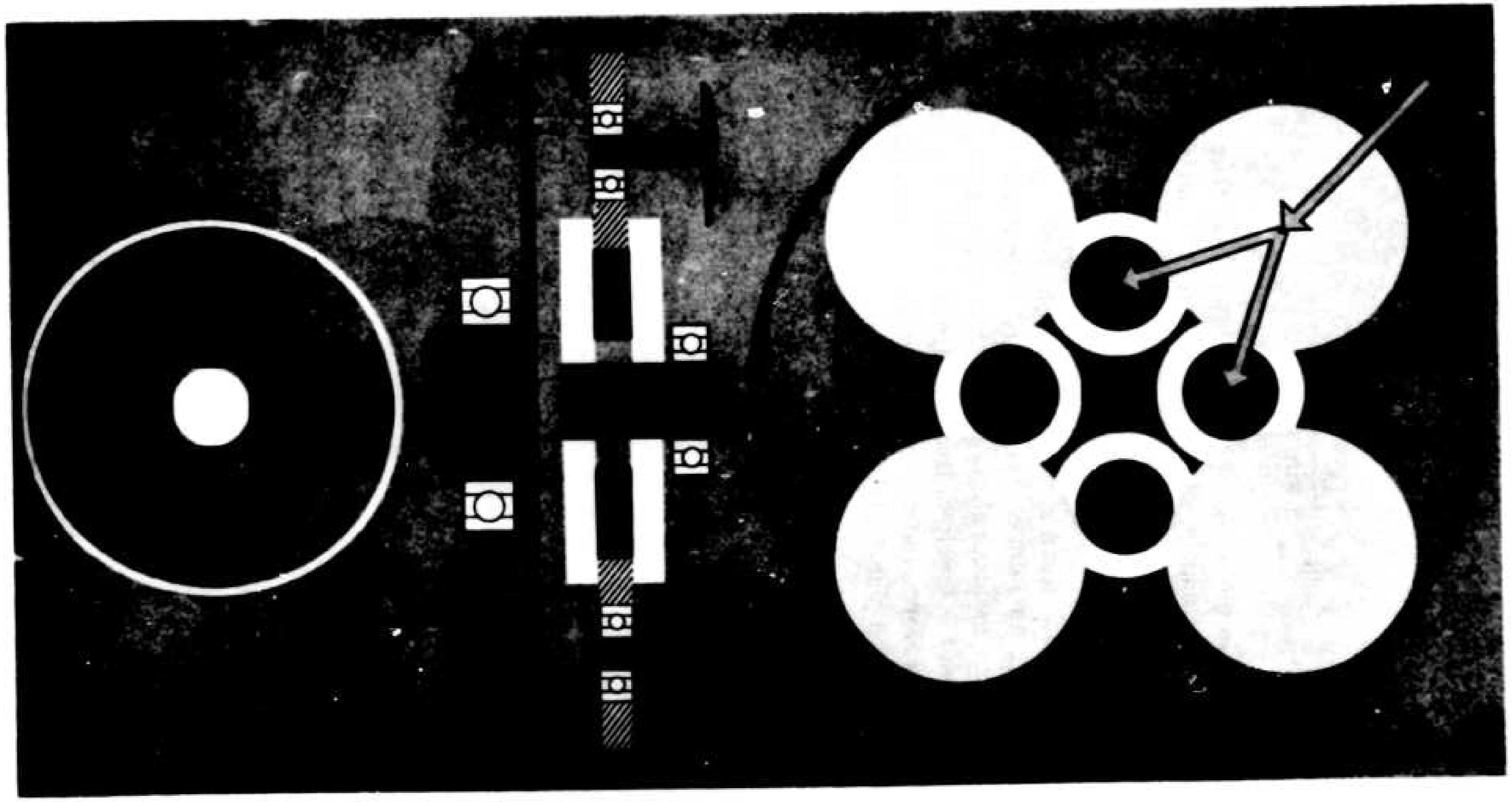


Figure 2. Transition to the Stepped Roller Planetary Drive.

going outward from the center to the ring roller. This is caused by the slower rotation of each succeeding row of planets, resulting in their larger torque.

The adequate forces are obtained by toggle action phenomena. The preload forces are multiplied according to the law:

$$P_{out} = \frac{P_{in}}{2 \sin \alpha}$$

This formula describes a simple wedge without friction, and that is essentially what the roller friction drive force multiplication is. Figures 3 and 4 show the force diagram. The value of $1/\sin \alpha$ can be matched to the stepped planet ratios by the selection of the proper geometry.

In general, the planets can be either stepped or nonstepped. In the case of the planets being nonstepped, they act as idlers. Thus the only ratio advantage in this case is the larger dimensional differences attainable between the sun and ring rollers. The forces are still magnified because the toggle action results in a larger than required preload on the outer planets. Therefore, the drive with the nonstepped planets is heavier and less efficient than the stepped planet drive.

The roller friction drive can be made up of many cluster arrangements. Three of the possible arrangements are shown in Figures 2, 3, and 4. In order to maintain a clear designation, the following system of indicating arrangement is adhered to for both the roller friction and roller gear drives.

The numbering system to indicate cluster arrangement of drives is $n_1 - n_2$ or $n_1 - n_2 - n_3$, where n_1 is the number of planets in the first row, n_2 the number of planets in the second row, etc. For instance, a drive with 3 rows of planets with 3 planets in the first row and 6 planets in both the second and third rows would be designated as a 3-6-6 drive. The indices are not used, as the location of the number in sequence designates the row counting from the center outward.

A roller friction drive was built for the Naval Underwater Ordnance Station (NUOS) at Newport, R. I., for noise study purposes. The drive has a 3-6-6 arrangement. It will be referred to throughout this report as the NUOS roller friction drive. A photograph of this drive is shown in Figure 5; a schematic is shown in Figure 6. It is approximately 15 inches in diameter and transmits 500 horsepower. The input speed of 53,000 rpm (counterclockwise viewed from sun input) provides an output of 1100 rpm (clockwise) through a reduction ratio of approximately 48.2:1.

The ring roller rotates and the cluster is stationary. This is necessary to provide for two accessory pads: one for 60 horsepower rotating at 3913 rpm (clockwise) located on one of the third row planets (see spline in Figure 5) and the other for 10 horsepower rotating at 12,096 (counterclockwise) located on one of the second row planets.

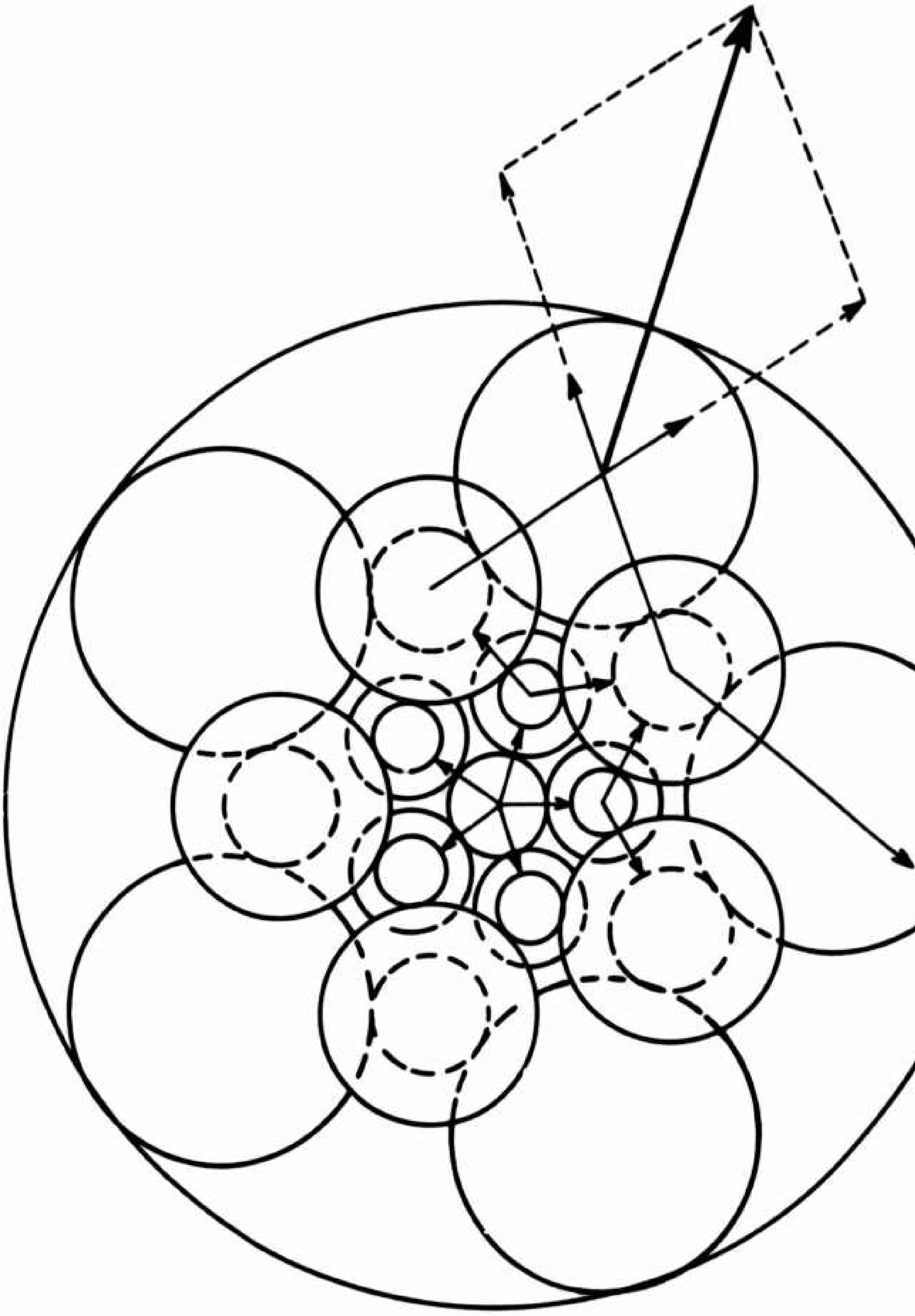


Figure 3. 5-5-5 Friction Drive Preload Vectors.

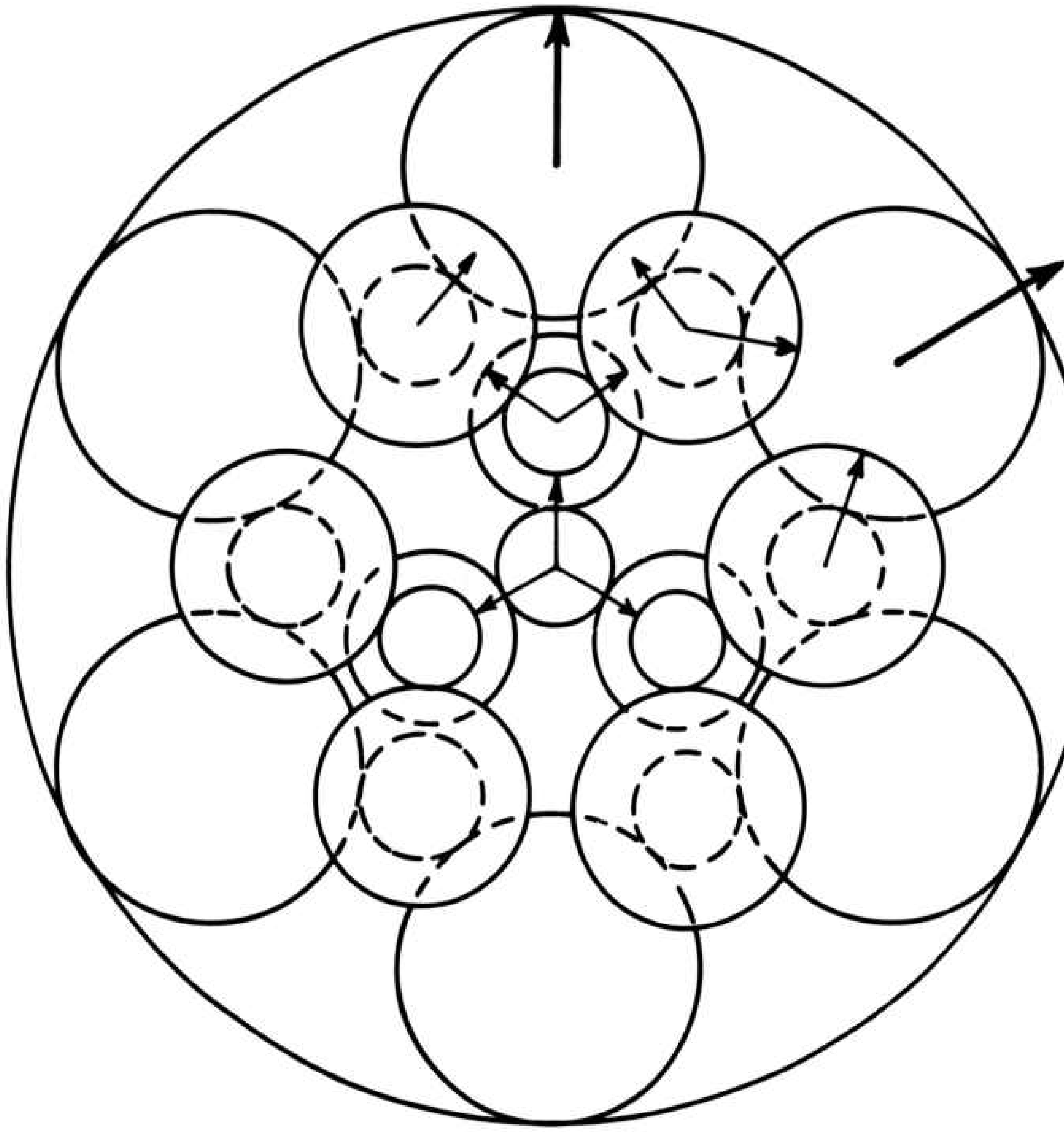


Figure 4. 3-6-6 Friction Drive Preload Vectors.

Figure 5. NUOS Roller Friction Drive.

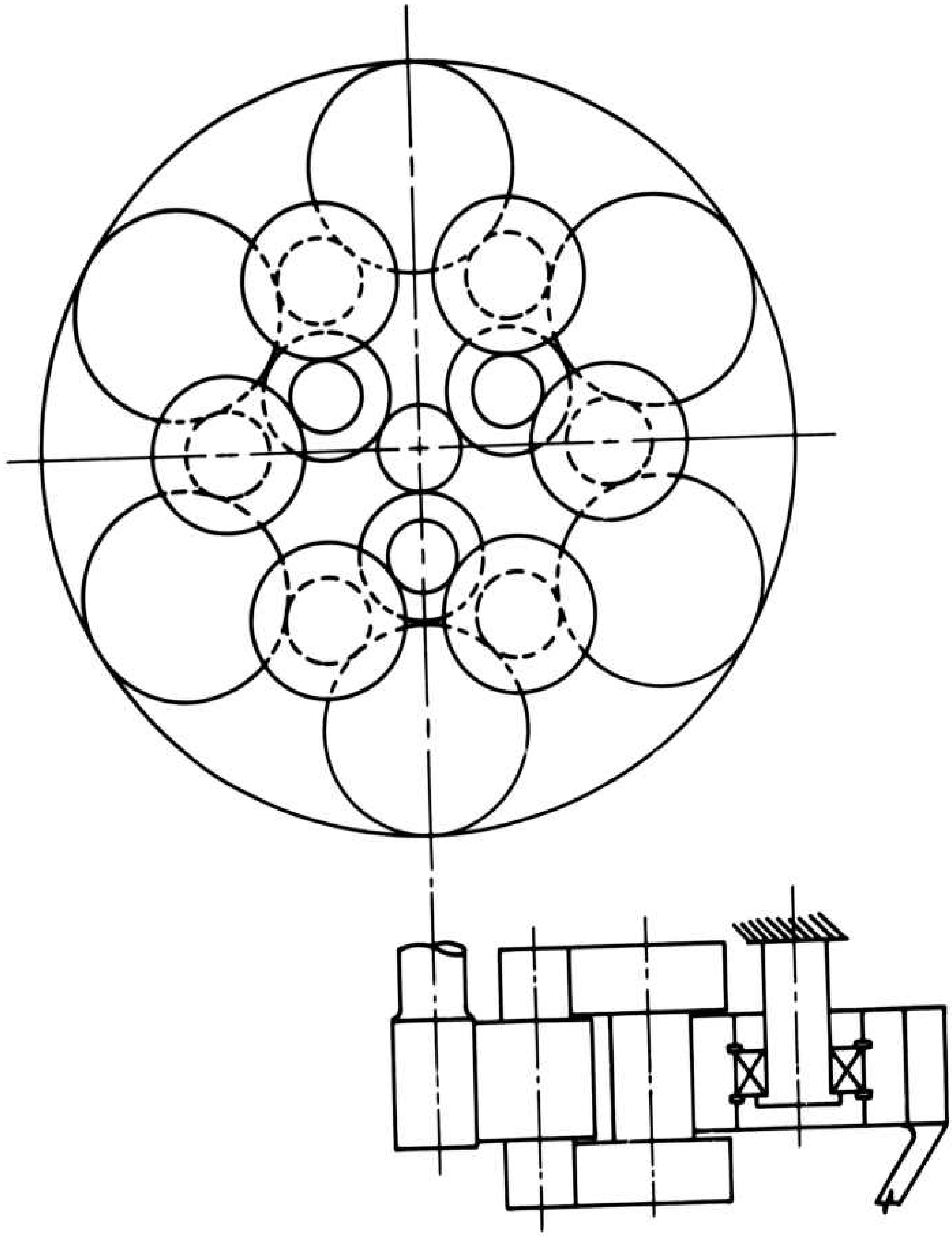


Figure 6. NUOS Roller Friction Drive Schematic.

Note that there are bearings only in the last row planets. Thus, the 10-horsepower accessory is being driven by a planet containing no bearings. This really brings to light one of the main features of the drive - rollers do the job of bearings. The planets are located precisely in their positions by their three line contact. They do not contain placement misalignments inherent in planets located on bearings. The rollers hold them parallel. Spherical roller bearings are used to eliminate possible distortion of the roller cluster by reaction forces on the bearing shaft.

Bearings are placed in the last row planets for two reasons: to take the torque reaction and to provide stability. A stability study showed that the outer row planets lose preload when they are not in an equally spaced arrangement. The drive capacity is limited by preload; therefore, it is necessary to assure the geometrically symmetrical position to maintain the maximum preload. This can be done by other methods, but the simplest is to place bearings in the last row planets and mount them in a symmetrical position.

To get a balanced design, the stepped roller has three rolling planes: two of equal diameter on each side of the planet, and one of a different diameter in the middle.

The roller friction drive is unique in that it has flexibility that allows a thermal differential without significantly affecting the preload. The less loaded internal contacts must move a larger radial distance than the heavily loaded ring contacts to satisfy the energy equations. Therefore, the sun roller can expand due to relatively higher temperature, resulting only in a slight increase in preload. That is, the ring appears to the sun roller to be more flexible than it actually is.

The assembled NUOS roller friction drive is preloaded by the use of a smaller than required ring roller. This leaves it susceptible to contact deformation if the load is high enough and the drive has not been rotated for a long time. There are two apparent means of preloading only during torque application. One is to incorporate a ball screw (making the sun roller and first row roller input conical in shape) in the sun roller; the other is to devise a method of moving the last row rollers out of position and forcing them to approach the geometrically symmetrical position proportionately to the transmitted load increase.

The main advantages of the roller friction drive over the conventional or simple planetary friction drives are: (1) elimination of bearings on all planets except one row where bearings are necessary to transmit the reaction torque to the frame; (2) forcible parallel alignment of all elements within manufacturing accuracy; (3) high efficiency due to exact parallelism and matched-roller preload due to toggle action phenomena; (4) high flexibility of preloaded roller cluster, guaranteeing low sensibility to differential thermal expansion; and (5) low cost due to inexpensive rollers. The roller friction drive is very compact and lightweight for high speed input and moderate torque output. It can also be miniaturized to achieve sizes and very high rotational speeds which until now were not possible. Higher power levels and slower output rotation with corresponding very high torques require very large and long rollers and a heavy supporting ring cylinder to absorb the bending load due to preload. Thus, the roller friction drive is best suited for the low horsepower and high speed applications.

Roller Gear Drive Concept

The roller gear drive evolved from the roller friction drive. It retains some beneficial features of the roller friction drive while increasing efficiency and decreasing weight and size for higher power and torque applications.

In the roller gear drive the kinematic stability is provided by the rollers, and the power is transmitted by the gears.

The high preload required in the roller friction drive is eliminated by the incorporation of gear teeth on all rolling contacts. Figure 7 is a photograph of a roller gear planet. This configuration aligns the rolling surfaces with the pitch diameters of the gears. With such a system, much higher torques can be carried with the same planet width. The only function of the preload is to prevent separation at the contacts.

The main desirable features retained from the roller friction drive concept are: (1) elimination of bearings in all planets except in one row where they are necessary to transmit the drive reaction torque to the frame; (2) forced parallel alignment of all elements within manufacturing accuracy of the rollers because each roller is preloaded by three and in some cases four line contacts almost equally distributed around the circumference; and (3) high efficiency due to both of the above.

In some cases the resultant force on the bearings, made up of the torque reaction and the tooth separation forces, has a component in the inward direction. In this case the drive is self-preloading and the bearing serves only to transmit the torque reaction.

In both cases the resultant force has a component in the outward direction. Since the rollers are usually omitted from the ring gear, the bearing transmits the torque reaction plus a preload force. In most cases the amount of preload force necessary is small; thus, the resultant force transmitted through the bearing is not appreciably increased over the torque reaction force.

When compared to the conventional planetary drive, the roller gear drive offers several advantages. These advantages result from the unique design features noted below.

1. The elimination of bearings in all but the planets in the last row and the substitution of rolling surfaces present the following advantages.

    a. Allows the roller gear drive to provide for the use of the smallest gears compatible with load carrying ability, and the roller contacts provide an extremely efficient support system for the gear meshes. Thus, it is possible to achieve an extremely high reduction ratio in one plane without the limitations of conventional antifriction bearings upon drive arrangement. Subsequently it yields a very low weight drive design because of a minimum of support structure.

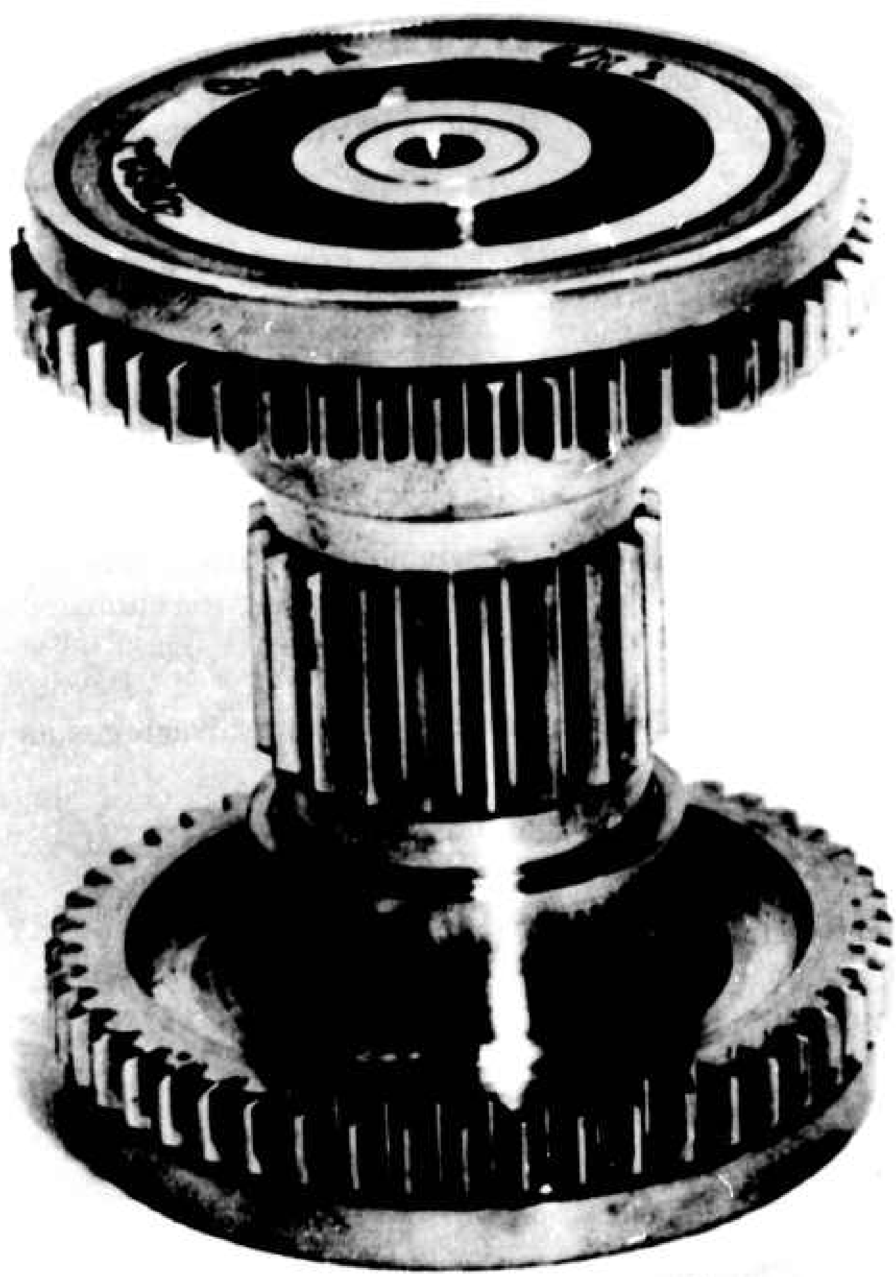


Figure 7. Roller Gear Stepped Planet.

b. Eliminates the centrifugal forces due to their high speed rotation and makes the roller gear drive design one of an inherently long life.

c. Permits the use of wider gears due to good alignment.

d. Offers a weight savings because

(1) Some of the gears have two input and/or output contacts. These gears work twice, resulting in a total reduction of weight.

(2) The total weight of the extra rolling surfaces on the gear cylinders is less than the corresponding bearing weight, one reason being that the rolling contact has a shorter axial length than the antifriction bearing.

e. Prevents the need for structural support except for planets in the last row where bearings are necessary.

2. The gear contact is always within 0.0002 to 0.0004 inch on the theoretical pitch line. This presents the possibility of using gear profiles other than in the involute form.

3. Each planet is very rigid with no appreciable deflection. This is attributed to the short axial length and balanced design allowing for balanced bending moments and less torsional vibration.

In summary, the roller gear drive renders the following advantages as compared to the conventional planetary drive:

1. Smaller elements

2. Efficient support system

3. High reduction ratio

4. Less weight

5. Longer life

6. Contact on the pitch line and parallel alignment

7. High efficiency due to perfect tooth contact; lower total number of contacts in the line or power flow (in series, disregarding parallel contacts); and lower windage losses because of smaller elements.

8. Less vibration

9. Load distribution

TRW built two roller gear drives for back-to-back testing to prove the feasibility of the concept. The drive, which throughout this report will be referred to as the AN-1 drive, was designed for 300 horsepower and a sun gear input speed of 42,000 rpm.

Figure 8 shows this drive as assembled with the radial cover removed. As can be seen, the drive has two planes, one the roller gear cluster plane and the other a conventional gear mesh with a bull output gear.

Details of the key parts are shown in Figure 9. In the lower right corner is the input or sun roller gear. The rolling contact surfaces of this part ride against the corresponding outer roller surfaces of the stepped roller gear or first row planet, in the upper left corner. The smaller, center roller gear on this part then engages the larger second row planet shown at the lower left of the figure. Note the integral pinion on this planet which feeds the bull output gear. The bull output gear is shown at the top right.

MULTICONTACT GEAR REQUIREMENTS

There are four limiting factors in the design of all roller gear clusters for drives. They are parallelism, indexing, location of last row rollers, and tooth number relationships. The first three affect equal load distribution, and the last one is a requirement for assembly.

Parallelism

The stepped-planet design of the roller gear drive planets requires that each planet be made up of two equal diameter roller gear elements on each end and one of a different diameter in the middle. The parallelism of corresponding teeth of each end element must be held very closely.

When the two end elements are the larger ones, the planet is assembled and they are finish ground as assembled. Then this assembly must be used as an integral unit in the final assembly. This procedure results in practically no error. The AN-1 drive first row planets were manufactured like this with no detectable error.

When the two end elements are the smaller ones, the finish grinding must be done with careful attention to the proper indexing of one end element to the other. The error involved here can be held to within 0.0001 to 0.0002 inch in the circumferential direction on the pitch line.

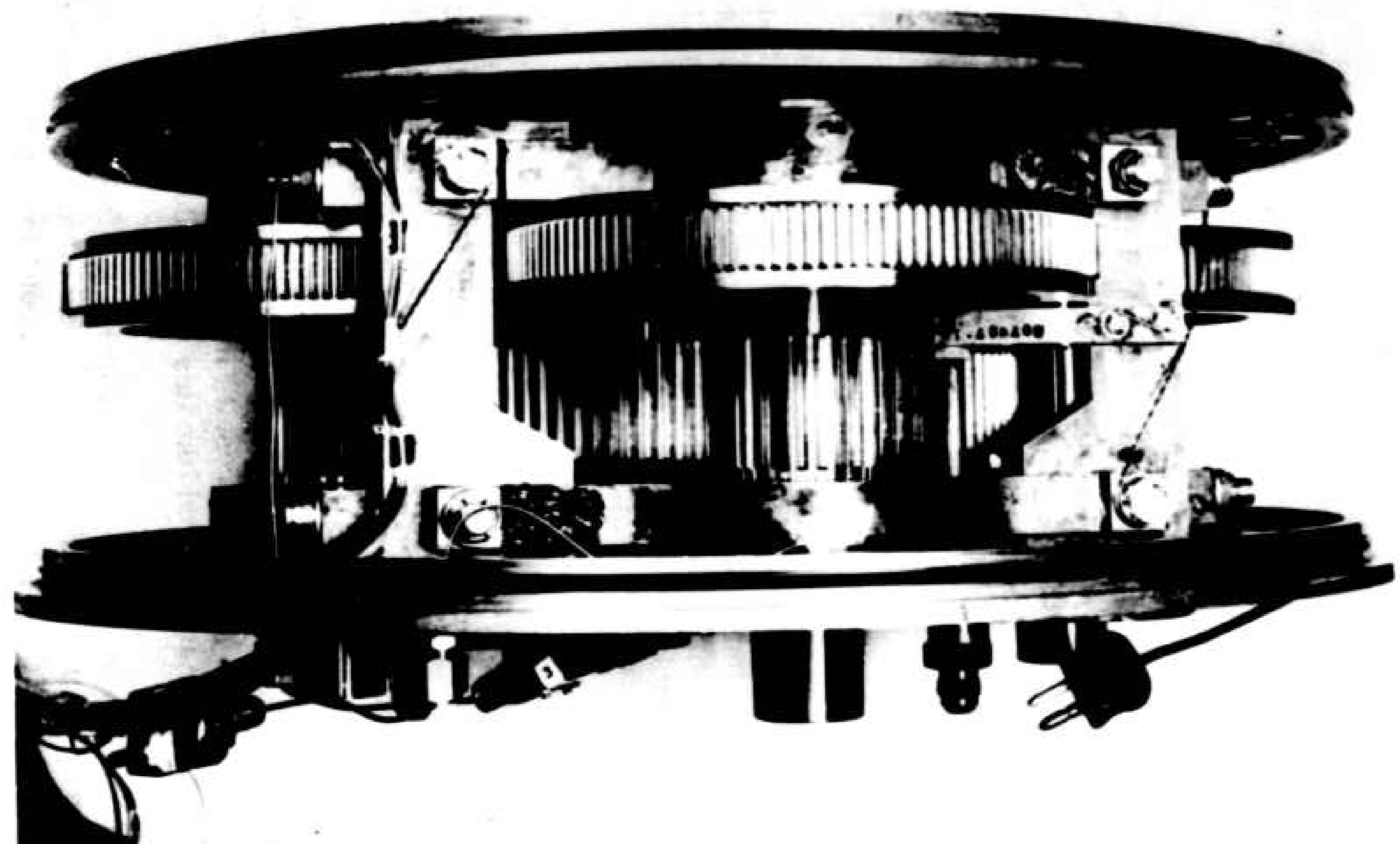


Figure 8. TRW's AN-1 Test Drive.

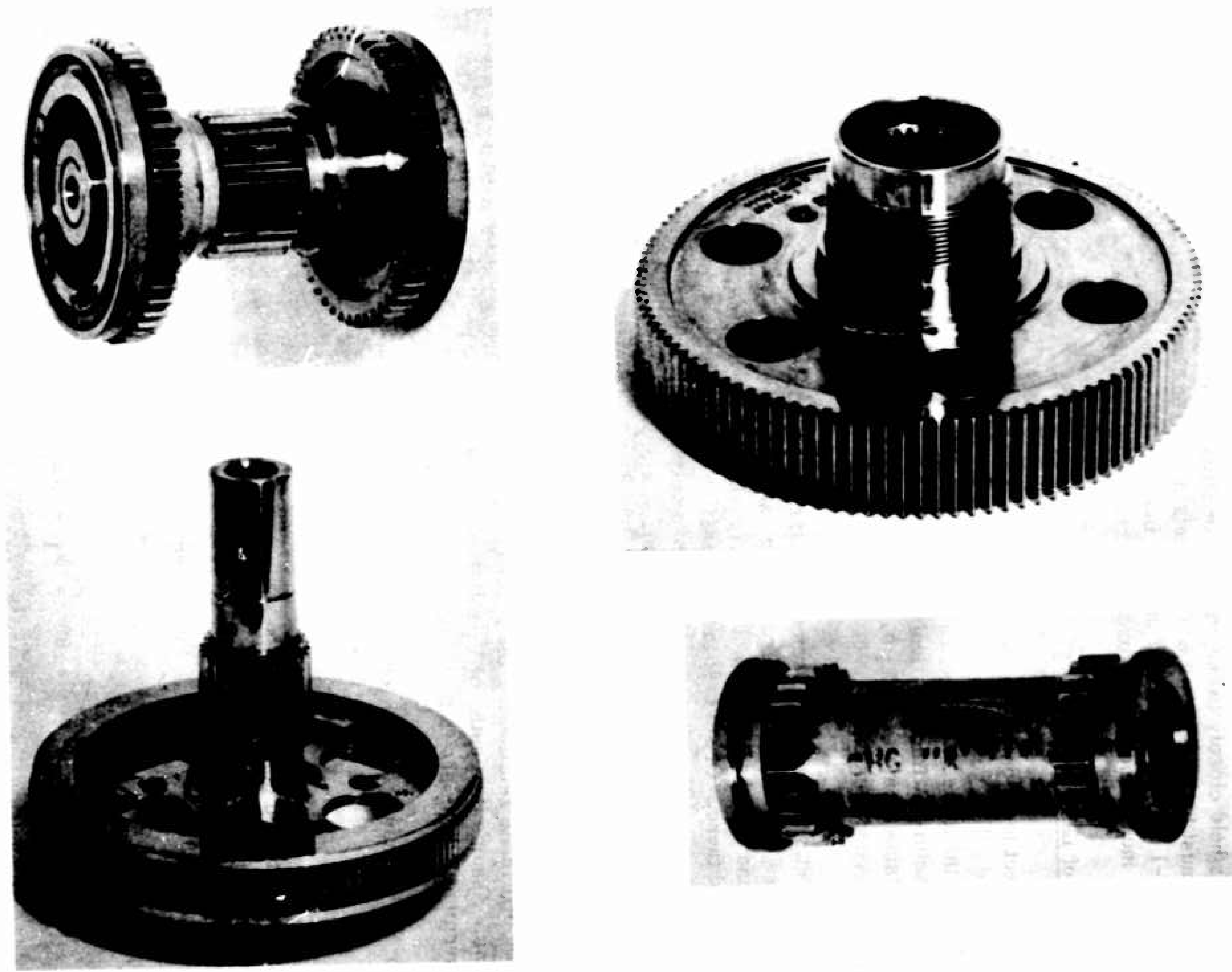


Figure 9. AN-1 Roller Gear Drive Components.

Indexing

The other stepped planet requirement is that the indexing of the larger, or input, element is held closely to the smaller, or output, element. The same indexing relation must be held for all of the planets in a given row. During the manufacture of the AN-1 drive it was found that this indexing error can be held to within 0.0001 to 0.0002 inch in the circumferential direction on the pitch line.

Location of Last Row Rollers

The correct location of the gear centers is a different type of problem. For an ideal assembled roller gear cluster, all angles and dimensions are symmetrical about the center of the drive. Theoretically, this should be attainable because the rollers can be manufactured without difficulty to very close tolerances, closer than ± 0.0001 inch on the outside diameter and ± 0.0002 inch concentric from one size element to its mating stepped element.

The theoretically perfect assembled roller gear cluster cannot be attained without restraints. This was discovered during assembly of the NUOS roller friction drive. It was found that the cluster would not remain in the geometrical position without being held there. Of several methods available, the best was to put bearings in all of the last row rollers (originally they had only the three that were theoretically required to transmit the torque reaction). Later, a stability study showed that if the last row rollers were not symmetrically located, the cluster could be contained within a smaller ring. It is this same stability problem that makes it imperative to hold the location of the last row planets within set limits.

An approximate relation for the required limit can be found from the following calculations. Assume the center of Y_1 roller is moved to a new position with an angular movement 2 dθ as shown in Figure 10.

For the symmetrical position, the angle relation equation is

$$180° = 90° + \alpha_1 + \gamma_1 + \theta. \qquad (1)$$

For the distorted condition caused by moving planet number X_{1-2} the angular distance 2 dθ, equation (1), from purely geometrical considerations, becomes

$$180° = 90° + \alpha_1 + d\alpha_1 + \gamma_1 - d\gamma_1 + \theta + d\theta - 2\,d$$

or, simplifying,

$$90° = \alpha_1 + d\alpha_1 + \gamma_1 - d\gamma_1 + \theta - d\theta. \qquad (2)$$

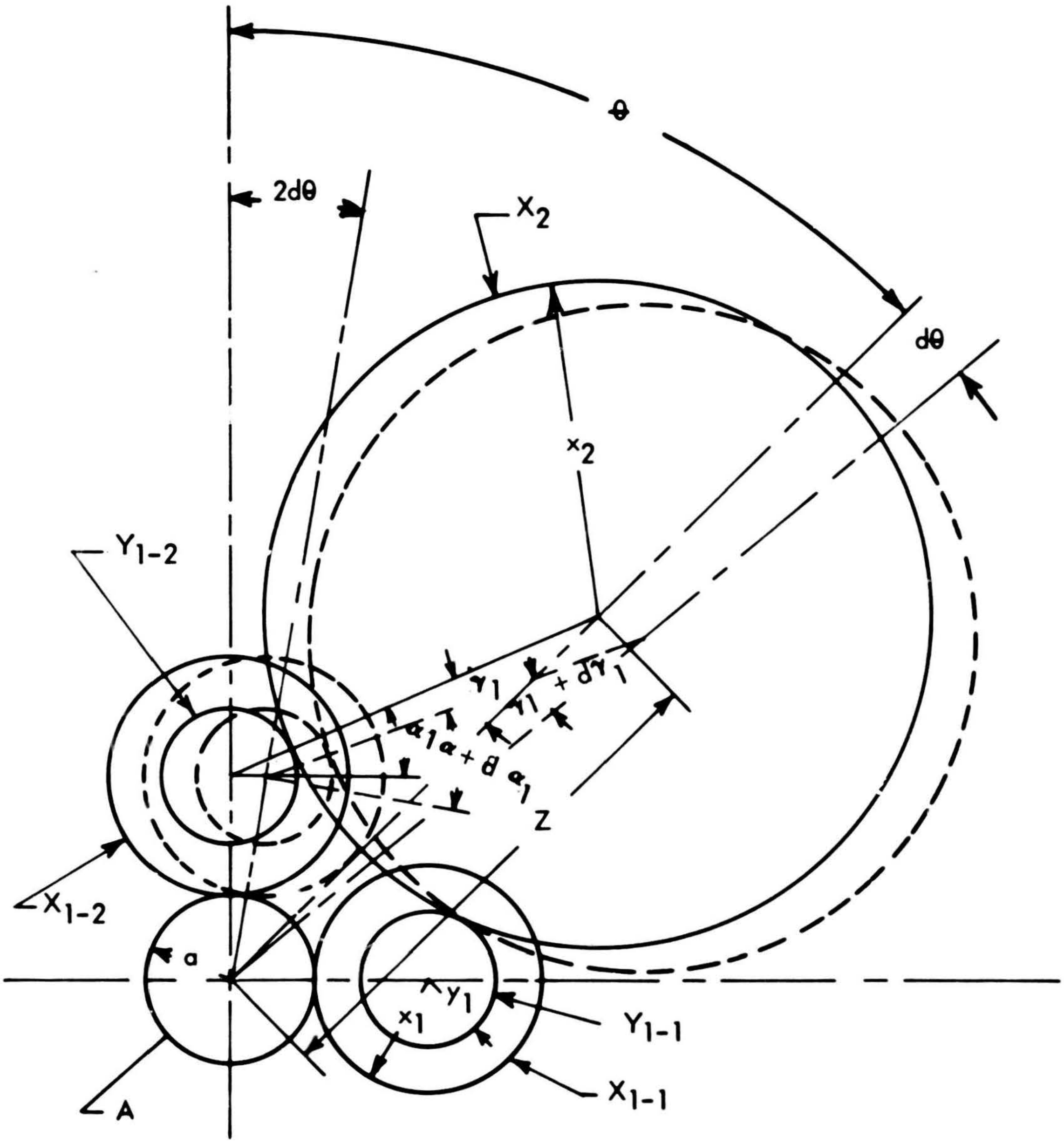


Figure 10. Planet Location Parameters.

Subtracting equation (1) from (2) yields

$$0 = d\alpha_1 - d\gamma_1 - d\theta. \tag{3}$$

The assumed movement of planet number X_{1-2} will introduce a movement of the contact point at the pitch line on the sun gear of

$$s = 2\,a d\theta. \tag{4}$$

To move this arc length (s), roller Y_{1-2} will rotate an angle $d\beta$ (it must rotate because of tooth contact).

$$d\beta = \frac{2a}{x_1} d\theta$$

This rotation will diminish the $(y_1 - x_2)$ contact error caused by the angle $(2\,d\theta)$ change. Thus, it must be added to the right side of equation (3).

$$0° = d\alpha_1 - d\gamma_1 - d\theta + d\beta$$

The remaining error is

$$d\alpha_1 = d\gamma_1 + d\theta - d\beta; \tag{3a}$$

or, substituting in the value of $d\beta$,

$$d\alpha_1 = d\gamma_1 + d\theta - \frac{2a}{x_1} d\theta.$$

From basic geometry,

$$(a + x_1) \sin\theta = (y_1 + x_2) \sin\gamma_1; \tag{5}$$

and after the $2\,d\theta$ angle change,

$$(a + x_1) \sin(\theta - d\theta) = (y_1 + x_2) \sin(\gamma_1 - d\gamma_1).$$

Using trigometric identities,

$$\frac{\sin(\theta - d\theta)}{\sin(\gamma_1 - d\gamma_1)} = \frac{y_1 + x_2}{a + x_1} = \frac{\sin\theta \cos d\theta - \cos\theta \sin d\theta}{\sin\gamma_1 \cos d\gamma_1 - \cos\gamma_1 \sin d\gamma_1}.$$

Using the approximations that cosines of small angles equal 1, sines of small angles equal the angle, and small changes in an angle are negligible, the above equation becomes

$$\frac{\sin\theta}{\sin\gamma_1} = \frac{\sin\theta - (\cos\theta)\ d\theta}{\sin\gamma_1 - (\cos\gamma_1)\ d\gamma_1} .$$

After multiplying out, cancelling terms, and rearranging,

$$\frac{\sin\theta}{\sin\gamma_1} = \frac{d\theta\cos\theta}{d\gamma_1\cos\gamma_1}$$

or

$$\frac{d\theta}{d\gamma_1} = \frac{\dfrac{\sin\theta}{\cos\theta}}{\dfrac{\sin\gamma_1}{\cos\gamma_1}} = \frac{\tan\theta}{\tan\gamma_1} .$$

Thus,

$$d\gamma_1 = \frac{\tan\gamma_1}{\tan\theta}\ d\theta. \qquad (6)$$

Equation (3) becomes

$$d\alpha_1 = \frac{\tan\gamma_1}{\tan\theta}\ d\theta + d\theta - \frac{2a}{x_1}\ d\theta$$

$$d\alpha_1 = d\theta\left(\frac{\tan\gamma_1}{\tan\theta} + 1 - \frac{2a}{x_1}\right).$$

If the location of the center of X_2 has a circumferential error of 0.001 inch, the angular movement is

$$d\theta = \frac{0.001}{z}$$

and equation (3a) becomes

$$d\alpha_1 = \frac{0.001}{z}\left(\frac{\tan\gamma_1}{\tan\theta} + 1 - \frac{2a}{x_1}\right).$$

The contact error on the pitch line is

$$\text{Error} = y_1\, d\alpha_1 = 0.001\left[\frac{y_1}{z}\left(\frac{\tan\gamma_1}{\tan\theta} + 1 - \frac{2a}{x_1}\right)\right]. \tag{7}$$

For the average drive, the value in the brackets is equal to less than 0.1 and the error on the first contact $(a - x_1)$ is equal to less than 0.00015 inch. This means that it is acceptable to locate the last row planets within ± 0.001 inch of the symmetrical position in the circumferential direction. Thus, a practical limit for the distance that the last row planets can be out of location is about 0.001 inch. Of course, this varies with each particular cluster arrangement.

The extent to which the load is affected by this error along with indexing errors can easily be calculated. The maximum error due to indexing and location at any one contact is not more than 0.0003 inch. Divide this by the total contact deflection of 0.0015 inch consisting of tooth bending, tooth contact surface compression, and deflection of the gear segment in the load area. The total error due to deflections now is

$$\text{Error} = \pm\frac{0.0003}{0.0015} = \pm 0.2.$$

This gives a load variation on the tooth of 20 percent. Therefore, all gear contact calculations are made for a 20 percent overload or 1.2 times the load.

Tooth Number Relationships

The last factor in multicontact gear requirements is tooth number relationships. To allow proper assembly, the roller gear drive must satisfy specific tooth number relationships. These relations are a function of the number of planets in a row and the toggle angle α.

In order to assemble cluster it is necessary to have a symmetrical closed curve made by teeth alternately of Y_1 and X_2 gears for two row systems, and/or by teeth alternately of Y_2 and X_3 gears for three row systems, which contains a whole number of teeth. A two row system along with the symbols used in the following derivations is shown in Figure 11.

The figure shows that the closed curve portion of the Y_1 gear has a ratio of $180 - 2\alpha_1/360$ with respect to the total number of teeth in the Y_1 gear. For the X_2 gear the ratio is $2\gamma_1/360$. Fractional number of teeth for the contact arc of either or both gears is permitted if the sum of these arcs for all the gears in both rows is a whole number. It should be noted, however, that in using fractional teeth, assembly and manufacturing problems may be introduced. Difficulties in accurately positioning the gears for assembly of the gearbox will be present when fractions of teeth are in the contact arc.

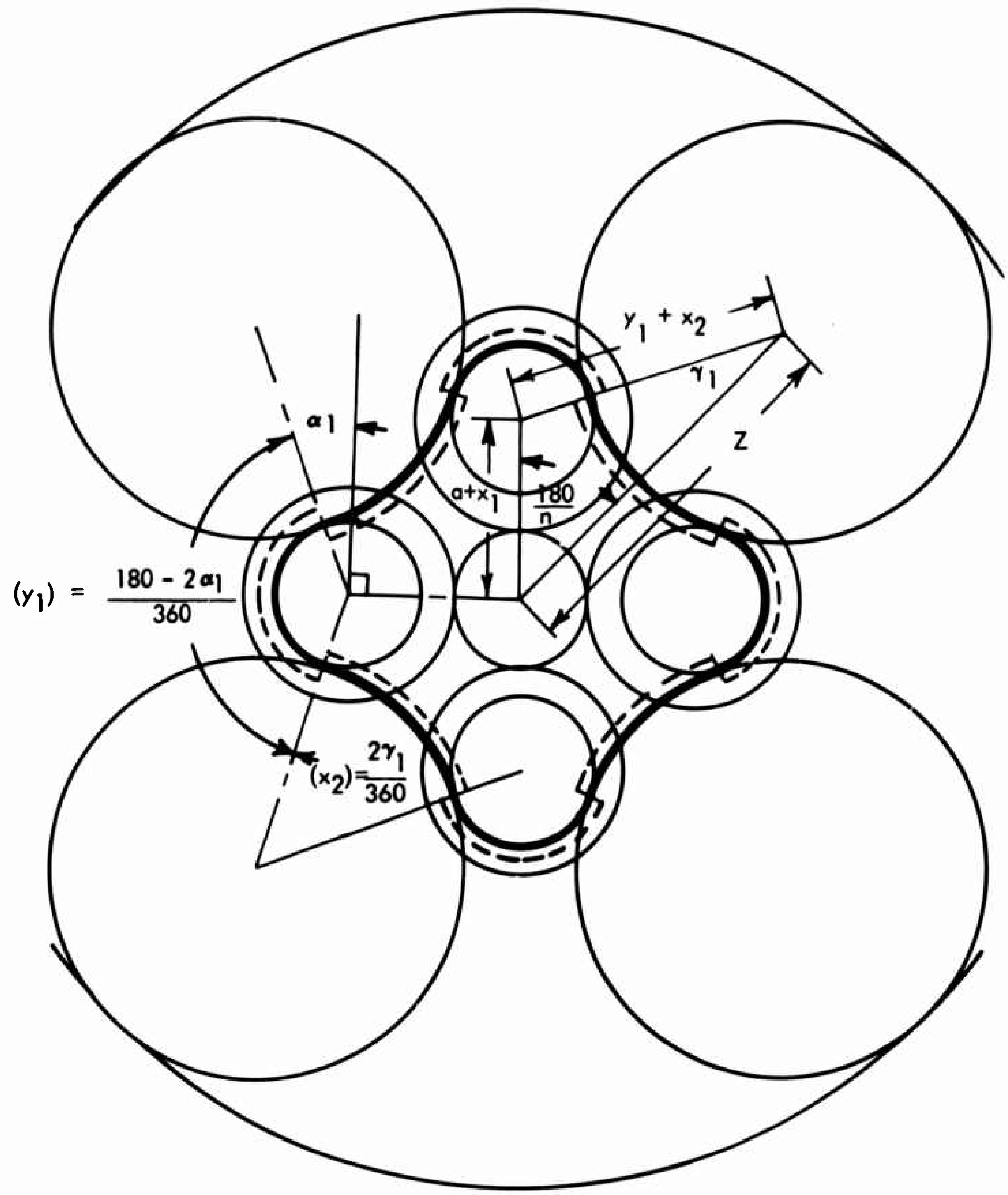


Figure 11. Drive Assembly Requirements and Parameters.

Special indexing may be required between the gears on radii x_1 and y_1 of the inner planets so that these gears will not be interchangeable among themselves. Thus, it is generally preferable to use whole numbers in the arc of contact, whenever possible.

It is also preferable to use a sun gear with a teeth number divisible by the number of first row planets. This allows the use of first row planets with identical indexing. When the sun gear teeth number is not divisible by the number of first row planets, the assembly is complicated by the use of different indexing per each first row planet. The second solution is inconvenient but offers a smoother operation of the sun gear because all teeth do not reach their maximum load at the same instance.

During the parametric study, TRW calculated a set of tables for the arc tooth numbers versus toggle angle for 4, 5, 6 and 8 planets per row. In addition, the basic required geometric relationships were calculated for whole angles.

Teeth numbers and geometric proportions are both a function of toggle angle. The governing equations for teeth numbers are as follows:

$$(y_1) = \frac{90 - \alpha_1}{180} \tag{8}$$

and

$$(x_2) = \frac{\gamma_1}{180}; \tag{9}$$

and for geometric proportions (see Figure 11),

$$(a + x_1) \sin \theta = (y_1 + x_2) \sin \gamma_1 \tag{5}$$

and

$$z_1 = (a + x_1) \cos \theta + (y_1 + x_2) \cos \gamma_1. \tag{10}$$

In addition, for three row systems,

$$(y_2) = \frac{90 - \alpha_2}{180} \tag{8a}$$

$$(x_3) = \frac{\gamma_2}{180} \tag{9a}$$

$$z_1 \sin \theta = (y_2 + x_3) \sin \gamma_2 \tag{5a}$$

and

$$z_2 = z_1 \cos \theta + (y_2 + x_3) \cos \gamma_2. \tag{10a}$$

MINIMUM WEIGHT AND VOLUME DESIGN

There are many considerations in determining the minimum weight and volume design. In an attempt to systematize the parametric analysis, these were narrowed down to the three most important ones: (1) number of planets and rows, (2) output gear selection, and (3) stepped or nonstepped gears in the last row planets.

Number of Planets and Rows

In general it is desirable to keep the number of components in a transmission to a minumum. Thus, it is more desirable to use the minimum number of planets per row and the minimum number of rows.

A drive made up of one row of planets is a simple planetary drive and does not contain the advantages of the TRW cluster arrangement. For this reason it was not considered in the parametric study.

A drive with two rows of planets has a better efficiency than one with three rows of planets. This is because it has only three contacts instead of the four associated with the three rows of planets.

In a drive with a small number of planets per row, the size of the planets in the next row may increase much faster than in the larger number of planets per row drive. This can be seen from basic geometry by using equation (5),

$$(a + x_1) \sin \theta = (y_1 + x_2) \sin \gamma_1, \tag{5}$$

and by writing the angles as a function of the toggle angle (α) and the number of last row planets n.

From Figure 11 it is observed that

$$\theta = \frac{180}{n}$$

and

$$\gamma_1 = 90 - \theta - \alpha_1$$

or

$$\gamma_1 = 90 - \frac{180}{n} - \alpha_1 .$$

Equation (5) becomes, after rearranging,

$$\frac{y_1 + x_2}{a + x_1} = \frac{\sin \frac{180}{n}}{\sin\left(90 - \frac{180}{n} - \alpha_1\right)} .$$

With a decrease in the number of planets in a row, the value of $(y_1 + x_2)$ is larger and vice versa.

For example,

for n equal to 3,

$$\frac{y_1 + x_2}{a + x_1} = \frac{\sin 60}{\sin (30 - \alpha_1)} ;$$

and for n equal to 12,

$$\frac{y_1 + x_2}{a + x_1} = \frac{\sin 15}{\sin (75 - \alpha_1)} .$$

The result of this geometry is a much larger ratio between the input sun gear and the output ring gear in the drive with a smaller planet number. In other words, higher ratios can be obtained by a drive with a smaller number of planets per row.

A peculiarity of the turbine prime mover is that overall dimensions of turbines do not grow in proportion to the horsepower increase. Higher horsepower turbines rotate slower. The sun gear input to the drive from the high horsepower turbine has a higher torque and therefore must have a large outside diameter. The drive design objective is to have the drive outside diameter as close as possible to the turbine dimensions. These two factors yield a fixed ratio between the sun gear and output ring gear dimensions; thus, they force the use of drives with more planets per row for high horsepower.

In general, it was discovered that the 3-3 drive is practical for less than 200 horsepower. Ranges of from 200 to 250 horsepower are practical limits for the 4-4 drive. For higher horsepower and large ratios, the drives using more planets per row and/or three rows of planets were the best. The three row system offers a higher ratio than the two row system for the same space.

In three row drives, two design concepts can be used, those with an equal number of planets per row (type n-n-n) and those with half the number of planets in the first row (type n/2-n-n). The half the number of planets in the first row type yields a higher ratio in the first row and higher total ratio can be obtained within the same drive outside diameter. This design is selected for the majority of optimized drive series.

Output Gear Selection

For minimum weight and volume it is essential to have the most efficient cluster arrangement. In general, there are two main possibilities: rotary planet cluster (rotating spider) and stationary ring gear, or stationary spider and rotary output gear.

For the stationary spider case, the formula for overall ratio is:

$$R_o = \frac{c\, x_i}{a\, y_i} \qquad (11)$$

where i equals 1, 2 . . . (k - 1)
k equals the number of rows.

For the stationary ring gear case, with the spider rotating in the same direction as the input sun rotation, the overall ratio becomes

$$R_o = \frac{c\, x_i}{a\, y_i} + 1; \qquad (12)$$

and for spider rotating in the opposite direction to the input sun rotation, the equation becomes

$$R_o = \frac{c\, x_i}{a\, y_i} - 1. \qquad (13)$$

The above three equations indicate that for a given ratio it is best to have the spider rotating with the sun gear (equation 12), next best to have a stationary spider (equation 11), and least desirable to have the spider rotating opposite the sun gear (equation 13). In addition, it was found that the case of the spider with reverse rotation to the sun input gear (equation 13) has inherent losses due to the higher pitch line velocity with the same forces. This feature decreases efficiency; therefore the designs of this type were considered less practical. The drives of this type are those of an even number of planet rows with fixed output ring gear or an odd number of planet rows with fixed output sun gear. These are illustrated schematically in Figures 12a and 12b.

The drives with the rotating spider in the same direction as the input sun gear (equation 12) are shown in Figures 12c and 12d. They are the fixed output ring gear type with an odd number of planet rows and the fixed output sun gear type with an even number of planet rows.

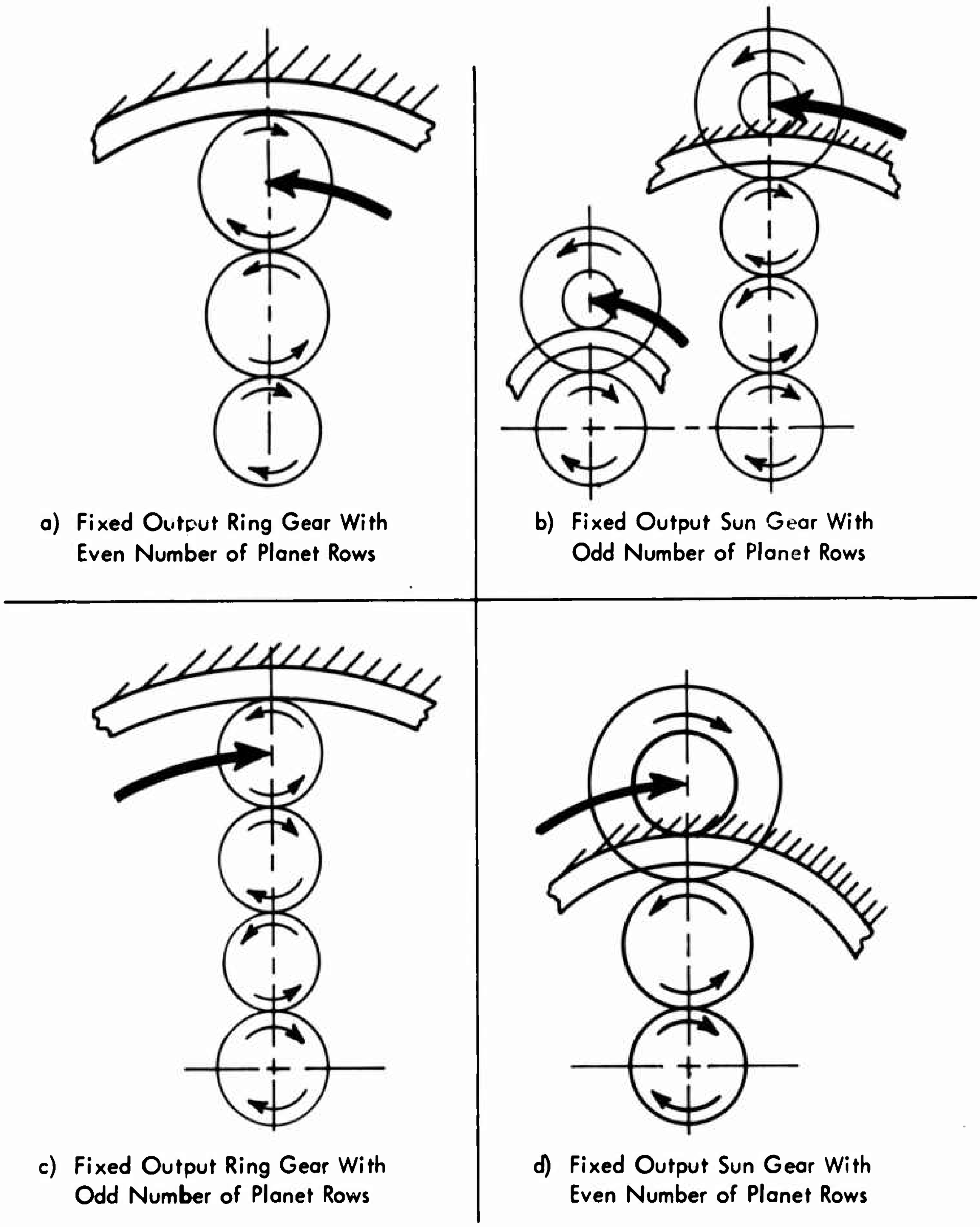


Figure 12. Rotation Direction Concepts.

After a systematic analysis, the stationary type was selected for the basic drive. This type of drive can have all of the arrangements shown in Figure 12, the only change being that the output ring or sun gear rotates and the cluster is anchored by the bearings to the last row planets. The use of the stationary spider drive offers the possibility of using the planets as accessory pads and simplifies the lubrication problems. It is much easier to lubricate through fixed connections than through rotating seals. The stationary spider does not require the close concentricity between the sun and ring gears that is required for the drive with rotating spiders. Thus, the wobbling of the input shaft at the same frequency as the output shaft, caused by concentricity errors inherent in rotating spider drives, is eliminated. Another feature of the stationary spider drive is that the overall ratio is not affected by the number of planet rows and types of output gears.

There are two possible types of output gear for the stationary spider drive: the output ring gear and sun gear type. The output ring gear type was selected for the following reasons:

1. The ring gear can be symmetrically located in the axial direction in relation to the roller gear cluster while the output sun gear must be located to one side of the roller gear cluster. This nonsymmetrical location introduces a bending moment on the last row planet shaft. At least two bearings are required to take the resulting bending moment reaction.

2. The output ring gear design can use a spherical roller bearing inside the last row planet. In this design, the roller gear alignment is not affected by the bearing location and has self-aligning features. Thus, the ring gear design has better efficiency than the output sun gear design which requires two ball or roller bearings for each planet.

3. The output ring gear has a larger ratio for the same pinion size than does the smaller diameter output sun gear. The smaller output sun gear pitch radius results in a higher load for the same torque. Therefore, the axial length of the output pinion is increased with a weight penalty of the last row planets for the output sun gear drive.

4. The output ring gear tends to preload the roller gear cluster owing to inward radially acting gear separation forces, while the output sun gear tends to unload the cluster owing to gear forces acting outwardly. Special preload must be provided through the bearings for the output sun gear type. This preload acts permanently, both while stationary and during use, and can damage the bearings while the drive is not being used.

The output sun gear type offers two advantages over the output ring gear type: last row planets can be used as accessory drives from the output side of the drive, and gear box structure is conveniently located for bearing support. Because the relative advantages between the two favor the output ring gear the output sun gear should be used only for special design objectives.

Stepped or Nonstepped Gear in the Last Row Planet

It is natural to use stepped gears in each row of planets, including the last one, to obtain the maximum ratio. However, for minimum weight and volume, this is not always true. The use of stepped gears in the last row planets with an output ring gear presents structure difficulties. Thus, in most cases the nonstepped last row planet is used.

One of the advantages of the roller gear drive is that the design can be made symmetrical, eliminating any bending moments which tend to distort parallelism of the gear and roller contacts. Further, for the nonstepped last row planet, a spherical roller bearing can be used. The bearing is conveniently located inside of the gear and is mounted on a stud fixed to the gear box wall. The spherical roller bearing does not transmit gear box wall deformations and stud bending.

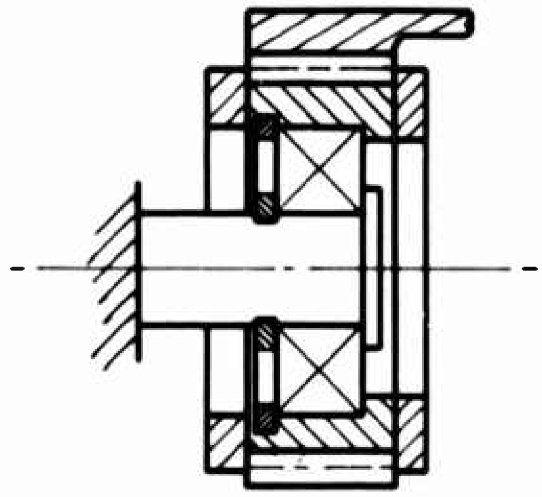
Figures 13 and 14 show basic last row planet design variants. Three of these are nonstepped and four stepped. The three nonstepped variations (Figure 13) have the advantages of the spherical roller bearing while only one variant (Figure 14a) of the stepped type has them. The last three (Figures 14b, 14c, and 14d) would use conventional roller bearings.

The nonstepped last row planet design presents ring gear assembly problems because the roller that mates with the next to last row planet interferes with the ring gear assembly. Three solutions are possible.

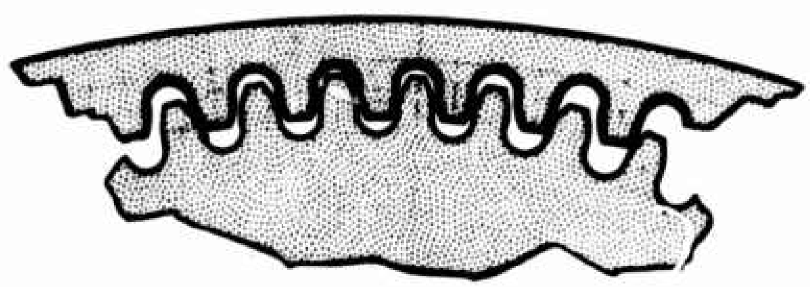
Figure 13a shows the single gear and two rollers on the sides approach. First, the roller gear cluster is assembled with the right side roller off. Then, after the ring gear is axially slidden into position, the roller is put into position and fastened to the gear. This approach appears to be one of the best solutions.

A unique solution is shown in Figure 13b. The addendum is removed from the ring gear, permitting the ring gear to be assembled axially with the roller already on the last row planet. For this design, the problem is the strength of the tooth of the gear that transmits the power to the last row planet. This is the smaller gear in the next to last row planet. In an attempt to gain in ratio, this gear must have as few teeth as possible. To obtain a contact ratio greater than one while not undercutting the tooth, it is necessary to use a large number of teeth and a low pressure angle. The low pressure angle reduces the tooth strength. Thus, this design is limited in ratio and tooth strength.

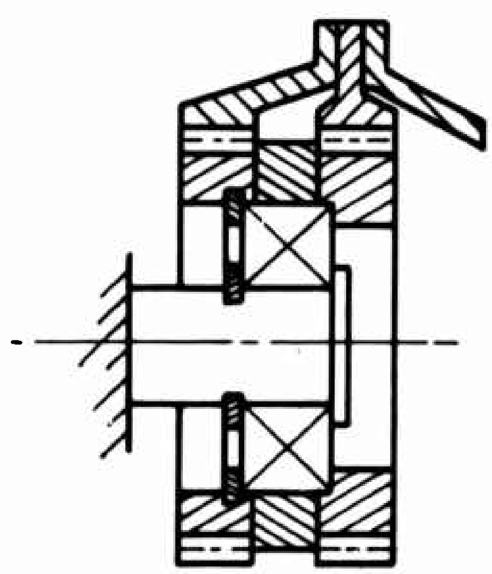
The third possible solution is shown in Figure 13c. In this arrangement, the roller is between a two-part last row gear. The ring gear can be assembled from both sides, bolted together, and positively located by dowel pins. This type of design results in inferior parallel alignment of the last row gears.



a. Nonstepped Output Concept

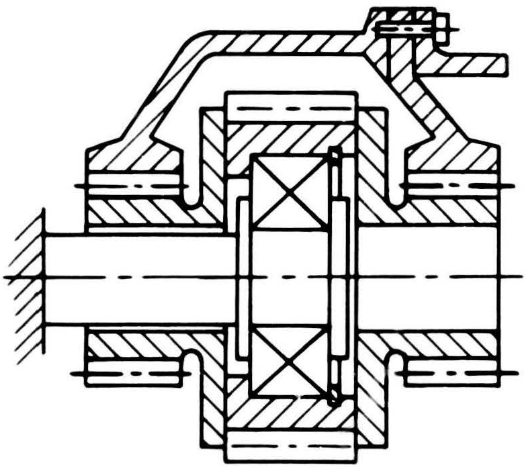


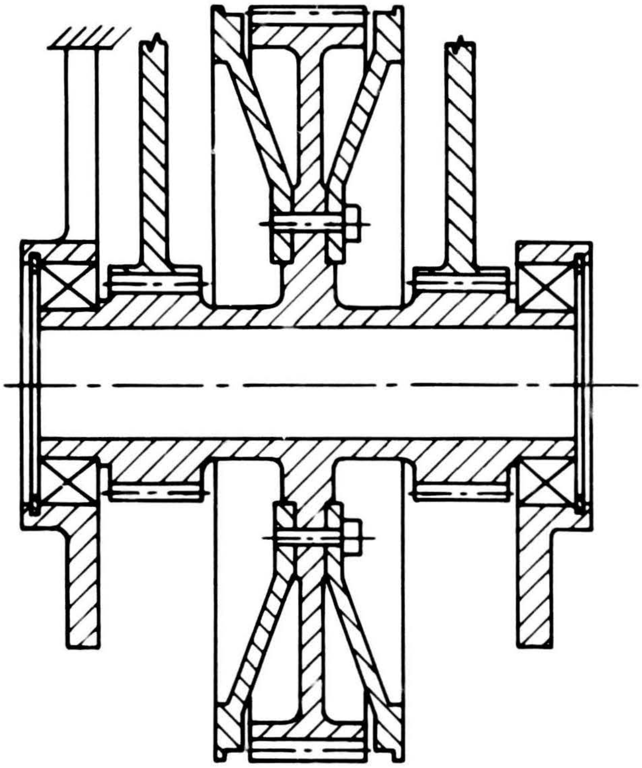
b. Half Toothed Ring Gear Concept



c. Center Roller Concept

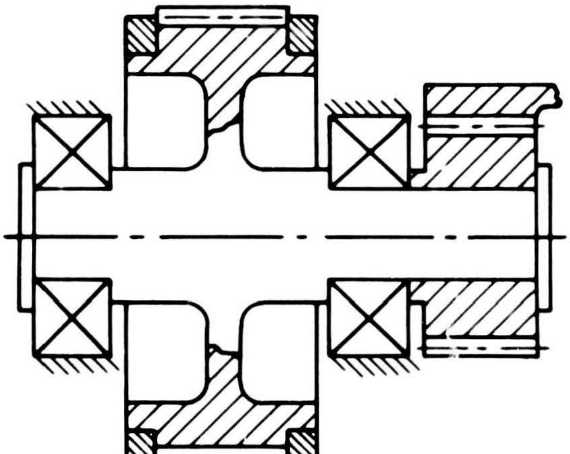
Figure 13. Nonstepped Last Row Planet Design Variants.

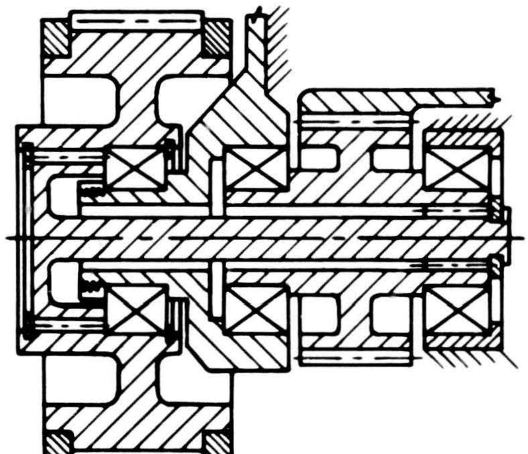




a. Stepped Output, One Bearing Concept

b. Stepped Output, Two Bearing Concept





c. Stepped Unbalanced Output, Two Bearing Concept

d. Stepped Unbalanced Output, Three Bearing Concept

Figure 14. Stepped Last Row Planet Design Variants.

The stepped last row planet design variants are shown in Figure 14. These designs do not have the last row roller interference with the ring gear assembly problem. However, they do have the following disadvantages:

The stepped output, one bearing concept shown in Figure 14a has a large overhung moment on the bearing stud owing to the long distance between the bearing and the mounting wall. A possible means of decreasing the stress caused by the moment is to put a support ring on the right side of the cluster assembly, extend the bearing shaft through to it, and fasten the shaft rigidly to it. This adds weight as well as axial length to the design; therefore, it should be used only in special designs. Other design problems for this arrangement occur. The size of the last row output gear which mates with the ring gear must be large enough to allow the insertion of an adequately sized bearing stud shaft. The output pinion reaction forces are transmitted to the gear body and bearings as plate trunnion bending because the planet assembly is a box type. Transmitting bending stresses through a trunnion plate is very inefficient. It results in a larger weight than other designs for the last row planet. Also, the ring gear is split and has a "U" shape to allow for clearance of the last row planet. This presents alignment problems and adds weight. The space between the trunnion plate walls and the smaller gears must be large enough to allow finish grinding of the teeth. This increases the bending moment.

The disadvantages of the stepped output, two bearing concept shown in Figure 14b are outweighted by the advantages. This concept only recently evolved from trying to solve the problems presented by other designs. In this design the roller bearings must be closely located so as to prevent loss of parallelism. This can be done by line boring the bearing mounts at assembly. There is no trunnion bending as in the previous design. The last row output gear that mates with the ring gear can be made as small as the tooth bending stresses will allow because the bearings are supported exterior to the shaft. The shaft can be lightweight because the large forces are near the bearing, thus creating smaller bending moments. There is adequate space to cut and grind the gear teeth. The split ring gear alignment problem still exists in this design. Another disadvantage is the requirement for support structure that must pass through the cluster, thus adding weight. This limits the use of this type of design. However, the design represents a considerable improvement over the stepped output, one bearing concept. Its main use is in the design with a smaller number of planets per row.

The disadvantages of the stepped unbalanced output, two bearing concept shown in Figure 14c are that uneven loads exist between the bearings, and reaction deflections are transmitted to the roller cluster contacts. It is hard, if not impossible, to match the uneven preload required by the bearings on each side of the planet because the requirement changes with output speed and horsepower. This was discovered during testing of the AN-1 roller gear drive. In that drive the right-hand bearing was outside the sun output gear. This aggravated the bending problem. The solution for the AN-1 drive was to make the bearing supports very rigid and to locate them to assure an adequate preload for all operating conditions.

The disadvantages of the stepped unbalanced output, three bearing concept shown in Figure 14d are, as in the design in Figure 14b, outweighed by the advantages. Similarly, this design evolved from attempts to improve the stepped unbalanced output, two bearing concept. The bending moments associated with the ring gear contact are not transmitted to the roller cluster. Rather, they are taken out through the roller bearings which are symmetrically located with respect to the ring gear contact. The roller gear cluster is connected to the conventional gear contact at the ring gear by a flexible spline shaft. The shaft compensates for any misalignment or indexing error. Thus, load equalization is attained through its use. The disadvantages of this drive are its long axial length and the use of three bearings per last row planet.

Another method of obtaining minimum weight and volume is to stagger the input gears of the first row planets. The maximum size of the planets is limited by the geometry of the cluster arrangement, and the axial staggering of these gears with respect to each other eliminates this limitation. By staggering these gears, the drive weight is increased slightly owing to the increased axial length. The ratio is also increased for a given outside diameter.

In summary, the minimum weight and volume design incorporates: (1) the minimum number of planets per row and the minimum number of rows consistent with the objectives of each design; (2) a stationary spider to allow the use of the planets as accessory pads; and (3) a nonstepped last row planet concept.

DISCUSSION

The designs studied were assumed to have a stationary spider, a nonstepped last row planet incorporating a spherical roller bearing as in the design in Figure 13a, and an outside diameter depending upon horsepower.

Because of the very broad scope of the investigations of the drive application to helicopters and because each helicopter has a different arrangement, the study was restricted to the basic drive, omitting suspension bearings, one-directional clutches, accessory pads, tail rotors, intermediate reductions, right-angle drive, etc. The drive design types were checked to ascertain whether there were any significant obstacles that would discourage their use in helicopters.

RATIO LIMITS FOR VARIOUS TYPES OF DRIVES

The results of the study of the maximum ratio obtainable versus horsepower at given outside diameters are shown in Figures 15 through 21. The establishment of limits of ratio capacity for various roller-gear configurations offers a comparative basis for selection of a specific configuration for a design of given horsepower, ratio, and envelope. The data presented in these figures are based upon the given reduction from the typical turbine rpm. They are meant to show the attainable ratios for a given horsepower relative to each configuration type.

The typical turbine rpm was taken from the curve presented in Figure 22. This curve, along with the one in Figure 23 representing typical rotor rpm, was received from the U. S. Army Transportation Research Command. From these curves, the typical drive series was averaged, and the results are shown in Table 1.

The maximum ratios for various configurations of the roller gear drive, using ring gear output, were obtained for the ring gear radii specified in the following table and certain multiples of these radii (1.25c, 1.50c, and 1.75c) (see Table 2).

Requirements that were satisfied in the analysis besides the radius limit were:

1. Bending and surface fatigue life of the sun gear to be 2500 hours.
2. Maximum tooth bending stress to be 30,000 psi.
3. Maximum pitch line velocity to be 16,000 feet per minute.
4. Maximum twist displacement to be 0.0003 inch at the pitch line.
5. Maximum twist angle to be 0.0002 inch per inch of gear length.
6. Minimum toggle angle to be 8 degrees.
7. Minimum clearance between gears to be 0.02 inch.

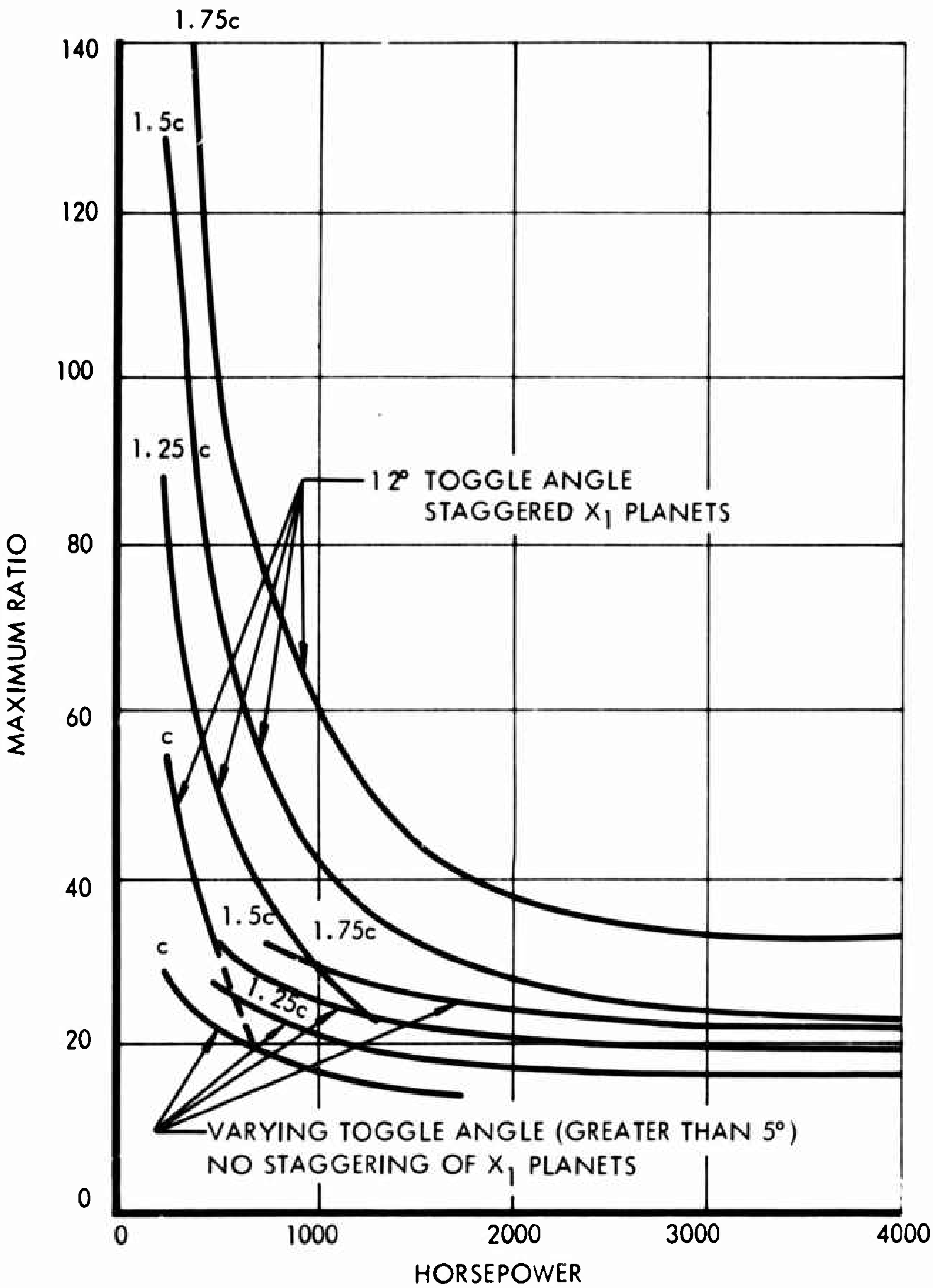


Figure 15. Maximum Ratio Versus Horsepower Curves (4-4 Drive)

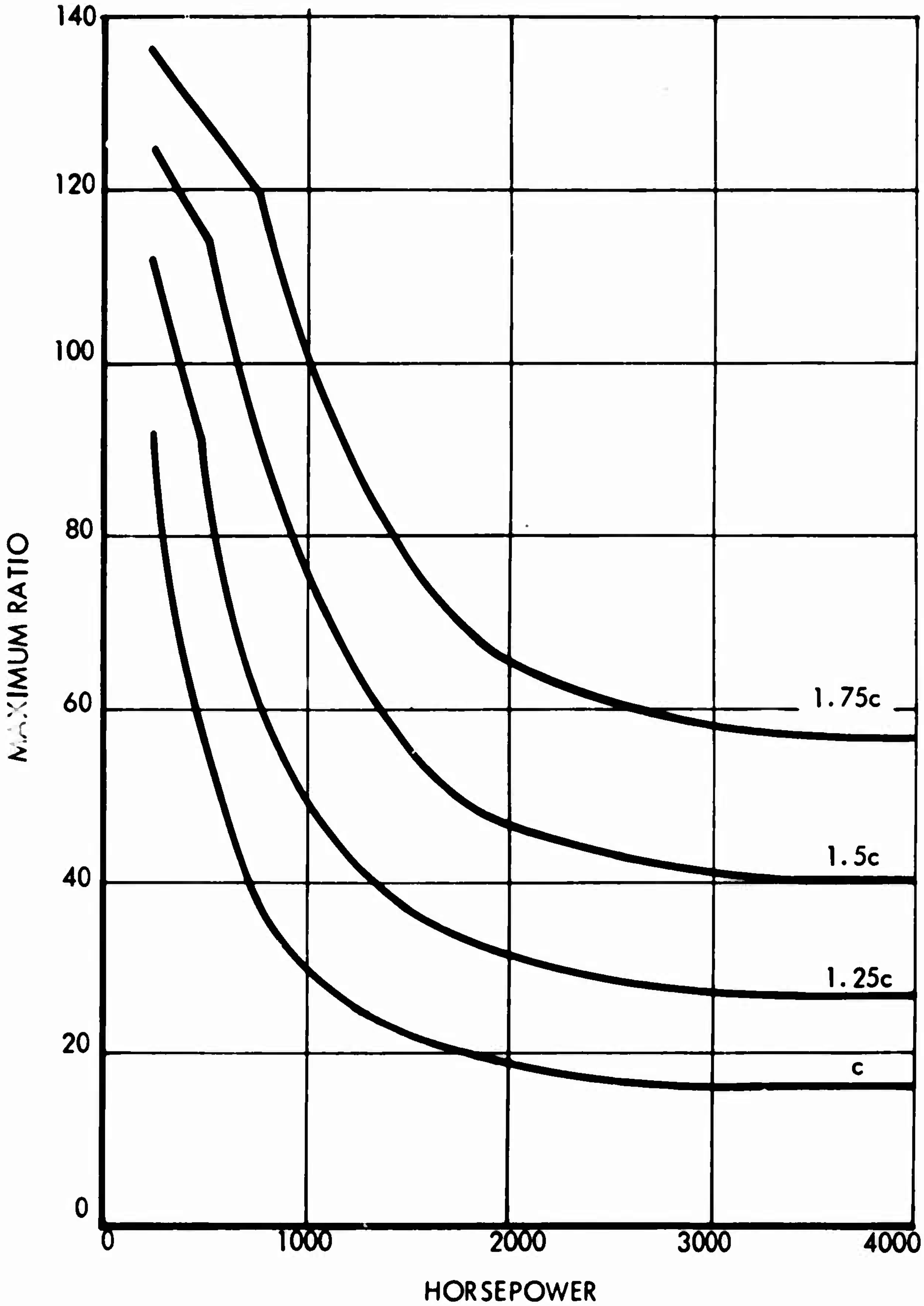


Figure 16. Maximum Ratio Versus Horsepower Curves (6-6 Drive), Staggered X_1

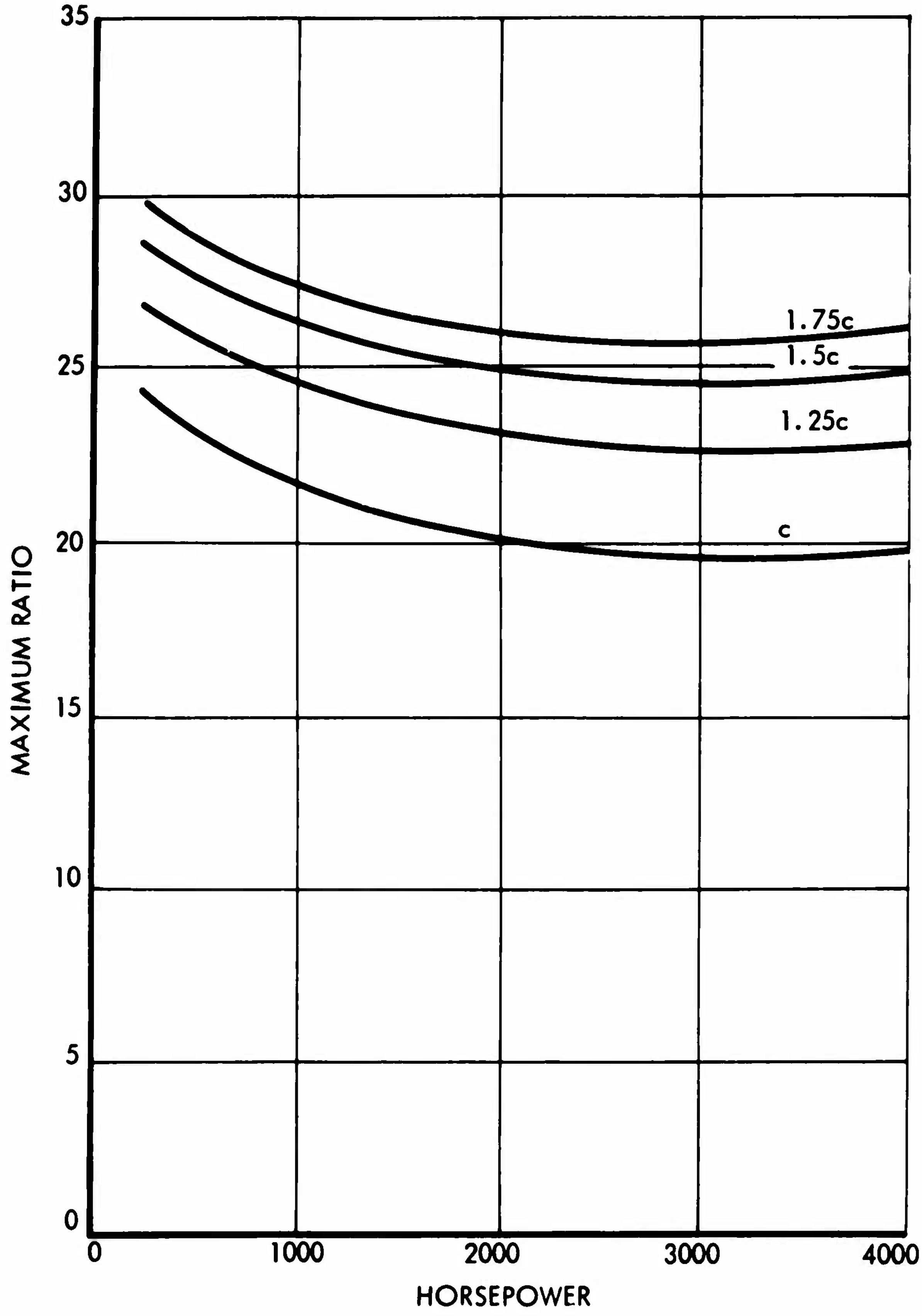


Figure 17. Maximum Ratio Versus Horsepower Curves (8-8 Drive), Staggered X_1.

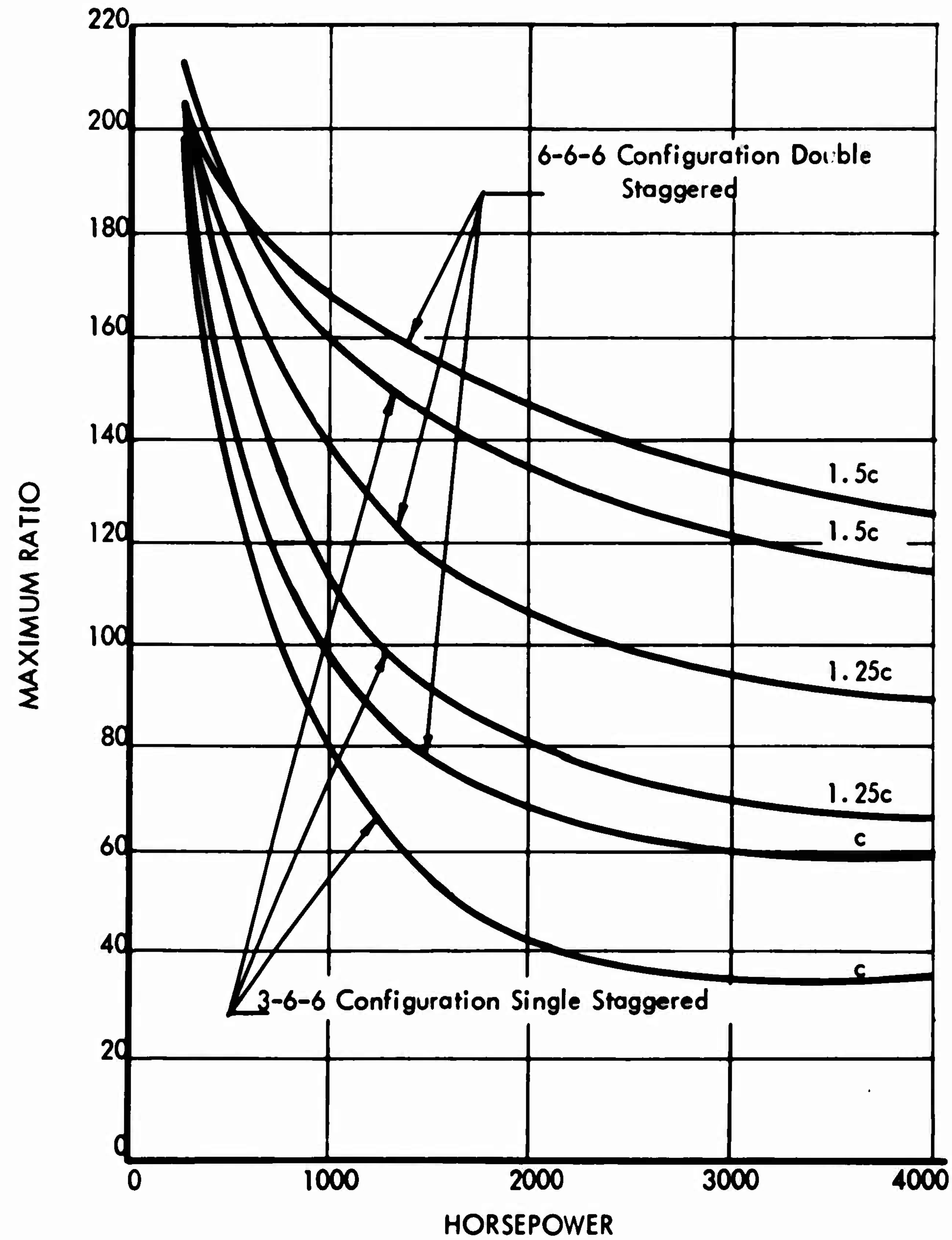


Figure 18. Maximum Ration Versus Horsepower Curves (3-6-6 and 6-6-6 Drive).

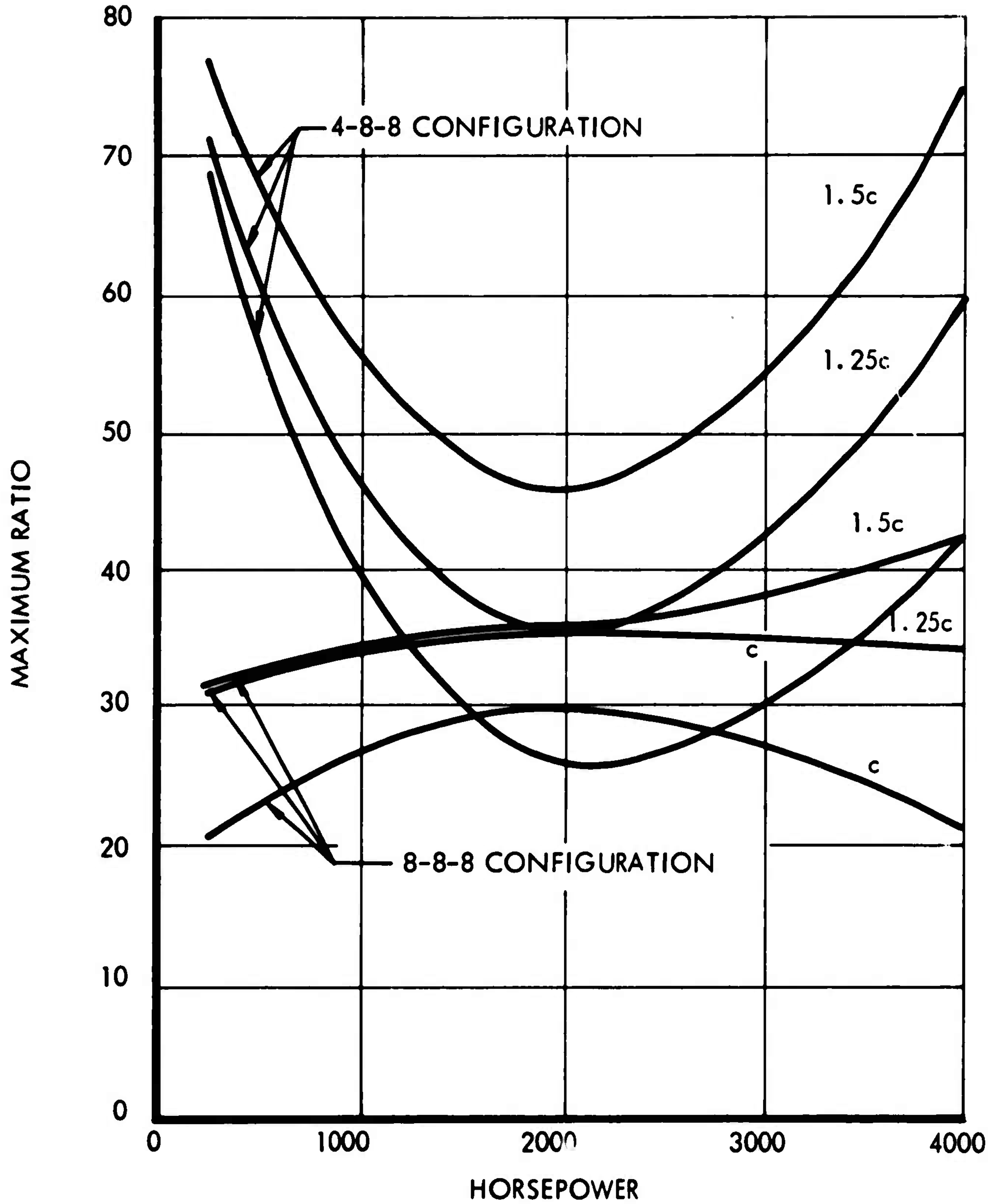


Figure 19. Maximum Ratio Versus Horsepower Curves (4-8-8 and 8-8-8 Drive), No Staggering.

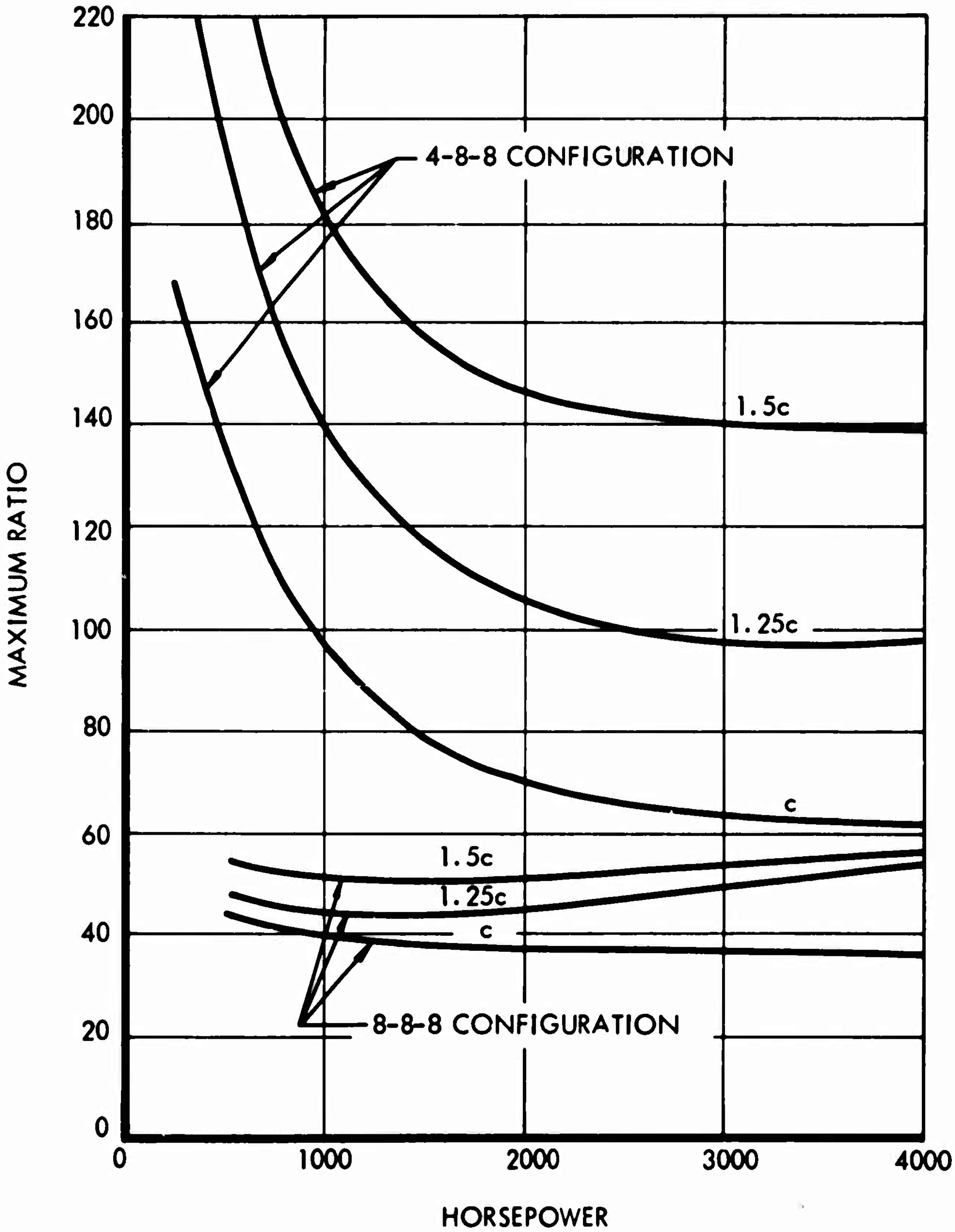


Figure 20. Maximum Ratio Versus Horsepower Curves (4-8-8 and 8-8-8 Drive), Single Staggering.

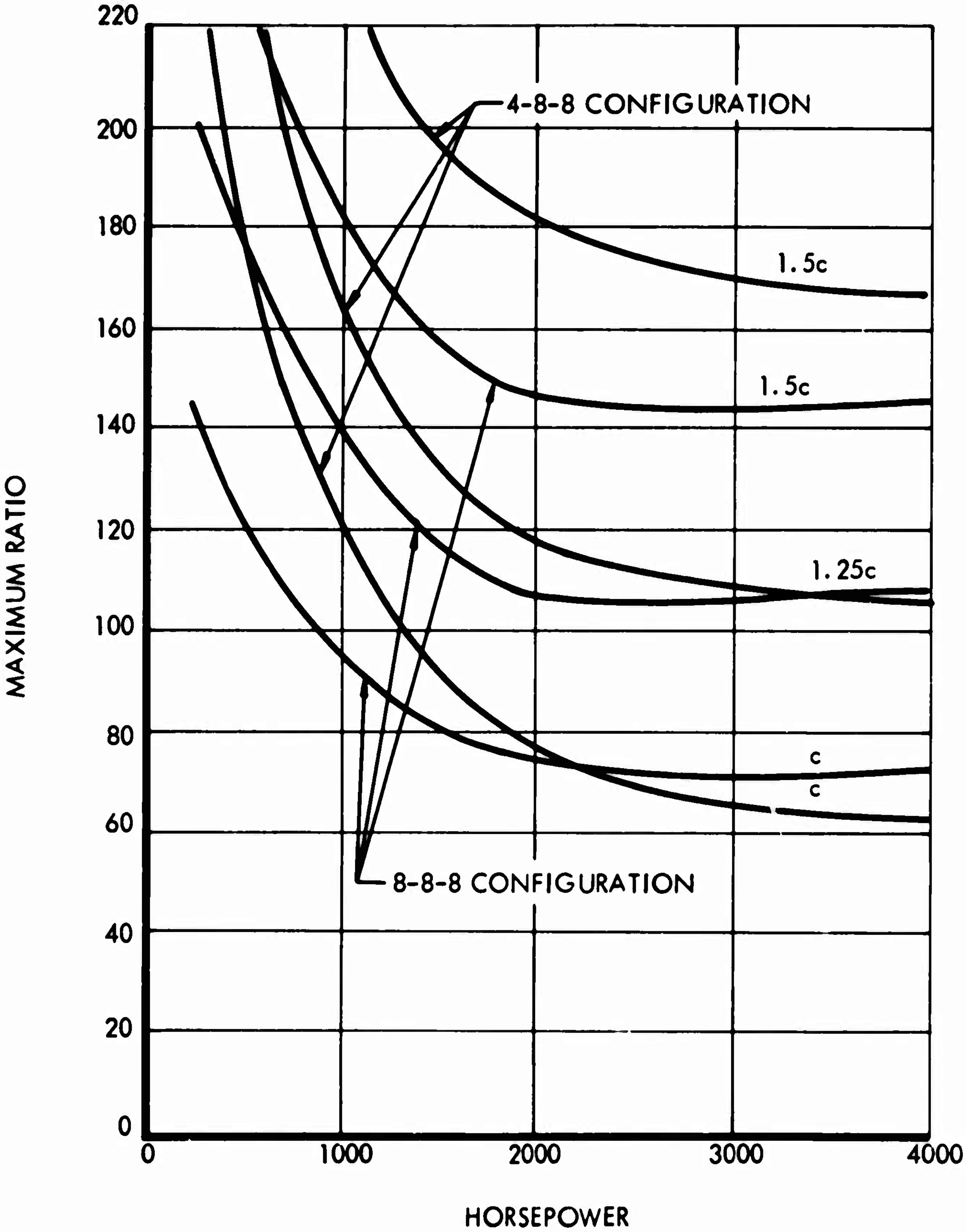


Figure 21. Maximum Ratio Versus Horsepower Curves (4-8-8 and 8-8-8 Drive), Double Staggering

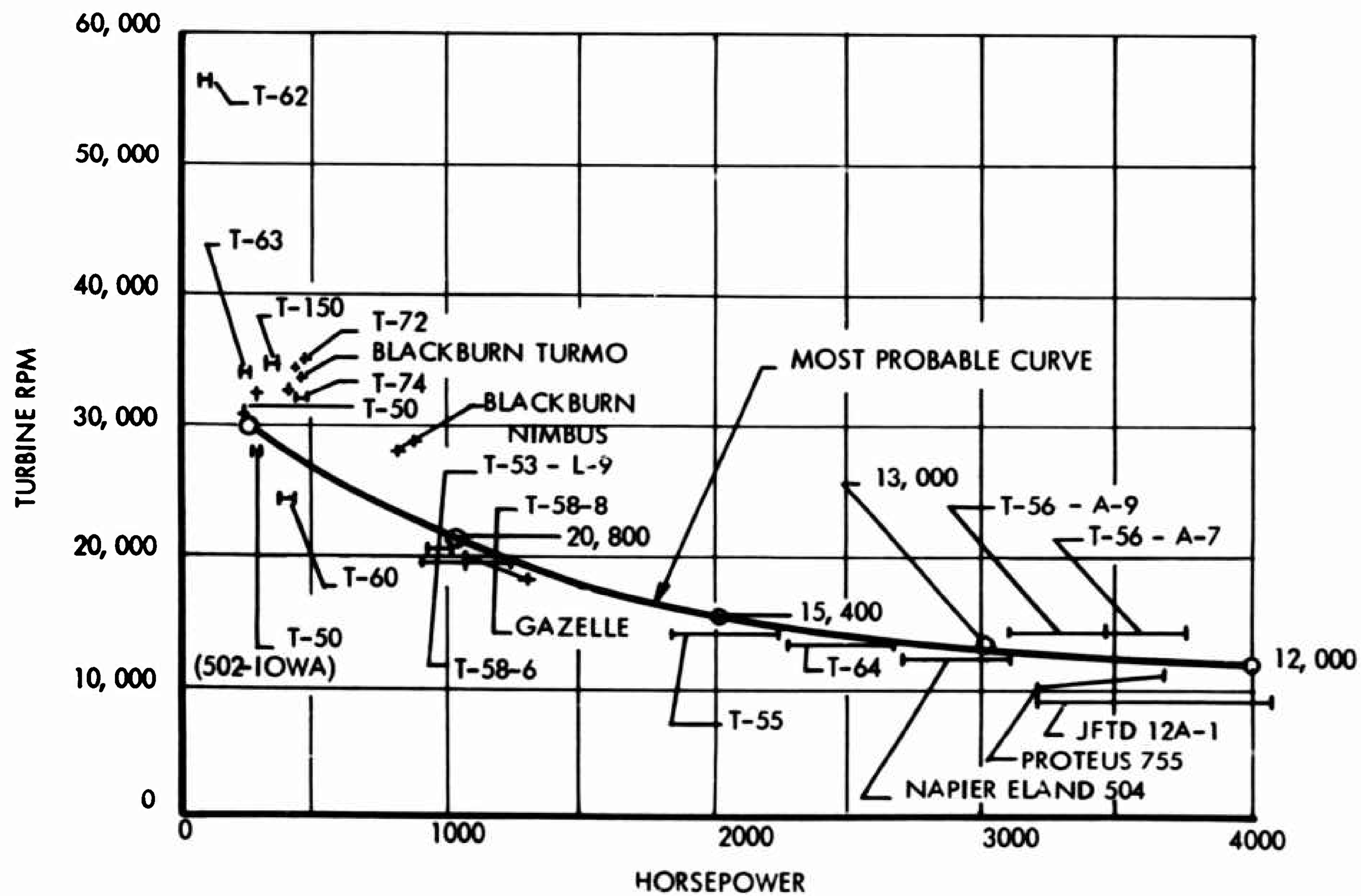


Figure 22. Parametric Analysis - Turbine rpm Versus Horsepower.

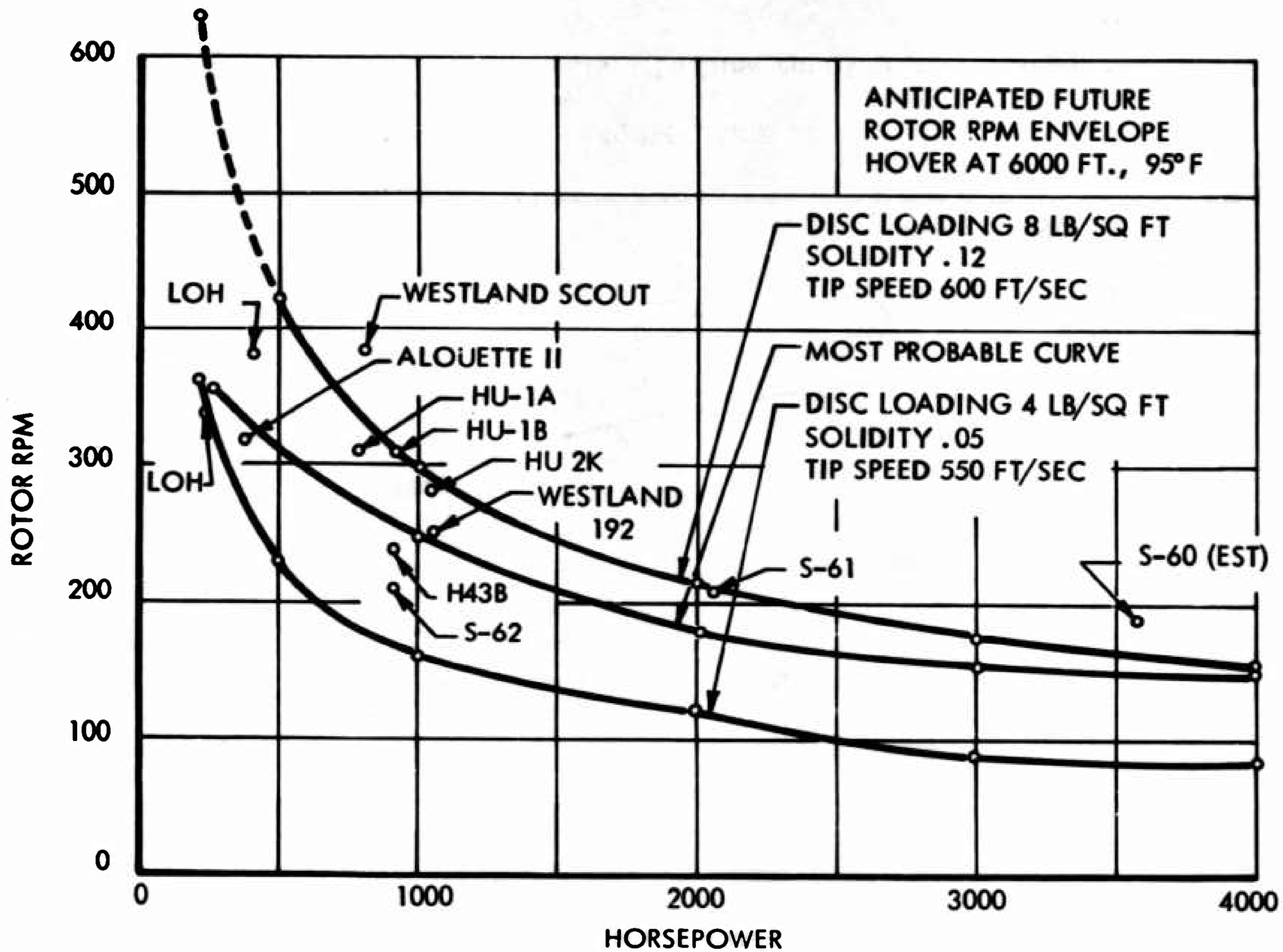


Figure 23. Parametric Analysis - Rotor rpm Versus Horsepower.

TABLE 1

HELICOPTER GEAR RATIOS

Turbine hp	Turbine rpm	Rotor rpm			Reduction Ratio		
		Min.	Avg.	Max.	Max.	Min.	Avg.
250	30,000	340	355	560	88.24	53.57	84.51
500	27,000	230	315	420	117.39	64.29	85.71
750	23,000	185	275	345	124.32	66.67	83.64
1000	21,000	170	244	300	123.53	70.00	86.07
1250	19,000	155	225	260	122.58	73.08	84.44
1500	17,500	135	210	240	129.63	72.92	83.33
1750	16,000	130	190	225	123.08	71.11	84.21
2000	15,500	120	181	210	129.17	73.81	85.64
2250	14,700	110	175	200	133.74	73.50	84.00
2500	14,000	105	170	190	133.33	73.68	82.35
2750	13,500	100	160	182	135.00	74.18	84.38
3000	13,000	95	153	175	136.84	74.29	84.97
3250	12,800	90	150	170	142.22	75.29	85.33
3500	12,600	87	148	165	144.83	76.36	85.14
3750	12,500	85	145	160	147.06	78.12	86.21
4000	12,000	82	141	155	146.34	77.42	85.11

TABLE 2

SELECTED RING GEAR RADIUS FOR EACH HORSEPOWER

Input Hp	Ring Gear Radius, (c) Inches
250	8
500	8.5
750	9
1000	9.5
1250	10
1500	10.5
1750	11
2000	11.5
2250	12
2500	12.5
2750	13
3000	13.5
3250	14
3500	14.5
3750	15
4000	15.5

Equations were developed relating the fatigue life, deflection, stress, and twist replacement which permitted calculations of the sizes of the pinions and their ratios with the mating gears. Pitch line velocity and twist angle were checked in separate calculations. For the configurations with two rows of planets, the toggle angle and clearance were calculated as a check. For the configurations with three rows of planets, the equations for clearance and toggle angle became too cumbersome for rapid solution so a sketch was made to confirm the clearances and toggle angles. In all cases the outer row of planets was checked for space to use a bearing pin without excessive bending stress in the pin.

The general approach to the study of maximum ratio versus horsepower was to find the smallest possible sun gear based upon stress, life, bending, and displacement due to twist. Equations were developed for these parameters. These factors were checked in the other gears and used to determine their sizes when they were the limiting factors; but generally, geometrical limitations prevailed as the limiting factors for the intermediate gears.

The general equations used through the study are as follows:

A. <u>By definition of the K-factor</u>

$$K = \frac{F_t}{D_p \cdot (\text{f.w.})} \cdot \frac{R+1}{R}$$

where:

$$F_t = \text{tangential tooth force}$$

$$= \frac{\text{torque}}{\text{number of contacts} \cdot \frac{D_p}{2}}$$

$$= \frac{2T}{n \cdot D_p}$$

$$D_p = \text{pitch diameter}$$

$$\text{f.w.} = \text{gear face width}$$

$$R = \text{ratio between the gears in contact}$$

$$K = \frac{2T}{n \cdot D_p \cdot (\text{f.w.})} \cdot \frac{R+1}{R}$$

Rearranging,

$$D_p^2 \cdot (\text{f.w.}) = \frac{2\,T}{K \cdot n} \cdot \frac{R+1}{R} \tag{14}$$

The pitch can be selected if the K-factor is known. The K-factor selection, which is based on gear life for the required cycles and case-hardened steel of 58Rc minimum, is determined from practical charts used at TRW.

For a life of 2500 hours, the number of cycles is as follows:

$$N = \text{rpm} \cdot 60 \cdot \text{nc} \cdot 2500$$

$$= 150{,}000 \cdot \text{rpm} \cdot \text{nc}$$

where:

nc = number of contacts per revolution

B. Twist of Sun Gear

Twist Angle, $\theta = \dfrac{T\,L}{G\,J}$ radians

For a maximum displacement of 0.0003 inch at pitch circle

$$\theta \frac{D_p}{2} \leq 0.0003 \text{ inch}$$

Then

$$\frac{T\,L}{G\,J} \cdot \frac{D_p}{2} \leq 0.0003 \text{ inch}$$

where $J = \pi \dfrac{D_p^4}{32}$

and $L = 3$ f.w. (approximately - for split sun)

$$\frac{T \cdot (3\text{ f.w.}) \cdot D_p \cdot 32}{G\,\pi\,D_p^4 \cdot 2} \leq 0.0003 \text{ inch}$$

$$\frac{48\,T \cdot (\text{f.w.})}{G\,\pi\,D_p^3} \leq 0.0003 \text{ inch} \tag{15}$$

C. Pitch Line Velocity Check

$$\frac{\pi D N}{12} \leq 16,000 \text{ feet per minute for spur gears} \qquad (16)$$

D. The Twist Should Be Less Than 0.0002 Inch Per Inch

$$\theta = \frac{T \cdot (f.w.)}{G J}$$

$$\theta = \frac{s}{D_p/2} ;$$

$$\frac{s}{f.w.} \leq 0.0002 \text{ in./in.}$$

$$s = \theta \frac{D_p}{2}$$

$$\frac{\theta \frac{D_p}{2}}{f.w.} \leq 0.0002 \text{ in./in.}$$

Insert $\theta = \frac{T \cdot (f.w.)}{G J}$

$$\frac{T \cdot (f.w.)}{G J \cdot (f.w.)} \cdot \frac{D_p}{2} = \frac{T D_p}{2 G J} \leq 0.0002 \text{ in./in.}$$

$$J = \frac{\pi D_p^4}{32}$$

$$\frac{T D_p \cdot 32}{2 G \cdot \pi D_p^4} \leq 0.0002 \text{ in./in.}$$

$$\frac{16 T}{G \cdot \pi D_p^3} \leq 0.0002$$

$$D_p^3 \geq \frac{16\ T}{\pi\ G \cdot 0.0002}$$

$$G = 11.5 \cdot 10^6$$

$$D_p^3 \geq 2.214329642 \cdot 10^{-3}\ T \tag{17}$$

E. Bending Stress

Bending stress on the teeth should be less than 30,000 psi. Since the pinion should have between 18 and 36 teeth for such high horsepower, the average number (24 teeth) was chosen for the following equations:

$$S_b = \frac{Ft \cdot Pd}{y \cdot (f.w.)}$$

where y is the Lewis form factor for 24 teeth, $y = 0.337$ and

$$Ft = \frac{2\ T}{n\ D_p}$$

$$Pd = \frac{N_p}{D_p}$$

Thus,

$$30,000 \geq S_b = \frac{\frac{2\ T}{n\ D_p} \cdot \frac{2\ 4}{D_p}}{0.337 \cdot (f.w.)} = \frac{48\ T}{0.337\ n \cdot (f.w.)\ D_p^2}$$

$$D_p^2 \cdot (f.w.) \geq \frac{48\ T}{0.337 \cdot 30,000\ n} \tag{18}$$

To find D_p, use equation (14)

$$D_p^2 \cdot (f.w.) = \frac{2\ T}{K\ .\ n} \cdot \frac{R + 1}{R}$$

and equation (15)

$$\frac{48\ T \cdot (f.w.)}{G\ \pi\ D_p^3} \leq 0.0003 \text{ inch}$$

Rearranging, combining, and using the equal sign to indicate the smallest size sun gear, the following equation is obtained:

$$D_p^5 = \frac{96\,T^2}{0.0003\,\pi\,G \cdot K \cdot n} \cdot \frac{R+1}{R}$$

Using $$G = 11.5 \cdot 10^6$$

$$D_p^5 = \frac{96}{3 \cdot 10^{-4}\,\pi \cdot 11.5 \cdot 10^6} \cdot \frac{T^2}{K \cdot n} \cdot \frac{R+1}{R}$$

$$= 8.85731857 \cdot 10^{-3} \cdot \frac{T^2}{K \cdot n} \cdot \frac{R+1}{R}$$

$$D_p = \sqrt[5]{8.85731857 \cdot 10^{-3} \cdot \frac{T^2}{K \cdot n} \cdot \frac{R+1}{R}} \qquad (19)$$

Equation 19 is applicable to a split sun gear design only. Most two row designs have a split sun gear. Such a design is shown in Figure 24. The upper half of Figure 24 shows a 4-4 system, nonstaggered. The lower half shows 6-6 staggered system. The dashed line on the stretchout view indicates the adjacent X_1 gears' axial location. Most two row systems considered are staggered.

Figures 25, 26, 27, and 28 show schematic representations of the configurations considered in the maximum ratio versus horsepower study. The systems that each configuration was used in are indicated by the numbers near the cluster view.

Maximum ratios based from the turbine rpm for all systems for 4000 horsepower are shown in Table 3. This table provides a quick comparison between types of systems. It cannot readily be shown graphically because it is a cross section of each graph shown in Figures 15 through 21 and of other systems not considered.

OPTIMUM WEIGHT AND VOLUME

During the parametric study, helicopter drives with horsepowers of 250, 500, 750, 1,000, 1,500, 2,000, 3,000, and 4,000 were investigated, each for reduction ratios of 20, 40, 60, 80, and 100. In all, 37 drives were calculated. Each was based upon input speed equal to the turbine rpm given in Table 1. For each drive a schematic layout was prepared, basic calculations for gear sizes were performed, and weights were estimated.

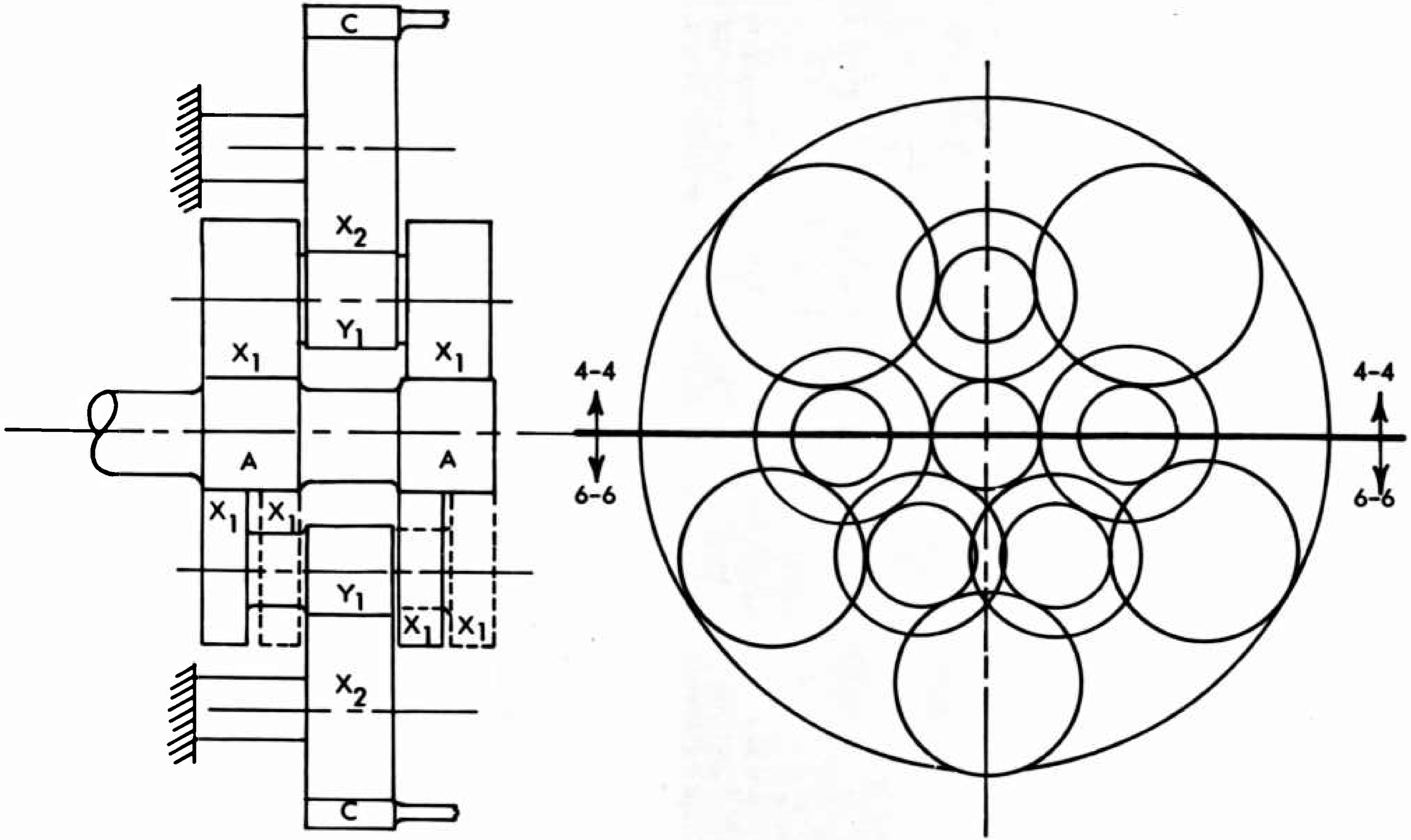


Figure 24. Roller Gear Cluster, Two Row Configuration

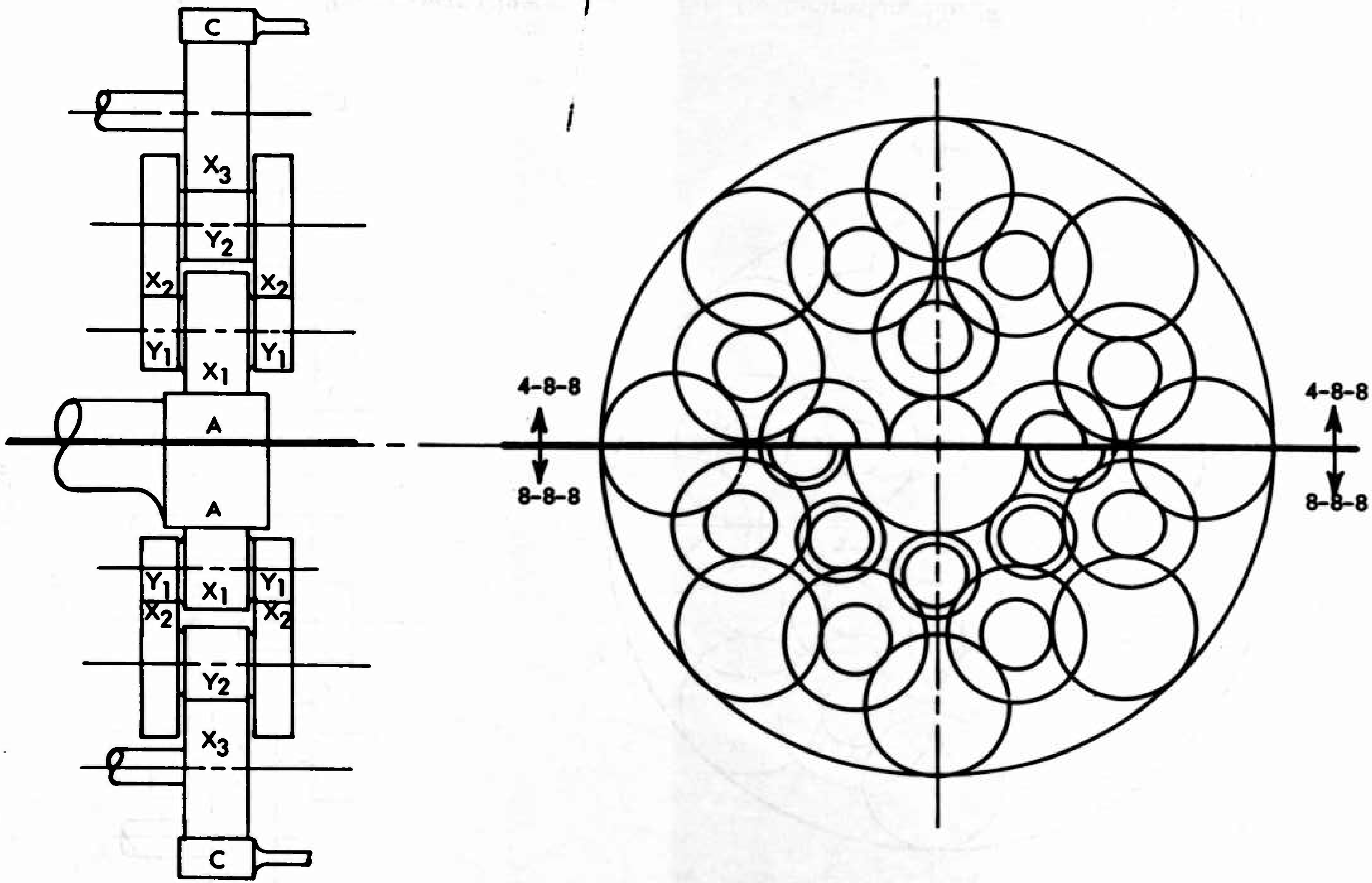


Figure 25. Roller Gear Cluster, Three Row Configuration,Nonstaggering.

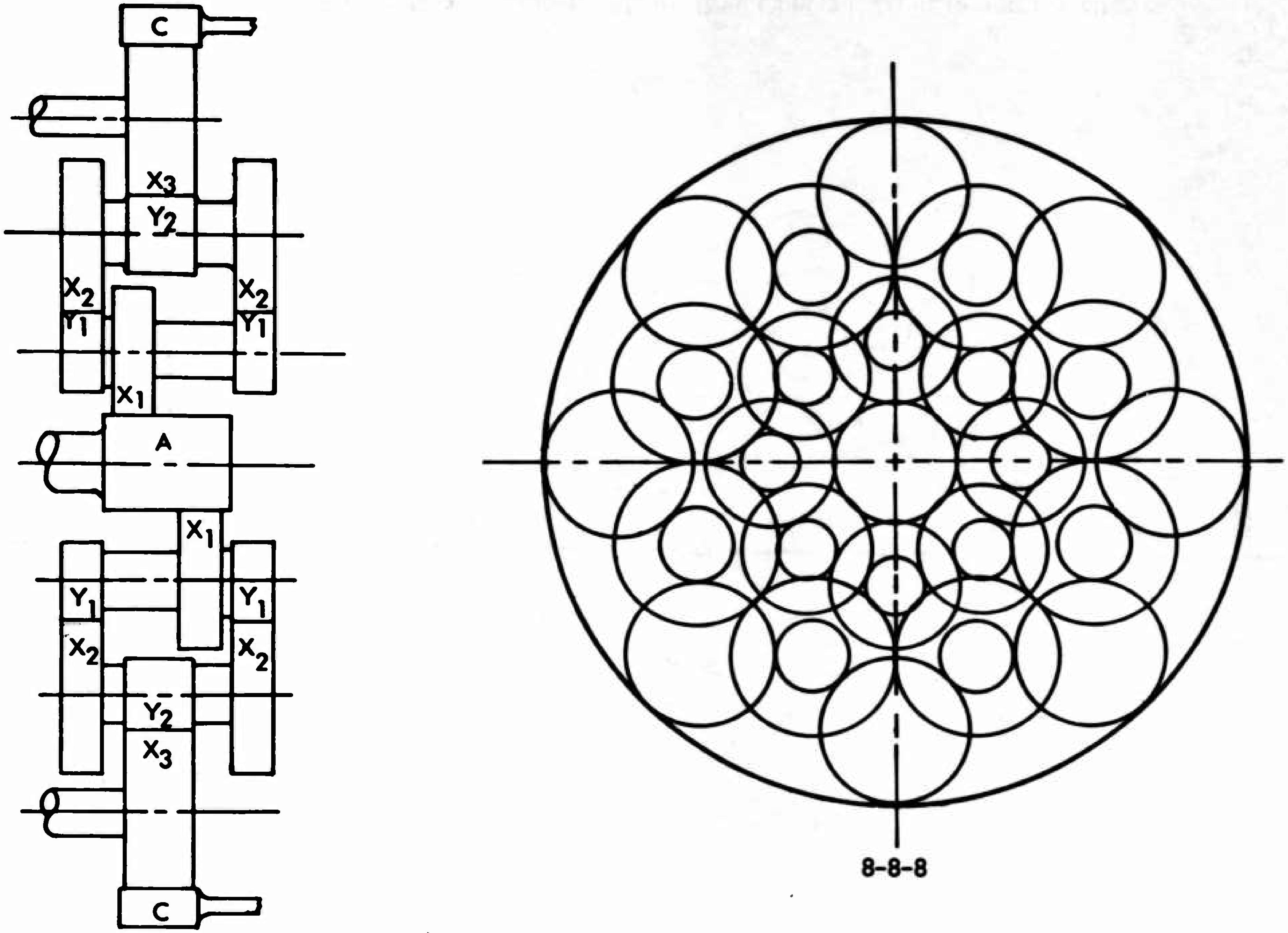


Figure 26. Roller Gear Cluster, Three Row Configuration, Single Staggered (X_1 - X_1).

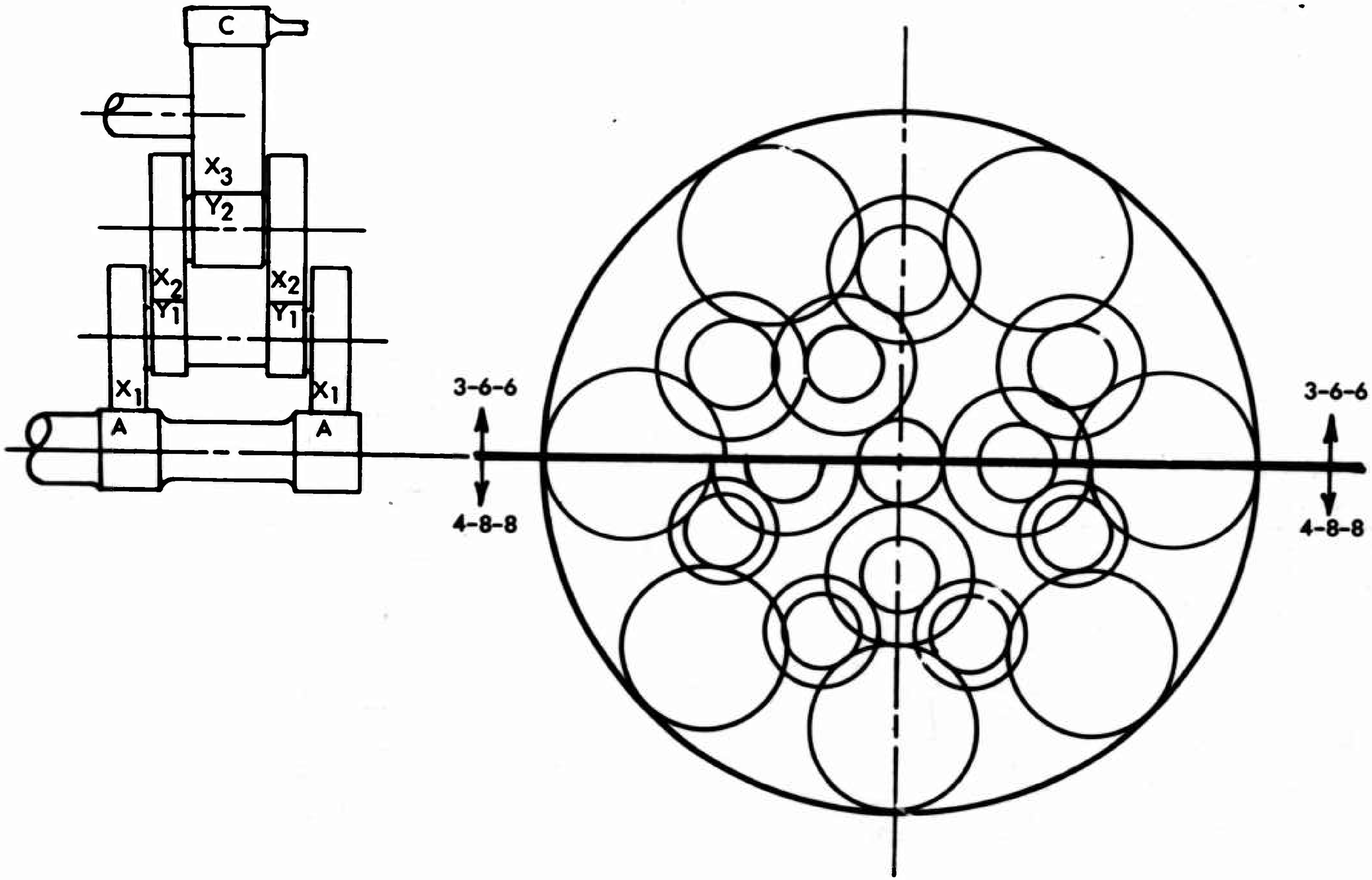


Figure 27. Roller Gear Cluster, Three Row Configuration, Single Staggering (X_1 - X_3).

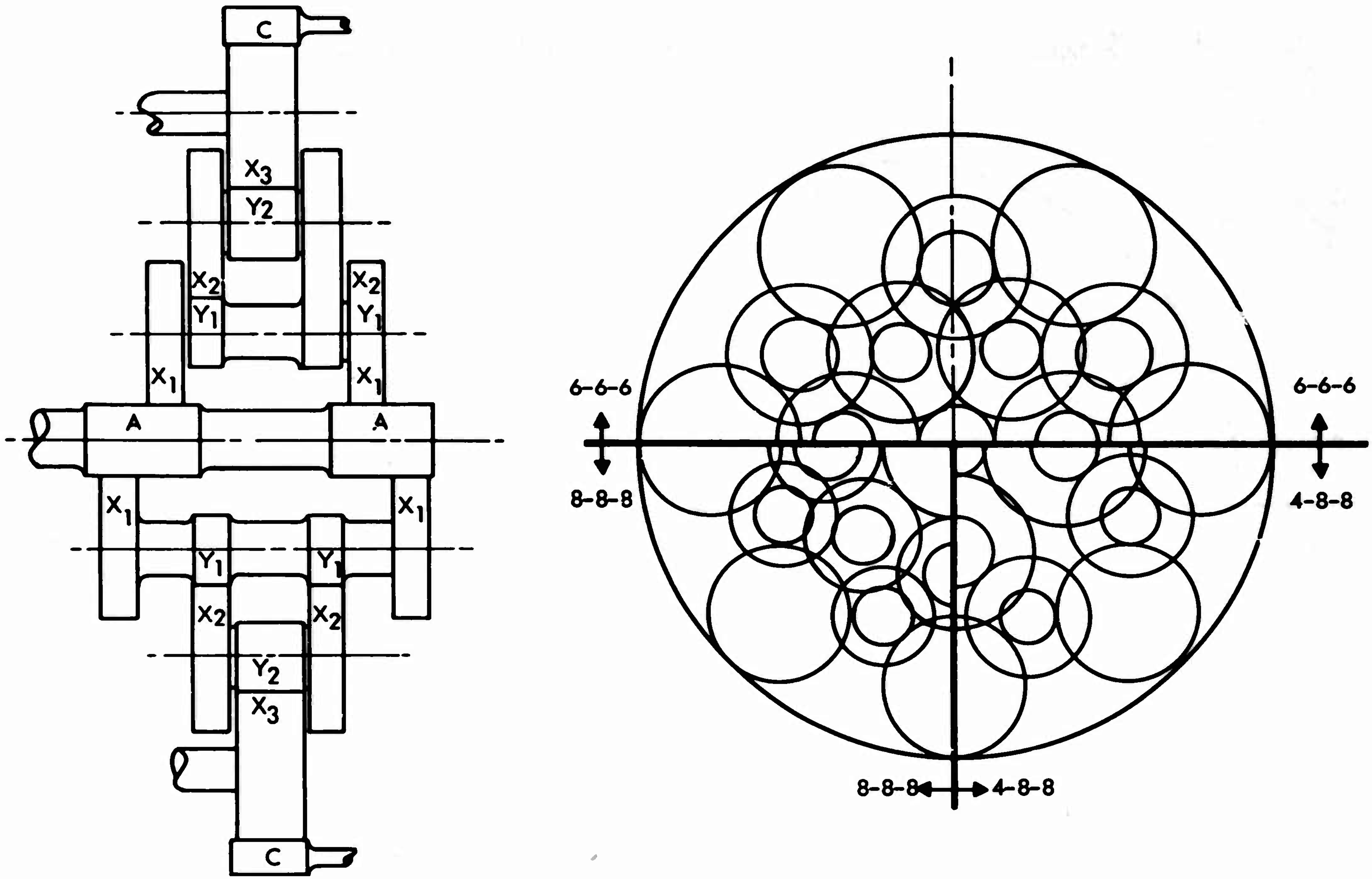


Figure 28. Roller Gear Cluster, Three Row Configuration, Double Staggering (X_1 - X_1 and X_3).

TABLE 3

MAXIMUM RATIOS - 40[illegible] HP, 12,000 RPM INPUT RING GEAR OUTPUT

Number of Rows	Staggering	Type (Fig. No.)	Maximum Ratio (Ring Gear Radius c, Inches)		
			(15.5)	(19.375)	(23.25)
2	Not Staggered	4-4(15)	8.9	16.0	19.2
		5-5(*)	12.7	15.9	16.3
		6-6(*)	10.0	10.0	10.0
		8-8(*)	6.3	6.3	6.3
2	Staggered	4-4(15)	-	-	23.2
		6-6(16)	16.3	27.0	40.5
		8-8(17)	20.1	23.0	25.0
3	Not Staggered	4-8-8(19)	42.3	59.8	75.7
		8-8-8(19)	21.3	34.2	42.7
3	Single Staggered	3-6-6(18)	35.5	68.5	114.0
		4-8-8(20)	61.5	98.4	139.0
		8-8-8(20)	36.1	54.1	56.3
3	Double Staggered	6-6-6(18)	59.5	88.2	126.0
		4-8-8(21)	63.9	107.8	168.4
		8-8-8(21)	73.3	110.9	147.3

*These ratios were calculated only for this case

The gear teeth numbers were not calculated. It may not always be possible to obtain the reported weights without going to a fractional teeth number relationship. As mentioned earlier in the report, this is undesirable with respect to manufacturing and assembly problems, and it may be better to add some weight to obtain simplicity.

In general, the two row system with 4 planets per row is the simplest and least expensive drive. It was used wherever possible.

Next in complexity is the two row, 6 planets per row system. The calculations indicate that this system is heavier than the 3-6-6 and 4-8-8 systems, but it offers simplicity and less cost. It is used where the 4-4 system cannot yield the required ratio without going to a very large ring gear.

A system using 3 planets in row 1 and 6 planets each in rows 2 and 3 offers considerable weight savings over the 4-4 and 6-6 systems. This system is normally heavier than the 4-8-8 system. However, its main advantage is that it requires fewer gears; thus, it is less expensive. The 4-8-8 system is the most compact and the lightest in weight.

In the analysis, only the single staggered arrangement, such as shown in Figure 27, was used for the three row systems. In both cases, the X_1 gears stagger the X_3 gears.

The maximum possible ratio study ruled out the use of the following systems for drives based upon the use of the minimum allowable sun gear size:

1. Not staggered systems 6-6, 3-6-6, 4-8-8, 6-6-6, and 8-8-8. In each case the outside diameter would have to be too large to get the required ratio.

2. Single staggered systems 6-6-6 and 8-8-8. Neither system offers as much ratio as the systems chosen.

3. Double staggered systems 6-6-6, 4-8-8, and 8-8-8. All these systems indicate better ratios for a given output ring gear size; however, they are heavier because of the second row of X_1 gears, and, generally, all gears are larger.

The graphical results of the optimum weight and volume study are shown in Figures 29 through 33. All of these show the results based upon an input speed equal to the turbine rpm. Figure 29 shows the estimated drive dry weight versus horsepower for constant ratios. The estimated drive volume versus horsepower at various ratios is shown in Figure 30. Figure 31 represents a cross plot of the data shown in Figure 29. In this case the estimated drive dry weight is plotted against the reduction ratio as a function of horsepower. Figure 32 presents the same data as shown in Figure 29. This time the estimated drive dry weight is plotted against the drive input torque as a function of reduction ratio. The last curve in this series, Figure 33, shows the estimated outside

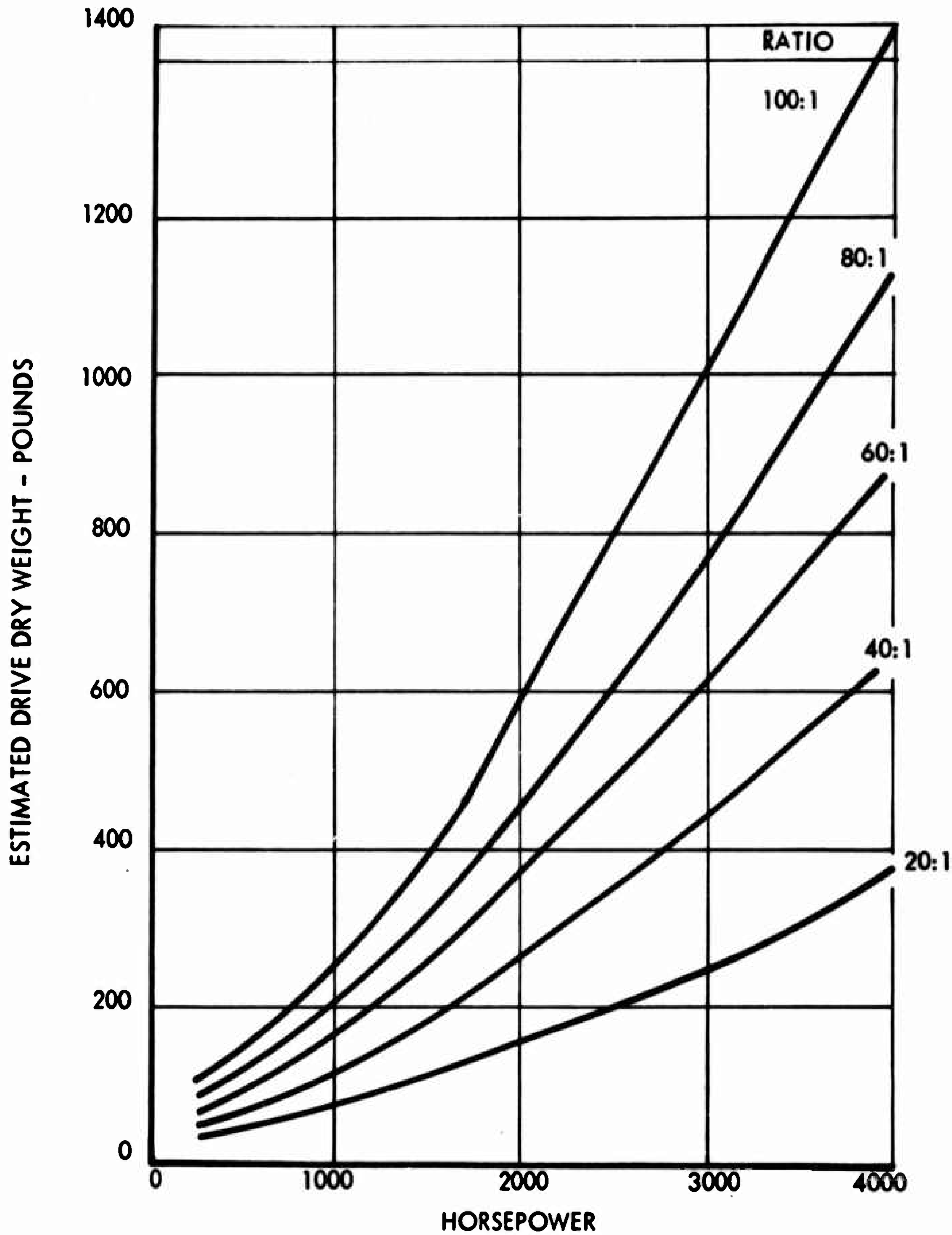


Figure 29. Estimated Drive Dry Weight Versus Horsepower for Constant Ratios - Turbine Input rpm.

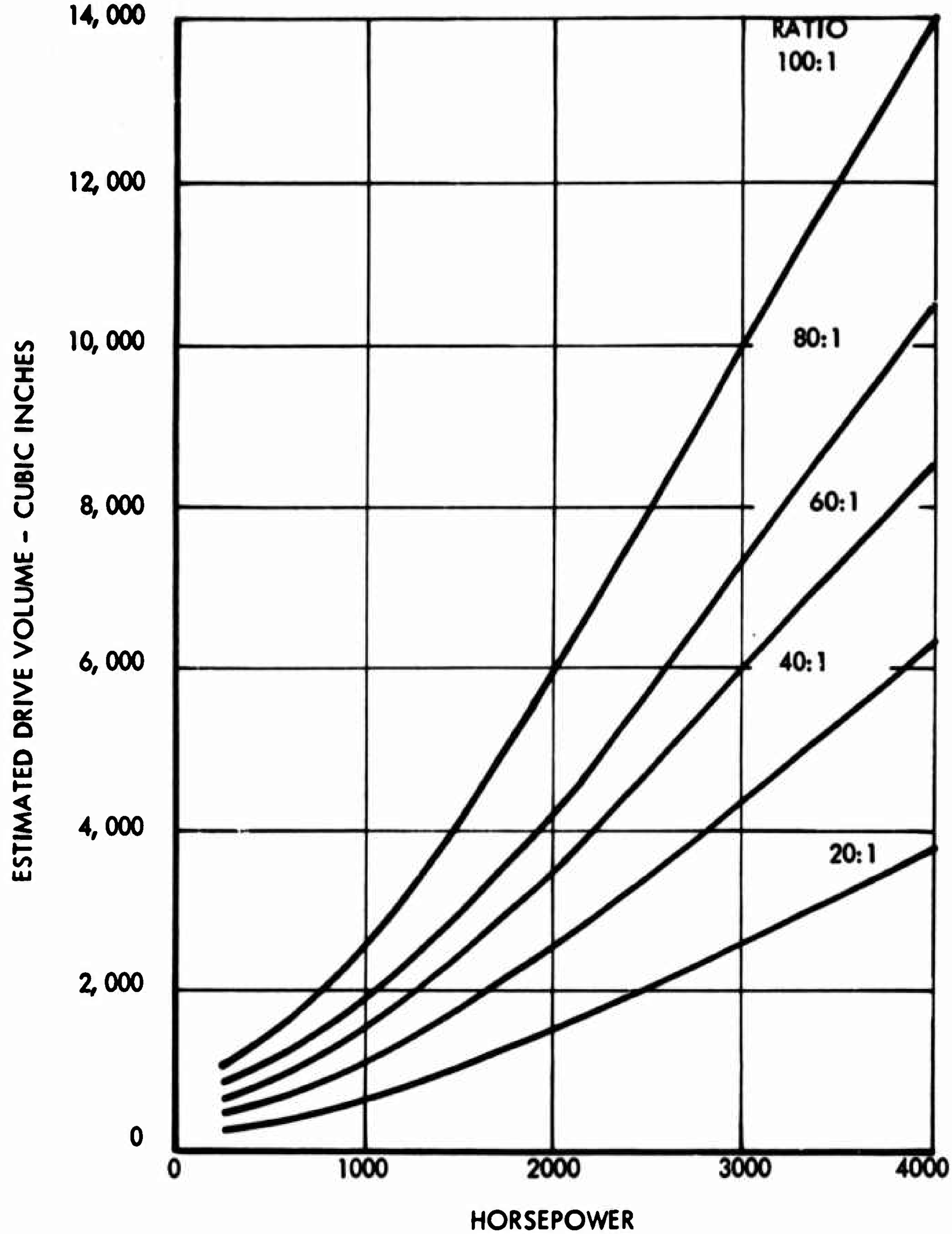


Figure 30. Estimated Drive Volume Versus Horsepower for Constant Ratios - Turbine Input rpm.

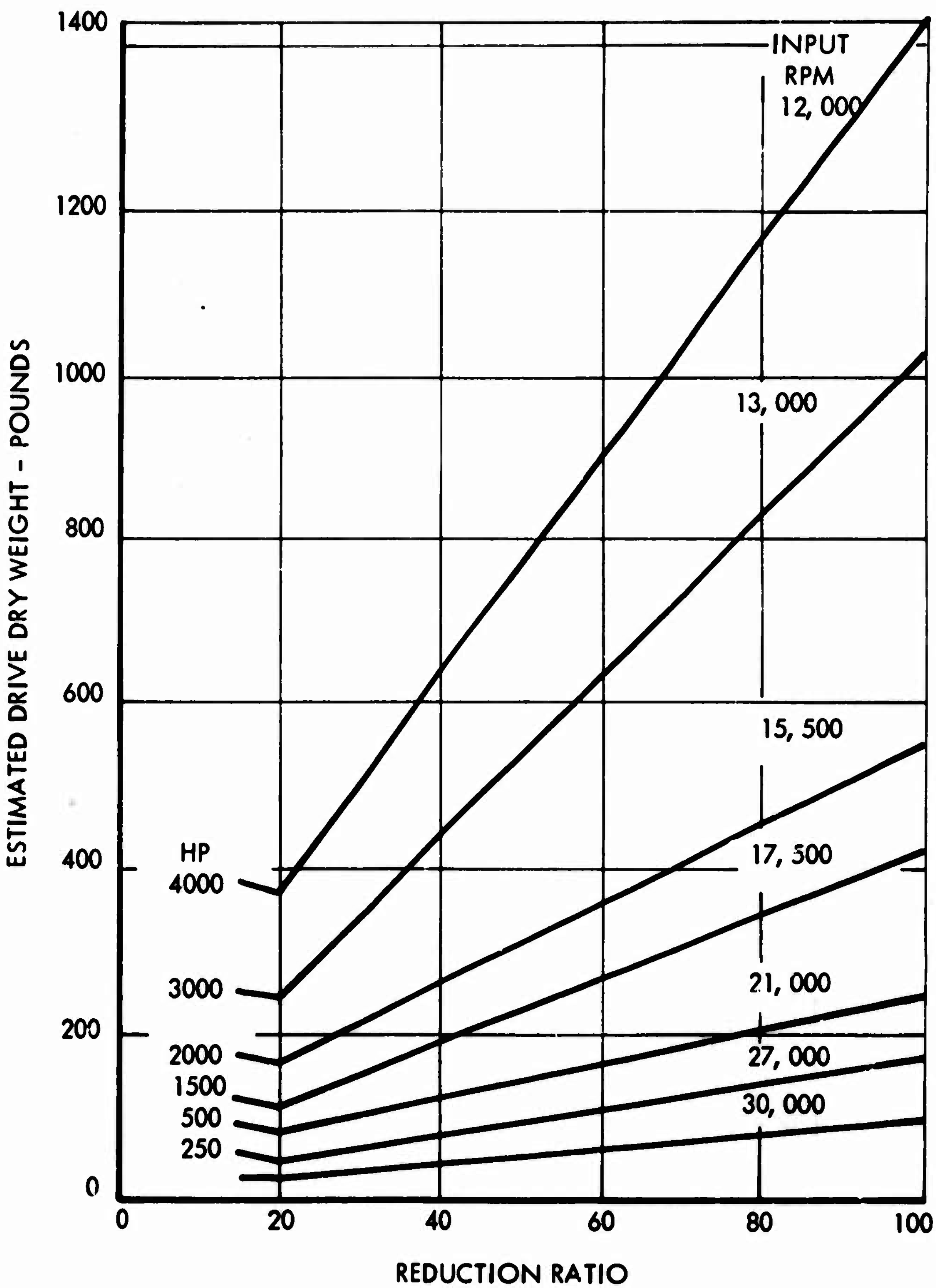


Figure 31. Estimated Drive Dry Weight Versus Reduction Ratio for Constant Input Torque - Turbine Input rpm.

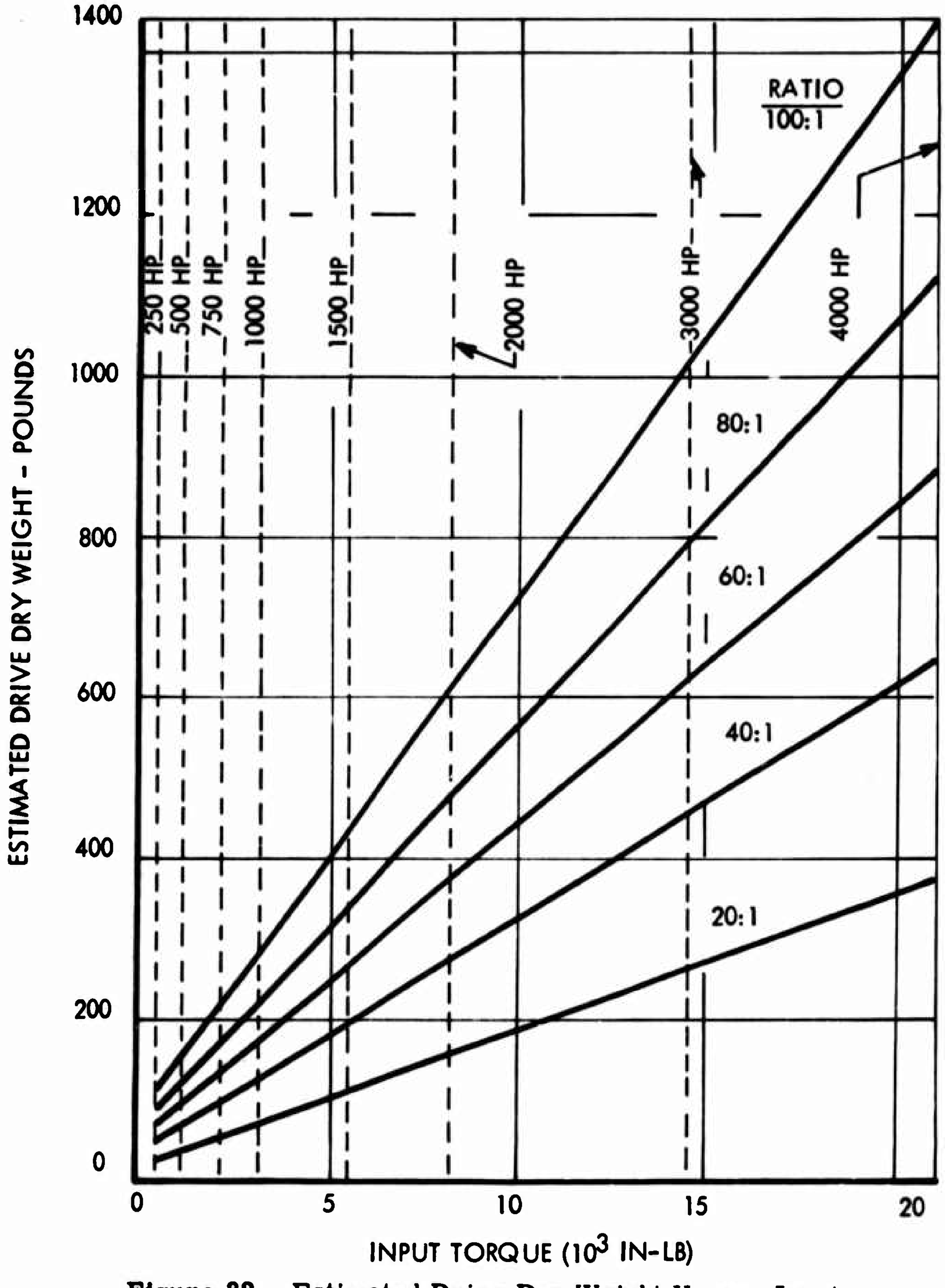


Figure 32. Estimated Drive Dry Weight Versus Input Torque - Turbine Input rpm.

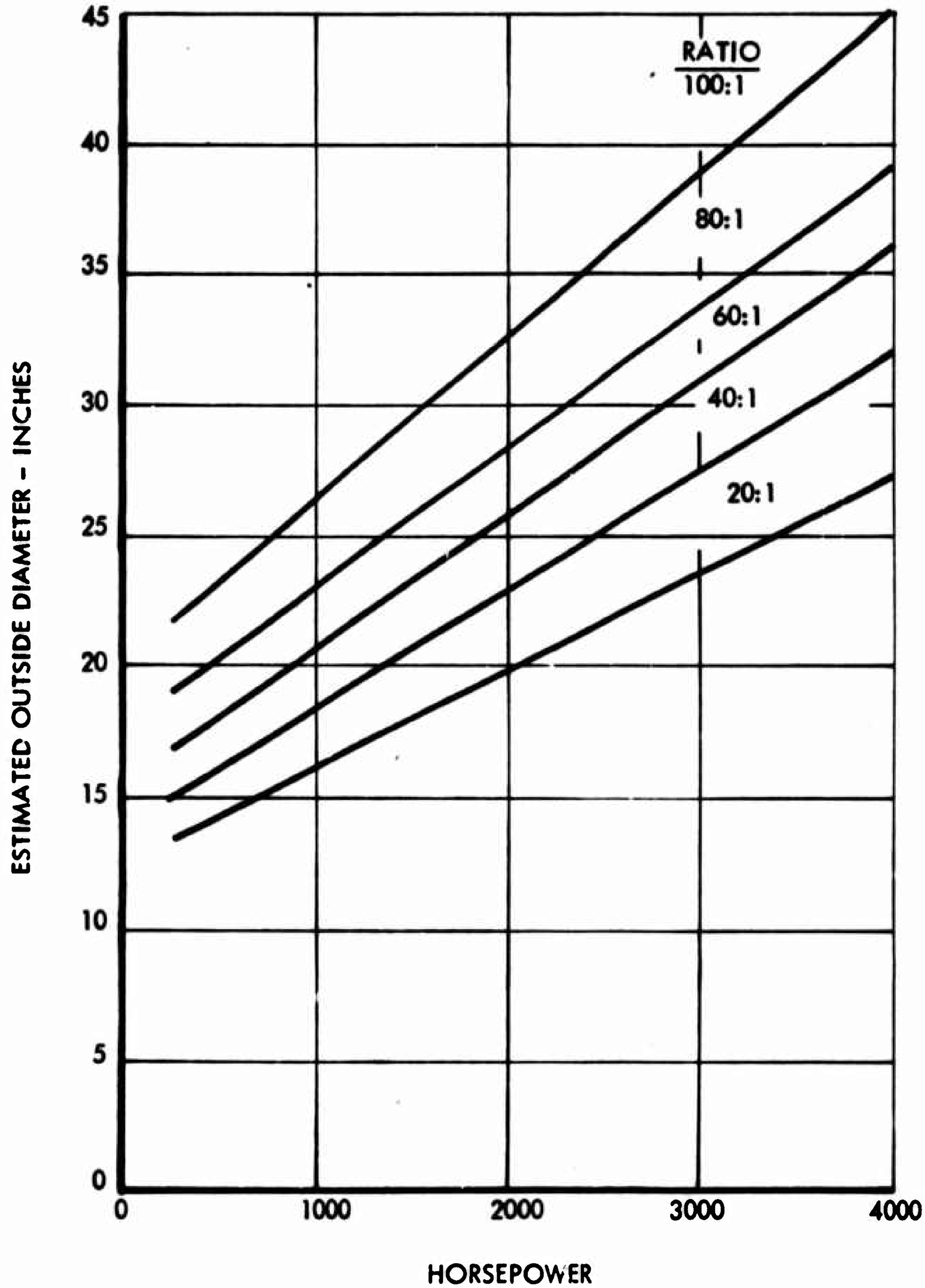


Figure 33. Estimated Outside Diameter Versus Horsepower for Constant Ratios - Turbine Input rpm.

diameter versus horsepower as a function of ratio. Appendix A contains the basic equations used in the analysis for weight and volume. Figures 24 through 28 show the basic schematic representation of the various types of drives considered during the study. Appendix B contains the schematic drawing, Figure 43, and detailed calculations for a drive of 1,000 horsepower and 100:1 reduction ratio.

Limiting geometric factors which affect the various designs are:

1. Minimum toggle angle - this allows for the possibility of the second row of planets' slipping inward out of position.

2. Maximum toggle angle - this allows for the possibility of the second row of planets' slipping outward out of position.

3. Interference between X_1 - X_1, X_2 - X_2, X_1 - shaft of Y_1, X_1 - bearing shaft of X_2 (in two row systems) and X_1 - bearing shaft of X_3 (in three row systems) - enough space must be allowed for the tooth addendum and some additional clearance for oil.

4. Bearing size - the last row gears must be large enough to contain the bearing.

The sample calculations indicate which conditions affect that particular design.

A considerable number of helicopter applications of the TRW roller gear drive may not use the roller gear drive next to the turbine; rather, they may use an intermediate reduction, such as a right angle bevel gear.

For this reason the calculated data are projected to a torque equivalent to the output or rotor speed.

From Figure 32 it is observed that weight is proportional to torque. Thus, all of the calculated data can very easily be converted to a base torque. This permits the data to be presented at a given torque and horsepower for all ratios; or simply, it can be converted to a drive based upon rotor output conditions.

To do this, the weight being converted is multiplied by the base torque over the converted torque (T_b/T). If the base torque (T_b) is higher than the torque being converted (T), the weight will increase by that ratio (T_b/T). Also, the weight will vary as one over the cubed root of the base torque over the converted torque $1/\sqrt[3]{(T_b/T)}$ due to the change in cyclic life. If the base torque (T_b) is larger than the torque being converted (T), the cyclic life will be less by the ratio of one over the base torque divided by the converted torque ($1/T_b/T$); thus life is decreased by the cubed root of this value, and the weight proportionately decreased as the life.

$$W_b = W/\sqrt[3]{T_b/T}$$

This gives the weight for the base torque at the original horsepower.

$$W_b = W \frac{Tb/T}{\sqrt[3]{T_b/T}} = W \sqrt[3]{(T_b/T)^2}$$

Since the output torque is the input torque multiplied by the reduction ratio, the ratios may be substituted in the above equation. This relates the given weight based on input torque at turbine speed to the base torque at the base speed. The base torque and speed are chosen to be the rotor speed as given in Table 1.

$$W_b = W \sqrt[3]{(R_b/R)^2} \qquad (20)$$

The average reduction ratio is 85:1; thus, this is the base ratio.

For 100:1 ratio, the conversion factor is:

$$W_b = W_{100} \sqrt[3]{(85/100)^2} = W_{100} \times .897$$

For 80:1

$$W_b = W_{80} \sqrt[3]{(85/80)^2} = W_{80} \times 1.04$$

For 60:1

$$W_b = W_{60} \sqrt[3]{(85/60)^2} = W_{60} \times 1.26$$

For 40:1

$$W_b = W_{40} \sqrt[3]{(85/40)^2} = W_{40} \times 1.65$$

For 20:1

$$W_b = W_{20} \sqrt[3]{(85/20)^2} = W_{20} \times 2.62$$

The curves obtained by use of the above formulas using the weights in Figure 29 are shown in Figures 34, 36, 37, and 38.

The above conversion factors are equally adaptable to volumes. The curve thus converted is shown in Figure 35.

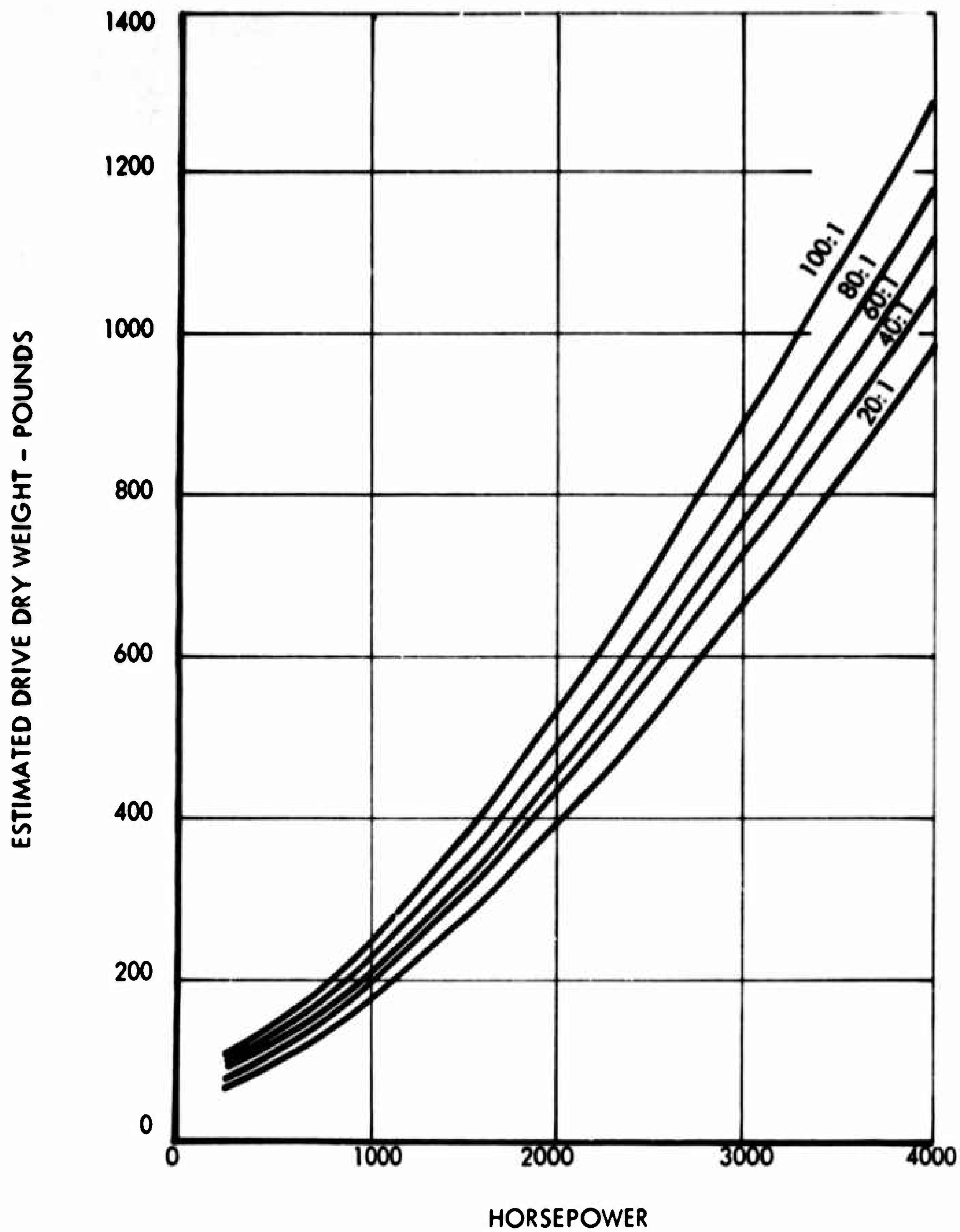


Figure 34. Estimated Drive Dry Weight Versus Horsepower for Constant Ratios - Rotor Output rpm

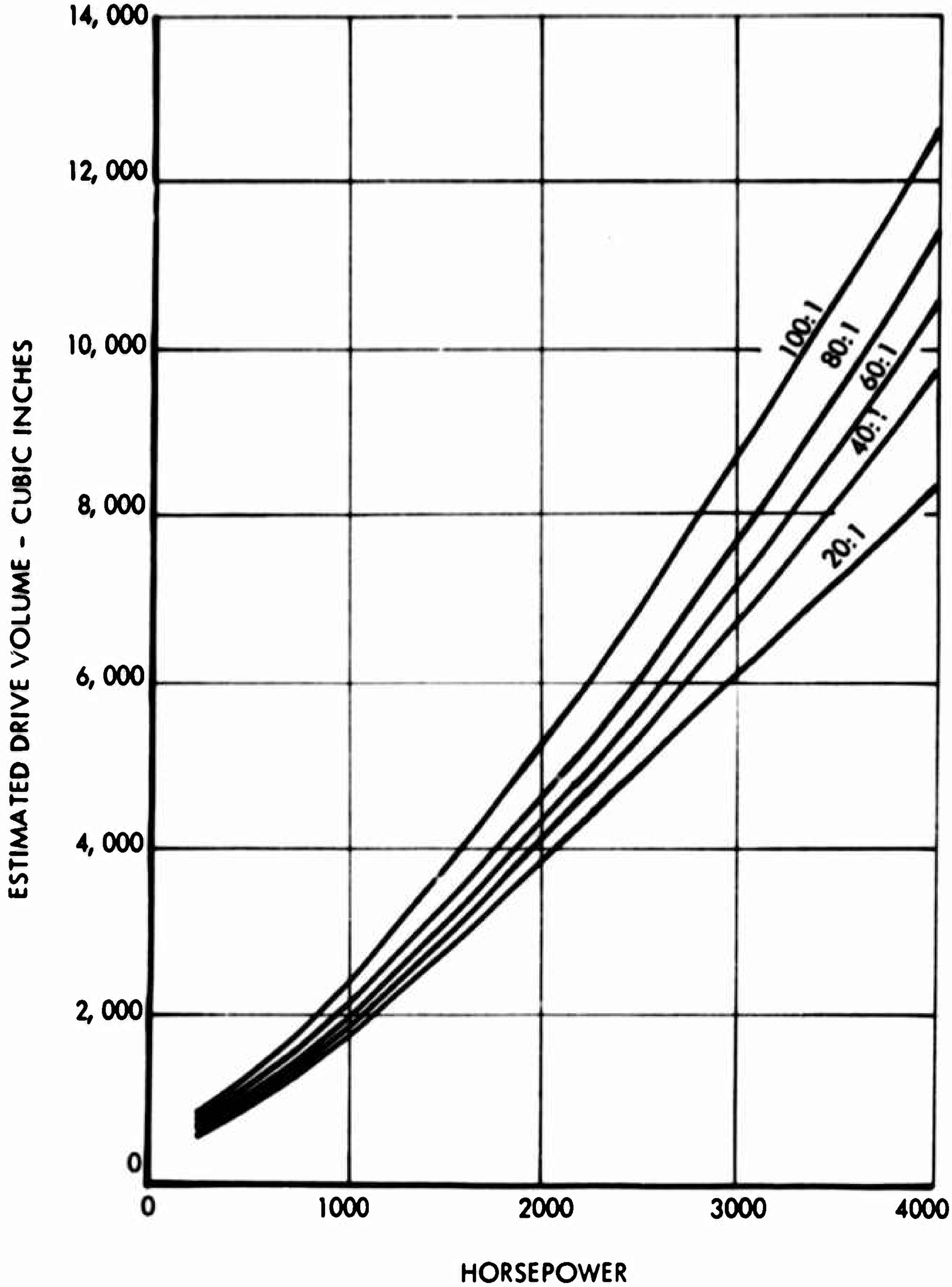


Figure 35. Estimated Drive Volume Versus Horsepower for Constant Ratios - Rotor Output rpm.

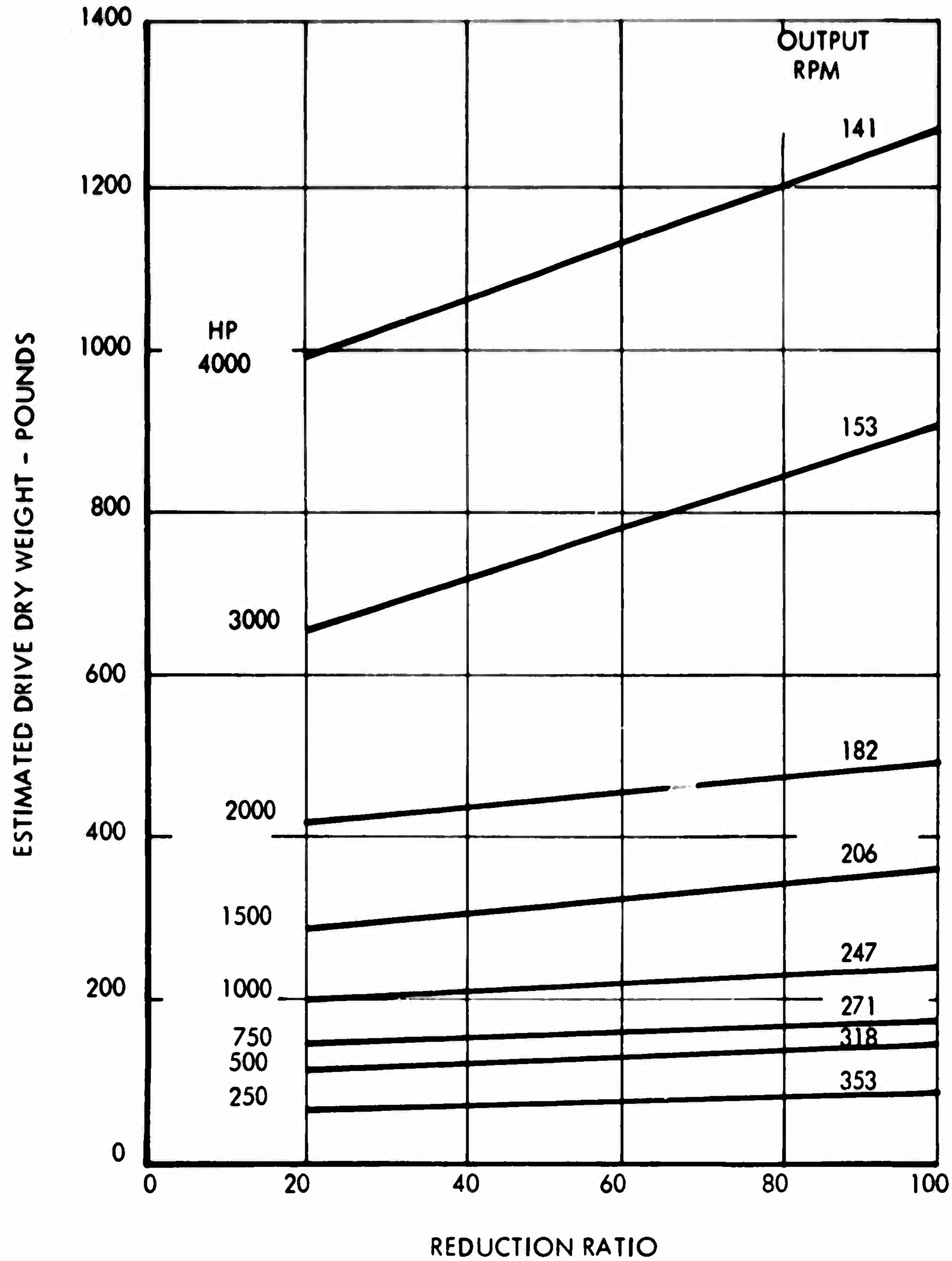


Figure 36. Estimated Drive Dry Weight Versus Reduction Ratio for Constant Output Torque - Rotor Output rpm.

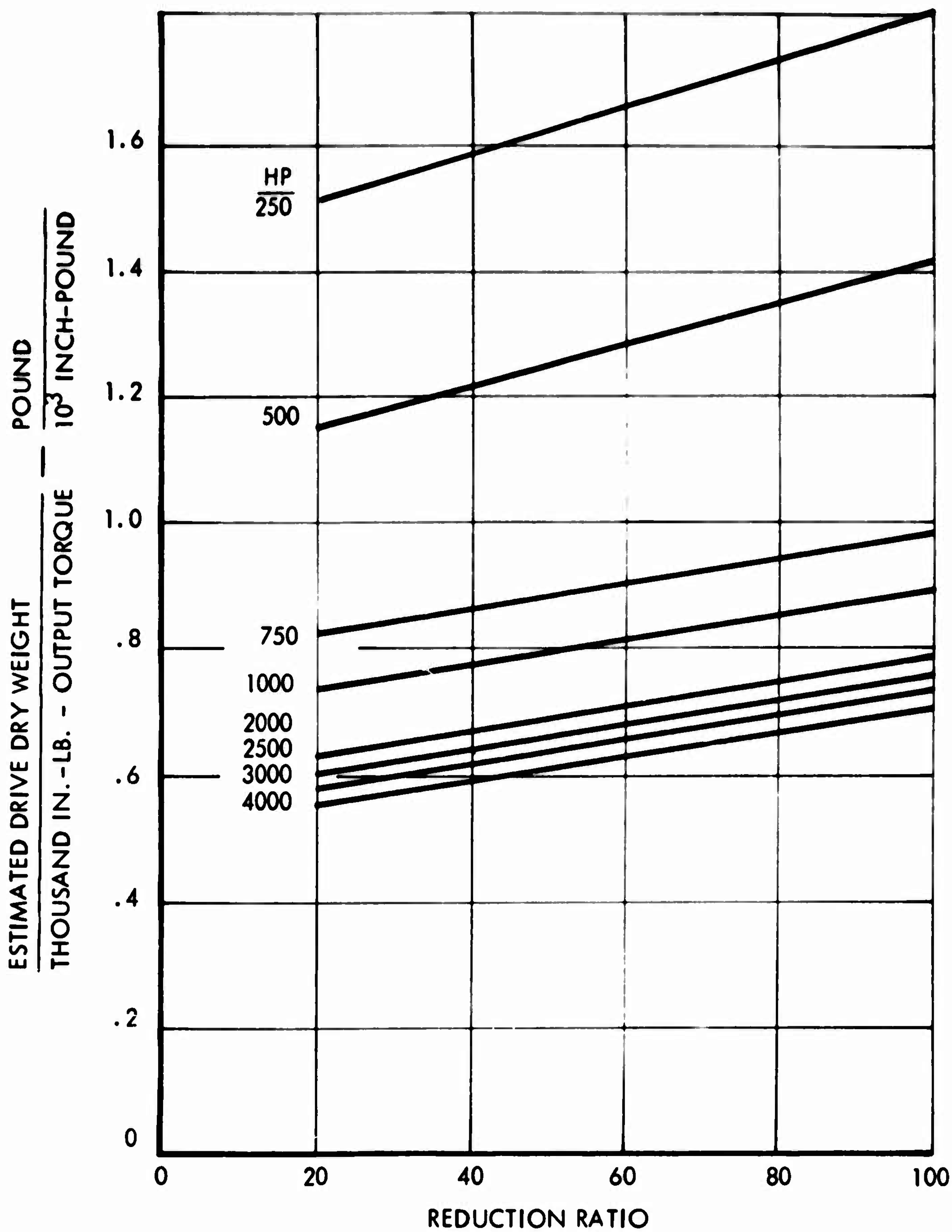


Figure 37. Estimated Drive Weight per Thousand Inch-Pound Output Torque Versus Reduction Ratio-Rotor Output rpm

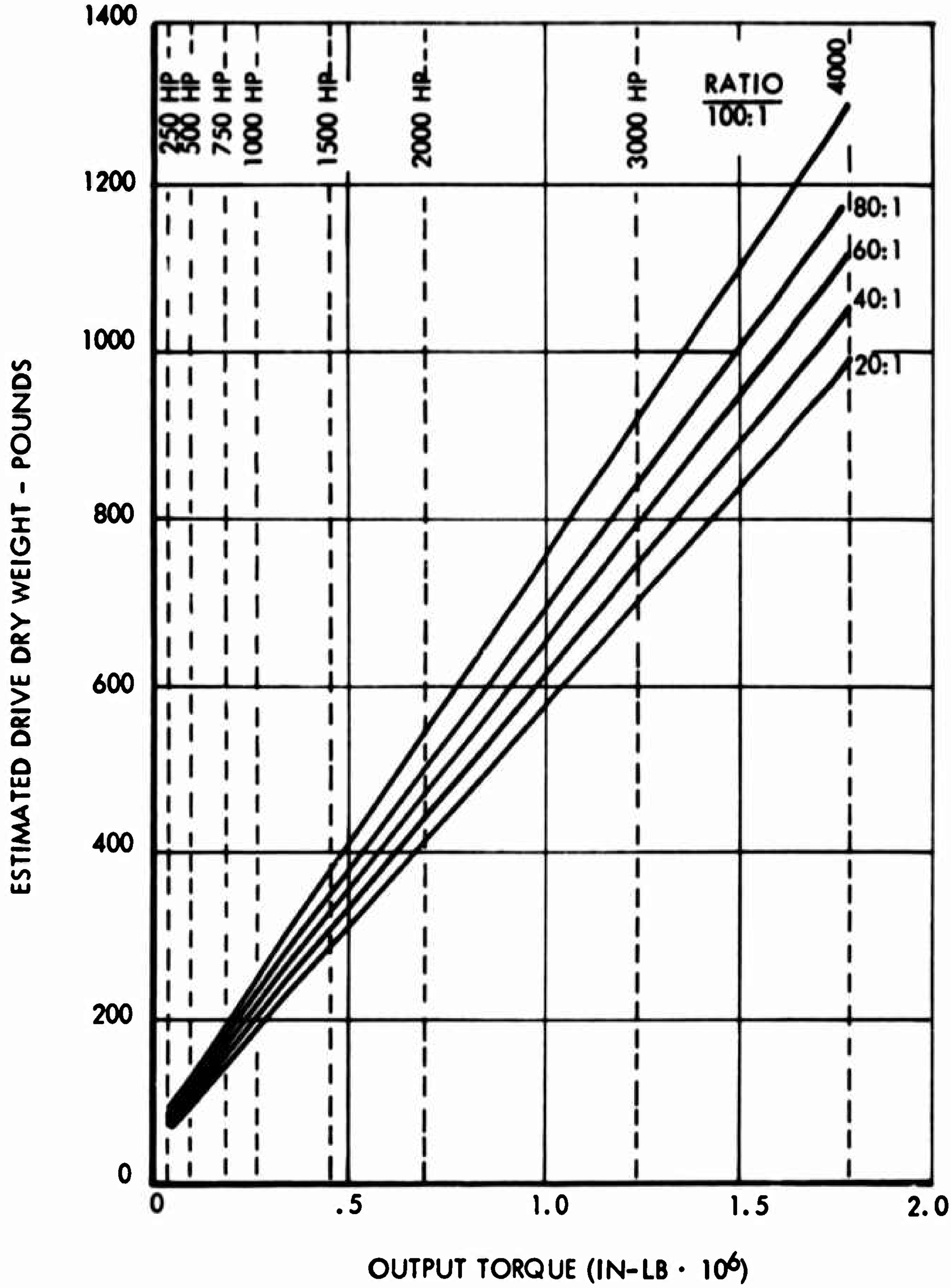


Figure 38. Estimated Drive Dry Weight Versus Output Torque for Constant Ratios - Rotor Output rpm.

Figure 39 shows a plot of outside diameter versus horsepower. The outside diameter varies as the cubed root of the torques; therefore, the conversion factor is the cube root of the base ratio over the converted ratio.

$$D_b = D \sqrt[3]{R_b/R} \qquad (21)$$

The outside diameter is practically independent of ratio; thus, only one curve is shown.

A discussion of the use of large hollow input gears is presented in Appendix C. It indicates that the roller gear drive can be adapted to many design variations. Different systems of cluster arrangement are optimum for different design specifications.

Figure 40 shows the estimated experimental drive cost versus horsepower for 20:1, 60:1, and 100:1 ratios. The costs are dependent upon the drive type. The 4-4 system is the least expensive and, of course, the 4-8-8 system the most expensive. In general, the costs are somewhat dependent upon the number of gears in each system. The costs include only the costs of the parts. Engineering effort, drawing preparation, and assembly are not included.

The estimated experimental drive tooling costs which are a function of horsepower and ratio are shown in Figure 41. As with the prototype cost, the tooling cost is a function of a number of different gears. Thus, the three row systems have the highest tooling costs. Again, the 4-4 system is the least expensive. The casting pattern cost for gear box walls and the gear tooling cost are included.

DESIGN FEATURES

The TRW roller gear drive has many design features. A few of these are high efficiency, low vibration and noise, long life expectancy, and high reliability.

Efficiency

The TRW Model AN-1 drive was extensively studied for efficiency at various speeds and loads in a test stand with two drives in the back-to-back arrangement. It can be safely estimated that efficiency data are accurate within $\pm$ 5 percent of the total losses. The test results, shown in Figure 42, show the total losses at full load to be approximately 2 percent. After modifications to the drive these losses were reduced to less than 1.5 percent.

The losses breakdown based on efficiency chart analysis is as follows:

One-gear contact average friction losses -0.35 percent.

Windage, oil splash losses at design speed and load -0.95 percent.

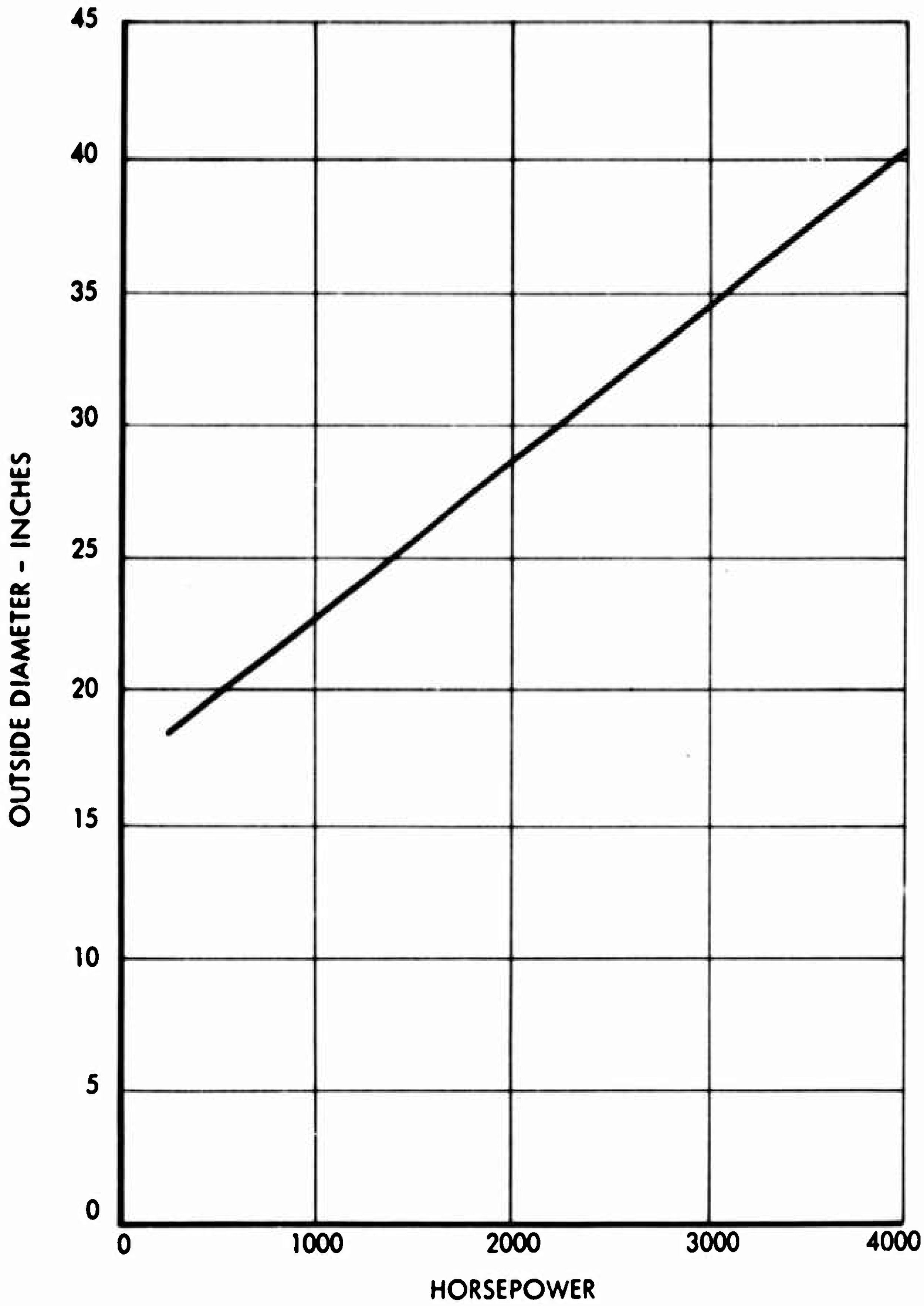


Figure 39. Estimated Outside Diameter Versus Horsepower - Rotor Output rpm.

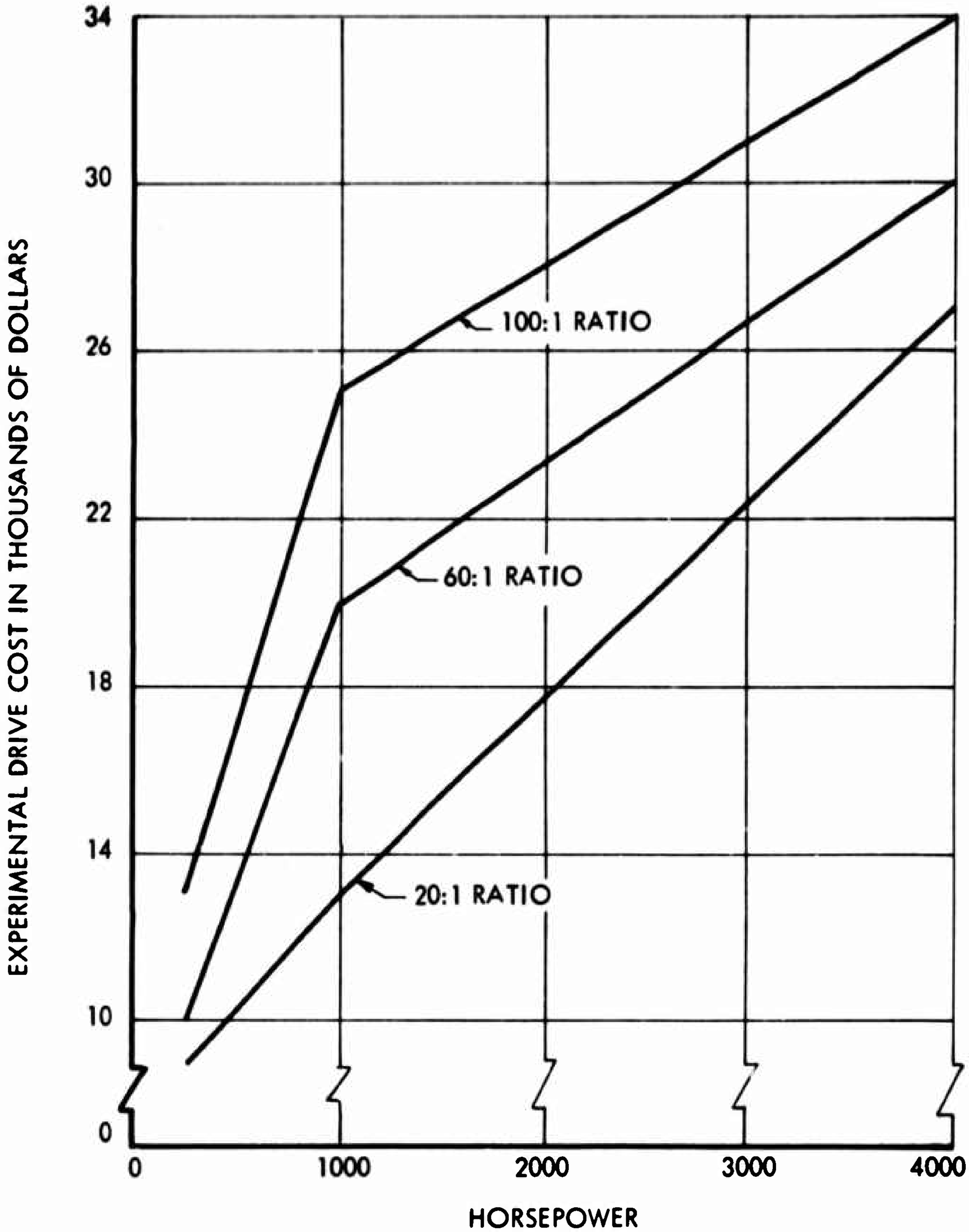


Figure 40. Estimated Experimental Roller Gear Drive Costs (Tooling Not Included).

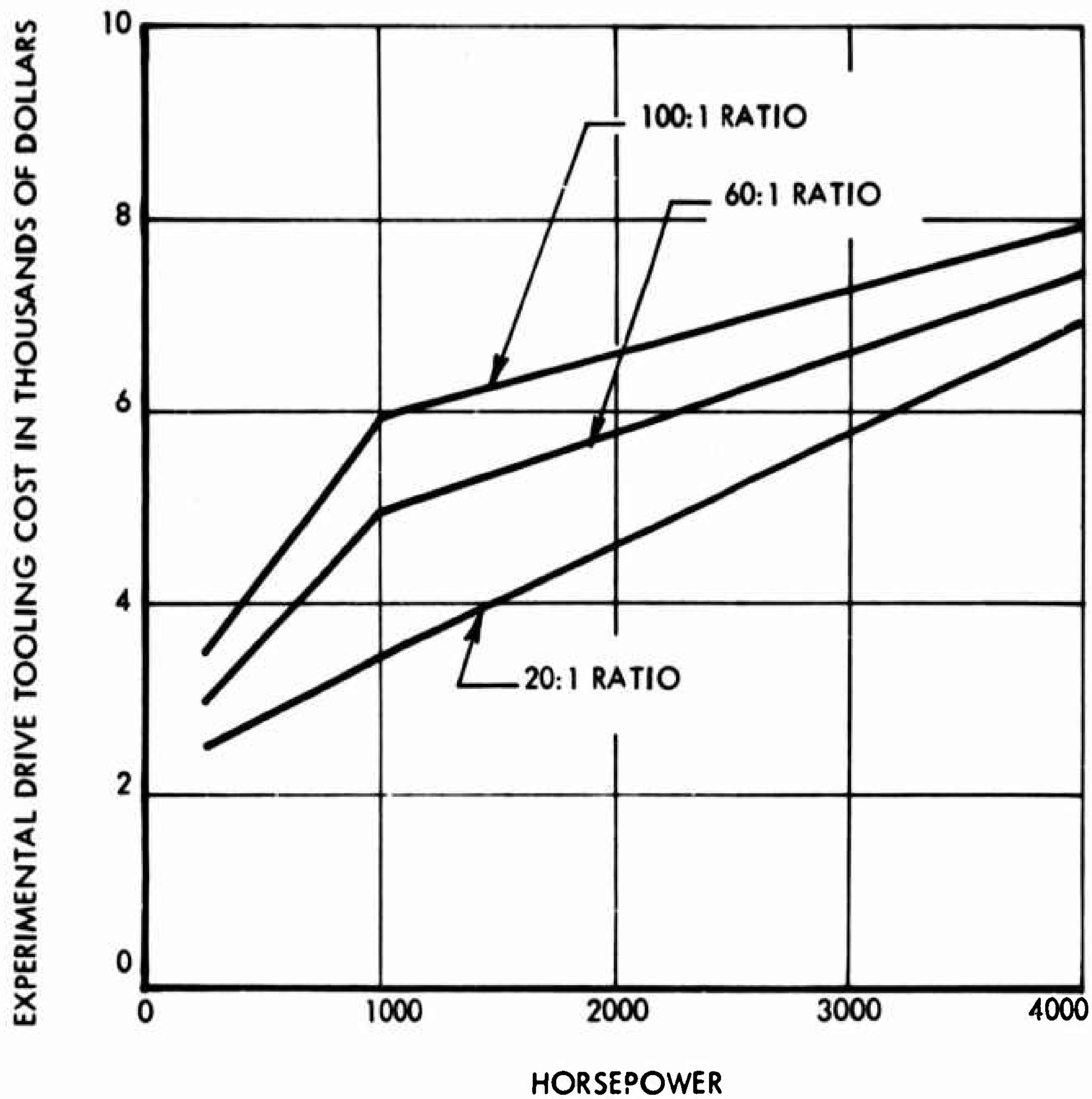


Figure 41. Estimated Experimental Roller Gear Drive Tooling Costs.

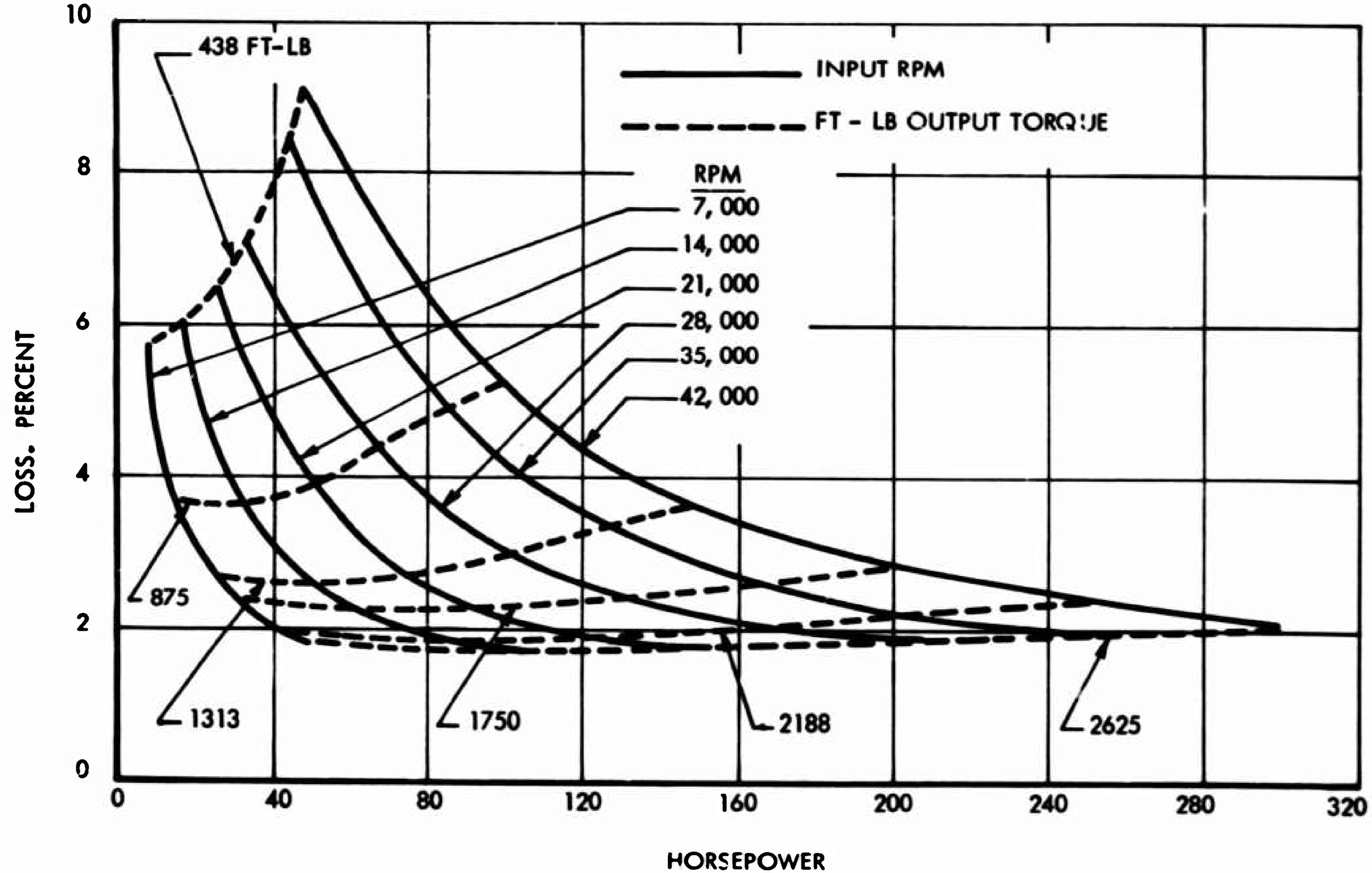


Figure 42. TRW 70:1 Ratio Roller Gear Drive; Percent Power Lost Versus Horsepower at Constant Input Speed and Constant Output Torque.

The windage losses in the roller gear drive are inherently very low for the stationary spider type of drive because the moving parts are small. In general, the windage losses are proportional to diameter to the 5th power, * as well as gear width and number.

If only one pinion were used on the output gear instead of the four used in the AN-1 drive, its diameter would have to be increased by the cubed root of four to accommodate the four times larger torque it must carry. The power to which the diameter is raised is obtained from the bending stress formula.

$$S_b = \frac{F_t \cdot P_d}{y \cdot (f.w.)} \quad \text{where } F_t = \frac{2T}{D_p} \text{ and } P_d = \frac{N}{D_p}.$$

Assuming the gear face width (f. w.) is increased proportionally to the pitch diameter (D_p), the diameter increase is then the cubed root of the torque increase ratio. Since the torque is four times larger for a single pinion, the diameter is increased by the cubed root of four. The windage losses increase would be

$$\frac{(\sqrt[3]{4})^5 \cdot \sqrt[3]{4}}{4} = 4 \text{ times.}$$

Thus, the roller gear drive, with its small multiple contact pinions, is inherently efficient.

Friction losses are lower due to very accurate gearing, accurate pitch line contact, and large diametral pitch (small tooth size, which is possible due to multiple contact). Additional savings are due to the replacement of high-speed bearings by simple rotating cylinders.

From the analogy based on the test data, it can be predicted that losses will be around 1.5 percent for all two row (three contact) drives of up to 750 horsepower with an input speed of not over 35,000 rpm. The drives in the 750 to 2,000 horsepower range will have 1.8 percent losses with 2 rows and 2.2 percent losses with 3 rows of planets (4 contacts). The three row roller gear drives with 4 contacts in the range of 2,000 to 4,000 horsepower will have 2.5 to 2.7 percent losses. All estimated losses are for design speed and power. Higher losses are estimated for higher horsepower due to the coarser gear pitch used.

---

*"Dudley, D.W., Gear Handbook - The Design, Manufacture and Application of Gears, First Edition, McGraw Hill Book Company, Inc., New York, New York, 1962, pages 14 - 20.

Vibration and Noise

The roller gear drive vibration features can be evaluated from AN-1 test data. The accelerometers mounted on the drive wall top show readings at all speeds less than 1 g. Higher frequencies were not filtered. The noise data are unavailable because the noise of the dynamometer located in the test cell did not permit recording. The drive has a distinct high-pitch sound, characteristic of all high-speed gear transmissions. The noise is originated by rapid air expulsion cycling in the gear mesh.

The vibration resulting from interaction of the gear contact and preloaded accurate rolling surfaces was expected to be low. The roller surfaces act as viscous and friction dampers. Press fitted cylindrical rings on the gear assembly damp out possible assembly disk radial and flexural modes. The stepped roller gear assembly is short and sturdy; therefore, torsional and flexural natural frequencies are very high. They are high enough so that the operating speeds will never reach them. Thus, no damage will occur owing to natural frequency vibration.

Life Expectancy

The bearings and gear surfaces are selected for a 2500-hour life. The rolling contact surfaces are good for a 10,000-hour life.

The roller gear cluster in the selected design types has the features of forced parallelism. In conventional planetary drives with longer shafts which are straddle-mounted, the ideal parallelism is impossible because of bearing location inaccuracies, load and thermal distortion in structural walls, and shaft bending.

The empirical K-factor values for gears include all of these inaccuracies, which yield uneven load distributions along the gear face. It is possible that due to high parallelism the roller gear drive can have up to 1.5 to 2 times longer life. Another life factor is even load distribution on contacts. The life expectations are based on a 1.2 overload factor due to uneven load distribution. Although not established analytically, it is possible that the drive has a small tolerance range and has a self-adjusting load sharing capability. If this is so, drive life will be prolonged again up to 1.5 times.

Because of multiple contacts, the drive is much better lubricated and cooled, since each contact has an oil jet on the outgoing side. In general, very high efficiency allows a prediction of long life, since surface damage is in direct relation to energy dissipation.

The shortest life is apparent in the output bearings, which have a 2500-hour B-10 life. Owing to low speed, predicted life is highly probable. With a small increase in weight, the bearing life can be prolonged, and there is reason to believe that the drive life can be matched to the best possible turbine life.

Reliability

In analyzing the reliability aspects of the roller gear assembly, comparison to a conventional high-speed gear reduction unit can be made. Although the number of gear mating surfaces has been increased in the new design, the functional operation has been changed. The number of bearings required and the problems of obtaining and maintaining accurate alignment have been reduced.

For comparison, consider the 4-8-8 roller gear system for a 4,000-horsepower drive. Table 4 shows a direct comparison with a conventional high-speed gear unit.

TABLE 4

COMPARISON OF A 4-8-8 ROLLER GEAR DRIVE WITH A CONVENTIONAL DRIVE

Drive Type	Number of Gear Mating Surfaces	Number of Gear Types	Number of Bearings
Roller Gear	36	6	8
Conventional Equivalent	21	8	25

Failure rate of roller gear assembly =

$$(F.R.)_{gears} \cdot \text{number of gears} + (F.R.)_{bearings} \cdot \text{number of bearings}$$

$$= (0.12 \cdot 10^{-6}) \cdot 36 + (0.65 \cdot 10^{-6}) \cdot 8$$

$$= 9.52 \cdot 10^{-6} \text{ hr.}$$

Failure rate of conventional gear reduction units =

$$= (0.12 \cdot 10^{-6}) \cdot 21 + (0.65 \cdot 10^{-6}) \cdot 25$$

$$= 18.77 \cdot 10^{-6} \text{ hr.}$$

For a 1000-hour mission

$$R_{\text{roller gear}} = e^{-t \cdot (F.R.)} = e^{-1000 \cdot 9.52 \cdot 10^{-6}} = \underline{\underline{0.99072}}$$

$$R_{\text{conventional unit}} = e^{-t \cdot (F.R.)} = e^{-1000 \cdot 18.77 \cdot 10^{-6}} = \underline{\underline{0.98123}}$$

The improvement in inherent reliability is approximately one order of magnitude, or 40 percent in terms of failure rates.

Qualitative reliability comparisons between the roller gear assembly and comparable conventional gear reduction units can be illustrated by the following:

1. Fewer bearings required. Alignment problems associated with gear train bearings are reduced considerably.

2. Elimination of high-speed bearings. The only bearings required are in the low-speed or outer row of planets.

3. Shorter shafts for greater rigidity. The unique design limits the number of mounting attachments required. The only attachments to the mounting fixture are the bearing shafts for the outer row of planets.

4. Less vibration and good dampening. The rolling cylinders dampen torsional vibration, and the short bearing shafts increase the natural torsional frequency to safer levels.

5. Less distortion. The spherical bearings on the outer row of planets do not transmit distortion from the mounting fixture to the gear cluster.

6. More accurate assembly. The cylindrical rollers can be manufactured to very close tolerances and the assembly is preloaded, so that accurate location can be assured.

7. Operation unimpaired by damaged or worn teeth. In the event that a tooth cracks or is worn, the load normally carried by that tooth is distributed over the remaining teeth in the respective gear.

8. Redundancy. The load transmitted by one planet gear in a set can be distributed over the remaining planet gears in the set.

9. Elimination of offset gearing problems. The normal distortion problems (e.g., thermal, load, and vibration) associated with the gear mounting requirements of conventional gear reduction assemblies are eliminated by the unique design of the roller gear assembly.

10. Elimination of heavy retaining fixture. Conventional high-speed gear reduction units require massive holding fixtures and internal fixtures (carriers) to insure accurate alignment during assembly and operation. The roller gear assembly design eliminates or reduces the requirements for these fixtures.

11. Less distortion due to internal friction. The higher mechanical efficiency of the roller gear assembly reduces the internal friction loss and attenoant temperature rise. The lower temperature also reduces the lubrication problems.

12. More easily mounted - Smaller overall size and reduced weight permit more ready application without modification. Where modification is required on conventional high-speed gear reductional assemblies to meet a mounting restriction, possible degradation of reliability can result.

DISTRIBUTION

US Army Materiel Command	12
US Army Mobility Command	9
US Army Aviation Materiel Command	5
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USAAVLABS Liaison Officer, US Army R&D Group (Europe)	1
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US Army Engineer Waterways Experiment Station	2
US Army Combat Developments Command Armor Agency	1
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APPENDIX I

BASIC EQUATIONS USED IN THE WEIGHT AND VOLUME ANALYSIS

The weight and volume analysis was performed by using the following basic equations. Variations were used where applicable.

GENERAL EQUATIONS FOR THE TWO ROW SYSTEM

For the split sun gear of three-face-widths length, the equation for D_p developed earlier in the report is

$$D_p^{\,5} = 8.857 \cdot 10^{-3} \cdot \frac{T^2}{K \cdot n} \cdot \frac{R+1}{R} \cdot \qquad (19)$$

If the torque is raised 20 percent to allow for misalignments, the equation becomes

$$D_p^{\,5} = 8.857 \cdot 10^{-3} \cdot \frac{(1.2T)^2}{K \cdot n} \cdot \frac{R+1}{R}$$

$$D_p^{\,5} = 1.275 \cdot 10^{-2} \cdot \frac{T^2}{K \cdot n} \cdot \frac{R+1}{R} \cdot \qquad (19a)$$

If the sun gear is not split, the equation becomes

$$D_p^{\,5} = \frac{1.275 \cdot 10^{-2}}{3} \cdot \frac{T^2}{K \cdot n} \cdot \frac{R+1}{R} \qquad (19b)$$

Tooth load on Y_1:

$$F_t = \frac{T \cdot x_1}{a \cdot n} \cdot \frac{1}{y_1 \cdot 2 \text{ contacts}}$$

Bending stress:

$$S_b = \frac{F_t \cdot P_d}{y \cdot (f.w.)}$$

with

$$P_d = \frac{18}{2y_1} \text{ for 18 teeth on } Y_1$$

$$y = .377 \text{ typical Lewis factor}$$

$$f.w. = 2y_1 \text{ for one piece gear}$$

$$S_b = 30,000 \text{ psi}$$

Thus,

$$30,000 = \frac{T \cdot x_1}{a \cdot n \cdot y_1 \cdot 2} \cdot \frac{18}{2y_1} \cdot \frac{1}{.377 \cdot 2y_1} .$$

Rearranging,

$$y_1^3 = \frac{18T \cdot x_1}{8 \cdot 0.377 a \cdot n \cdot 30,000} .$$

Raising the torque value by 20 percent to allow for overloads due to uneven load distribution,

$$y_1^3 = \frac{18 \cdot 1.2T \cdot x_1}{8 \cdot 0.377 a \cdot n \cdot 30,000}$$

$$y_1^3 = 2.3873 \cdot 10^{-4} R \frac{T}{n} \qquad (22)$$

For bending strength, with 18 teeth on the sun gear,

$$D_p 2 \cdot (f.w.) \geq \frac{\frac{2 \cdot (1.2T)}{n} \cdot 18}{0.377 \cdot 30,000}$$

$$D_p^2 \cdot (f.w.) \geq 3.820 \cdot 10^{-3} \frac{T}{n} . \qquad (23)$$

Bearing load (approximately):

$$P = \frac{1.2 \cdot T_o}{z_1 \cdot n}$$

where

$$T_o = R_o \cdot \text{input torque}$$

Thus,

$$P = \frac{1.2 \cdot R_o \cdot T}{z_1 \cdot n} \qquad (24)$$

Bearing speed:

$$N = \frac{\text{input rpm}}{R_o} \cdot \frac{c}{x_2} \tag{25}$$

Bearing dynamic load:

SKF spherical roller bearings are used for this analytical study. The basic dynamic load rating (C) is found from the C/P value determined from the SKF monograph and the calculated P value.

$$P = XVF_r + YFa \tag{26}$$

where

X = a radial factor = 1 for this case

V = a rotation factor = 1.2 for outer ring rotating in relation to the load

F_r = radial load calculated above (P)

F_a = thrust load = 0

$$C = \frac{C}{P} \cdot 1.2P$$

GENERAL EQUATIONS, THREE ROW SYSTEM

For all of the design types used in the analysis for the three row systems, the sun gear is split and equation (19a) applies. Since Y_1 is split, equation (22) must be altered by

$$\text{total f.w.} = 2.5y_1 = 1.25\ D_p$$

or a total face width of .625 D_p for each gear and

$$n_1 = \frac{n}{2} = \frac{\text{number of last row planets}}{2}.$$

For the 3-6-6 and 4-8-8 systems, the equation for the radius of Y_1 gear becomes

$$y_1^3 = \frac{18 \cdot 2T \cdot x_1}{4 \cdot 2.5 \cdot 3.77 \cdot a \cdot n \cdot 30{,}000}$$

$$= 3.183024 \cdot 10^{-4} \cdot \frac{T}{n} \cdot \frac{x_1}{a}.$$

Raising the torque value by 20 percent to allow for overload due to force unbalance,

$$y_1^3 = 3.183024 \cdot 10^{-4} \cdot \frac{1.2T}{n} \cdot R. \qquad (27)$$

The size of Y_2 is determined as follows:

$$F_t = \frac{T \cdot x_1 \cdot x_2}{n \cdot a \cdot y_1 \cdot 2y_2}$$

Substitute this in the stress formula:

$$S_b = \frac{F_t \cdot P_d}{y \cdot (f.w.)}$$

along with these:

$$P_d = \frac{18}{2y_2} \text{ for 18 teeth on } Y_2.$$

$$y = .377 \text{ typical Lewis factor}$$

$$f.w. = 2y_2 \text{ for a one piece gear}$$

$$S_b = 30,000 \text{ psi}$$

the value of y_2 can be found

$$30,000 = \frac{T \cdot x_1 \cdot x_2}{n \cdot a \cdot y_1 \cdot 2y_2} \cdot \frac{18}{2y_2} \cdot \frac{1}{0.377} \cdot \frac{1}{2y_2}$$

$$y_2^3 = 1.98939 \cdot 10^{-4} \cdot \frac{T}{n} \cdot \frac{x_1}{a} \cdot \frac{x_2}{y_1}$$

Multiplying the torque by 20 percent for possible load unbalance,

$$y_2^3 = 1.98939 \cdot 10^{-4} \cdot \frac{1.2T}{n} \cdot R \cdot \frac{x_2}{y_1}. \qquad (28)$$

The bearing equations are the same except x_2 becomes x_3 and z_1 becomes z_2.

The overall ratio equation becomes

$$R_o = \frac{c}{a} \cdot \frac{x_1}{y_1} \cdot \frac{x_2}{y_2}.$$

APPENDIX II

SAMPLE WEIGHT AND VOLUME STUDY CALCULATIONS FOR 1,000 HORSEPOWER

The weights and volumes for 37 designs were calculated during the study. The following calculations serve as a sample. They are for 1,000 horsepower and a 100:1 reduction ratio point. Figure 43 shows a schematic layout of this drive.

DIMENSIONS

The dimensions vere determined using the equations developed in Appendix A and through geometr. : considerations. The particular design is a 4-8-8 system.

Torque

$$T = \frac{hp \cdot 63,000}{rpm}$$

$$= \frac{1,000 \cdot 63,000}{21,000}$$

$$= 3,000 \text{ inch-pounds}$$

K-Value

Our charts show a K-value = 522 for 2500-hour life for the sun gear with four first row planets in contact with it.

Pitch Diameter of Sun Gear

Equation (19a) becomes

$$D_p^5 = 1.275 \cdot 10^{-2} \cdot \frac{T^2}{K \cdot n} \cdot \frac{R+1}{R} \tag{19a}$$

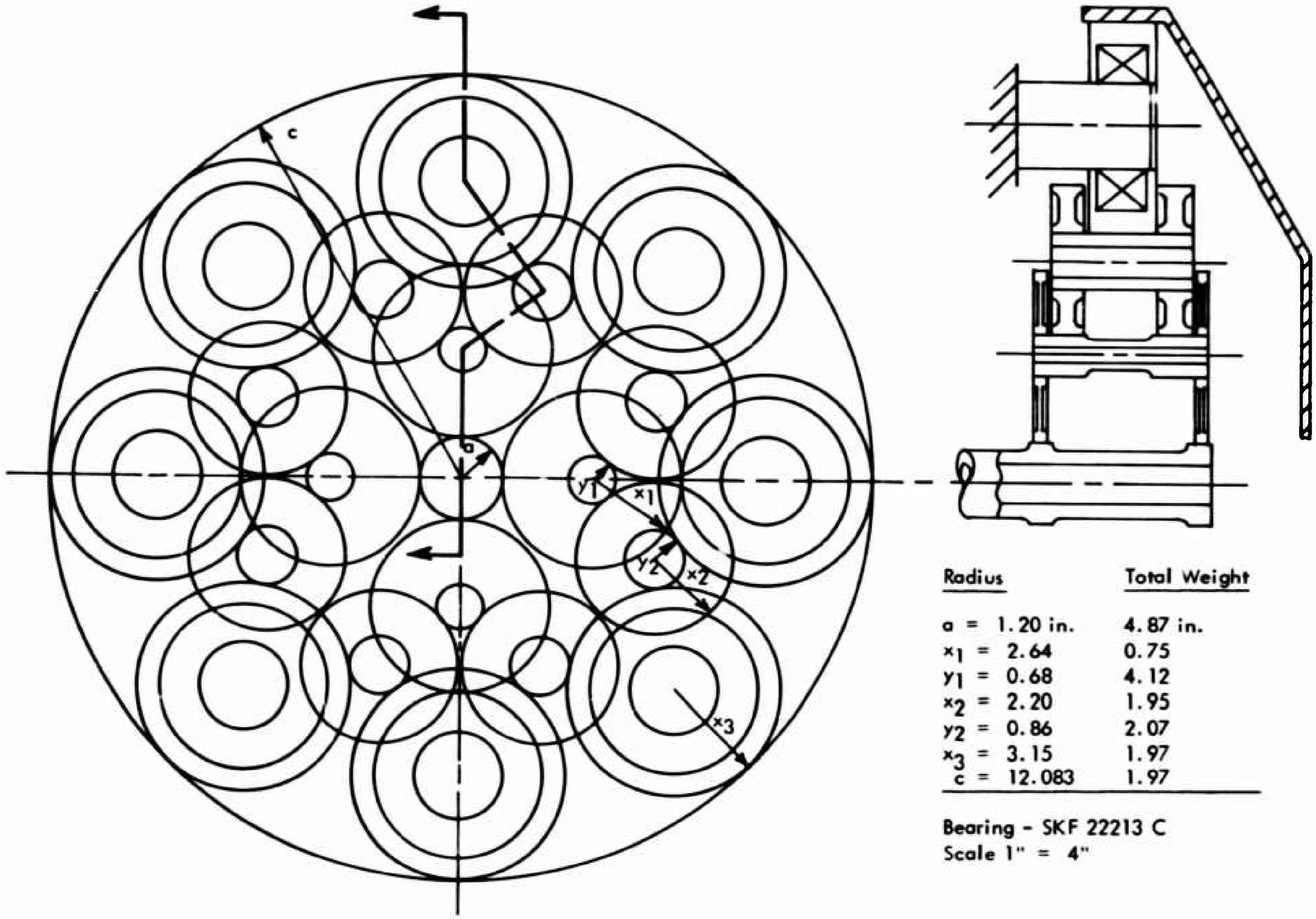
$$D_p^5 = 54.96 \cdot \frac{R+1}{R}.$$

Pitch Radius of Y_1

Equation (27) becomes

$$y_1^3 = 3.183024 \cdot 10^{-4} \cdot \frac{1.2T}{n} \cdot R \tag{27}$$

$$= .143 \text{ R in}^3 \text{ for } n = 8.$$



Radius	Total Weight
a = 1.20 in.	4.87 in.
x_1 = 2.64	0.75
y_1 = 0.68	4.12
x_2 = 2.20	1.95
y_2 = 0.86	2.07
x_3 = 3.15	1.97
c = 12.083	1.97

Bearing - SKF 22213 C
Scale 1" = 4"

Figure 43. Drive Design Layout for 1000 Horsepower, 100:1 Ratio, 21,000-rpm Input Speed.

Pitch Radius of Y_2

Equation (28) becomes

$$y_2^3 = 1.98939 \cdot 10^{-4} \cdot \frac{1.2T}{n} \cdot R \cdot \frac{x_2}{y_1} \quad (28)$$

$$= .0895\ R \cdot \frac{x_2}{y_1}\ \text{in.}^3$$

Face Width of X_1

Equation (23) becomes

$$D_p^2 \cdot (\text{f.w.}) \geq 3.82 \cdot 10^{-3} \cdot \frac{T}{n} \quad (23)$$

$$\text{f.w.} \geq \frac{2.865}{D_p^2}\ .$$

100:1 RATIO (4-8-8) DIMENSIONS

For the 100:1 ratio it was discovered after several trials that the ratio between the gears in the first contact is limited to 2.2:1. The limiting geometric factors are $x_1 - x_1$ and $x_2 - x_2$ interference.

The sun gear size is:

$$D_p^5 = 54.96 \cdot \frac{3.2}{2.2}$$

$$= 79.942\ \text{in}^5$$

$$D_p = 2.4\ \text{inches}$$

$$a = 1.2\ \text{inches}$$

$$x_1 = 1.2 \cdot 2.2 = 2.64\ \text{inches}$$

The size of Y_1 is:

$$y_1^3 = .143 \cdot 2.2$$

$$= .315\ \text{in}^3$$

$$y_1 = .68\ \text{inch}$$

The face width of Y_1 is:

$$(f.w.)_{y1} = 2.07 + 2.5 \cdot .68 + .25 + .1$$

$$= 4.12 \text{ inches including shaft, rollers, and clearances}$$

The size of Y_2 is:

$$y_2^3 = .0895 \cdot 2.2 \cdot \frac{2.2}{.68}$$

$$= .637 \text{ in}^3$$

$$y_2 = .86 \text{ inch}$$

The face width of Y_2 is:

$$(f.w.)_{y2} = 2 \cdot .86 + .25 + .1$$

$$= 2.07 \text{ inches including rollers and clearances}$$

The ring radius c found from the overall ratio equation (23) is:

$$c = \frac{100}{2.2} \cdot \frac{.68 \cdot .86}{2.2}$$

$$= 12.083 \text{ inches}$$

The face width of X_1 found from equation (23) is:

$$(f.w.)X_1 = \frac{2.865}{(2.4)^2}$$

$$= .4974 \text{ inch}$$

Use .25 inch for each side of X_1.

BEARING SELECTION

$$x_3 = 3.15 \text{ inches from layout}$$

$$z_2 = 8.9 \text{ inches}$$

Using equation (24) altered for the three row system,

$$P = \frac{1.2 \cdot R_o \cdot T}{z_2 \cdot n}$$

$$= \frac{1.2 \cdot 100 \cdot 3,000}{8.9 \cdot 8}$$

$$= 5,056 \text{ lb.}$$

and equation (25) altered for the three row system

$$N = \frac{\text{input rpm}}{R_o} \cdot \frac{c}{x_3} \qquad (25)$$

$$= \frac{21,000}{100} \cdot \frac{12,083}{3.15}$$

$$= 805 \text{ rpm}$$

C/P is equal to 4.2 from the SKF nomograph for 2500-hour life.

$$C = \frac{C}{P} \cdot 1.2P$$

$$= 4.2 \cdot 1.2 \cdot 5.056$$

$$= 25,482 \text{ lb.}$$

For this dynamic load value a SKF 22213c bearing can be used. Its dimensions are:

bore diameter	=	2.5591 inches
outside diameter	=	4.7244 inches
bearing width	=	1.2205 inches
weight	=	3.4 pounds

VOLUME ESTIMATE

$$\text{Vol.} = \pi\ (12.5)^2 \cdot 5.12$$

$$= 2,513 \text{ in.}^3$$

WEIGHT ESTIMATE

1. Sun Gear

$w = .283\pi\left[(1.2)^2 - (.75)^2\right] 4.87$ 3.799

2. First Row Gears

For X_1

$4 \cdot .283\pi\left[(2.64)^2 - (.375)^2\right] .75 = 18.214$

minus

$4 \cdot .283\pi\left[(2.4)^2 - (.9)^2\right] .375 = -6.601$

For Y_1

$4 \cdot .283\pi\left[(.68)^2 - (.375)^2\right] 4.12 = +4.715$ 16.328

3. Second Row Gears

For X_2

$8 \cdot .283\pi\left[(2.2)^2 - (.5)^2\right] 1.95 = 63.661$

minus

$8 \cdot .283\pi\left[(1.95)^2 - (1.1)^2\right] .98 = -18.071$

For Y_2

$8 \cdot .283\pi\left[(.86)^2 - (.5)^2\right] 2.07 = 7.208$ 52.798

4. Third Row Gears

For X_3

$8 \cdot .283\pi\left[(3.15)^2 - (2.3622)^2\right] 1.97 = 60.846$

minus

$8 \cdot .283\pi\left[(2.90)^2 - (2.5)^2\right] 1 = -15.363$ 45.483

5. Bearings

$8 \cdot 3.4$ 27.200

6. Ring Gear

$.283\pi\left[(12.333)^2 - (12.083)^2\right] 1.97$ 10.691

156.299

$$\text{Total Wt.} = \frac{156.299}{.65} = 240.5 \text{ lb.}$$

APPENDIX III

LARGE HOLLOW SUN GEAR DESIGN

The parametric study was directed by the objective to obtain the maximum ratio in one drive (from high-speed turbine to slow-speed rotor) with a minimum number of roller gear planets within the smallest possible outside diameter envelope. Practically, a drive of this type would be primarily for a vertical helicopter turbine with the elimination of the bevel gear contact. This concept will decrease drive and support structure weight while increasing drive efficiency.

But up to date the vertical turbine has not been used in helicopters, and its application would require major helicopter redesign. The roller gear drive can be easily adapted to a horizontal turbine with somewhat less than the total reduction ratio being taken through it. The reason is that the high-speed bevel contact reduces the input speed to the drive by the ratio of 1.5:1 up to 5:1.

A majority of the gearboxes in present helicopter design have a large and hollow input sun gear to accommodate the main rotor shaft extension through it. The longer rotor shaft, and consequently larger span between rotor support bearings, requires smaller bearing reactions and, in general, provides a more stable rotor shaft.

A study of the roller gear drive with the larger size sun gear shows no weight penalty for this type of design. A further study of it led to the conclusion that the lightest roller gear type is not with the least amount of planets in a row but rather with the largest possible number of roller gear planets per row.

Consequently, the drives with the larger sun gear and many planets per row are lighter because the planets are smaller, rotate faster, and carry less torque each for the same drive horsepower. The ring gear is especially lighter because the bending moment due to tooth separation forces is decreased by two factors. One is the smaller force and the other is the smaller moment arm owing to the larger number of forces. Another weight savings is realized from the bearings because their weight decreases faster than their load-carrying capacity. A further weight savings is in the structure due to decreasing bending moment in each bearing support.

There is a question as to the present practicability of the drive with 3, 4 and more times the total number of roller gear elements due to prohibitive costs of the present manufacturing and assembly methods. There are some new alleys of investigation of manufacturing and assembly methods that could lead to great savings. These were only recently discovered and thus they are out of the scope of this report. Only the results of a preliminary exploratory investigation are included here.

Three variants of a 1,000-horsepower, 21,000 rpm input, 20:1 reduction ratio drive are given here for comparison to illustrate the possible weight savings by use of more planets per row inherent in the large sun gear design. The 4-4 drive shown in Figure 44

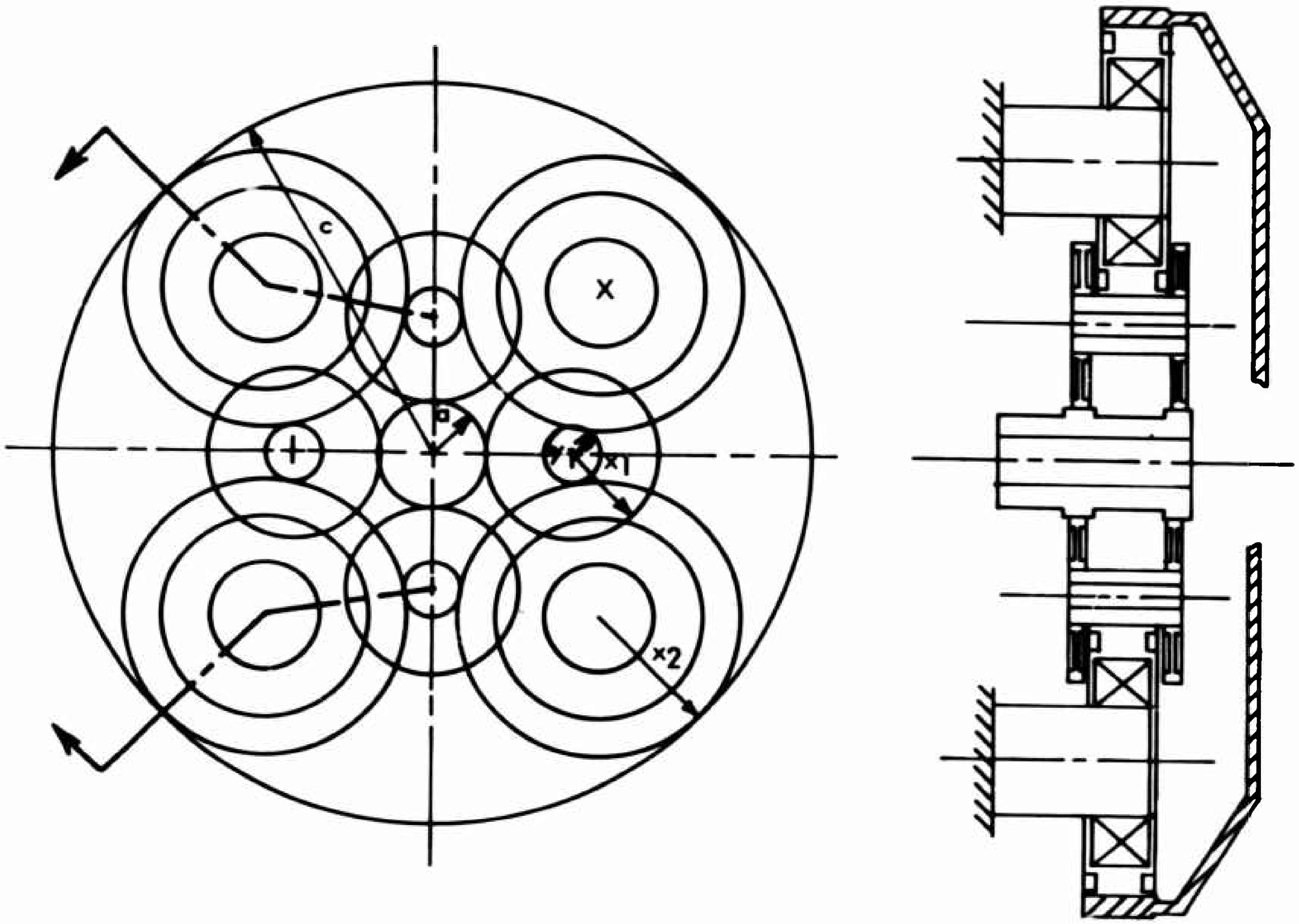


Figure 44. Small Sun Gear Drive; 20:1 Ratio, 21,000-rpm Input Speed,4-4 Drive.

represents the smallest possible sun gear design concept with the fewest number of parts. The 8-8 drive shown in Figure 45 represents the enlarged sun gear design concept. And the 12-12-12 drive shown in Figure 46 represents the optimum enlarged sun gear design concept.

In these illustrations all gear teeth were calculated for the same maximum bending stress of 25,000 psi and the respective ring gears for the same bending stress of 25,000 psi. Table 5 shows a summary of the resulting dimensions.

TABLE 5

DRIVE SYSTEM COMPARISON DIMENSIONS

	4-4		8-8		12-12-12	
	Radius	Width	Radius	Width	Radius	Width
a (inches)	1.231	0.750	2.750	0.25	2.625	0.187
x_1	1.908	0.750	1.500	0.25	0.750	0.187
y_1	0.653	1.556	0.602	0.375	0.387	0.275
x_2	3.100	1.556	2.407	0.375	0.986	0.275
y_2	-	-	0.794	0.813	0.435	0.531
x_3	-	-	-	-	1.400	0.531
y_3	-	-	-	-	0.748	0.656
c	8.426	1.556	7.250	0.813	6.500	0.656

The results of the comparative calculations assuming all gear webs have no lightening holes except in the space allowed for the bearings in the last-row gears are shown in Table 6.

The larger gears in the 4-4 drive can be made much lighter in relation to the original weight than can the smaller gears in the 8-8 and 12-12-12 drives. The reason is that they have more areas that may be lightened. Nevertheless, the 12-12-12 drive will remain more than twice lighter than the 4-4 drive.

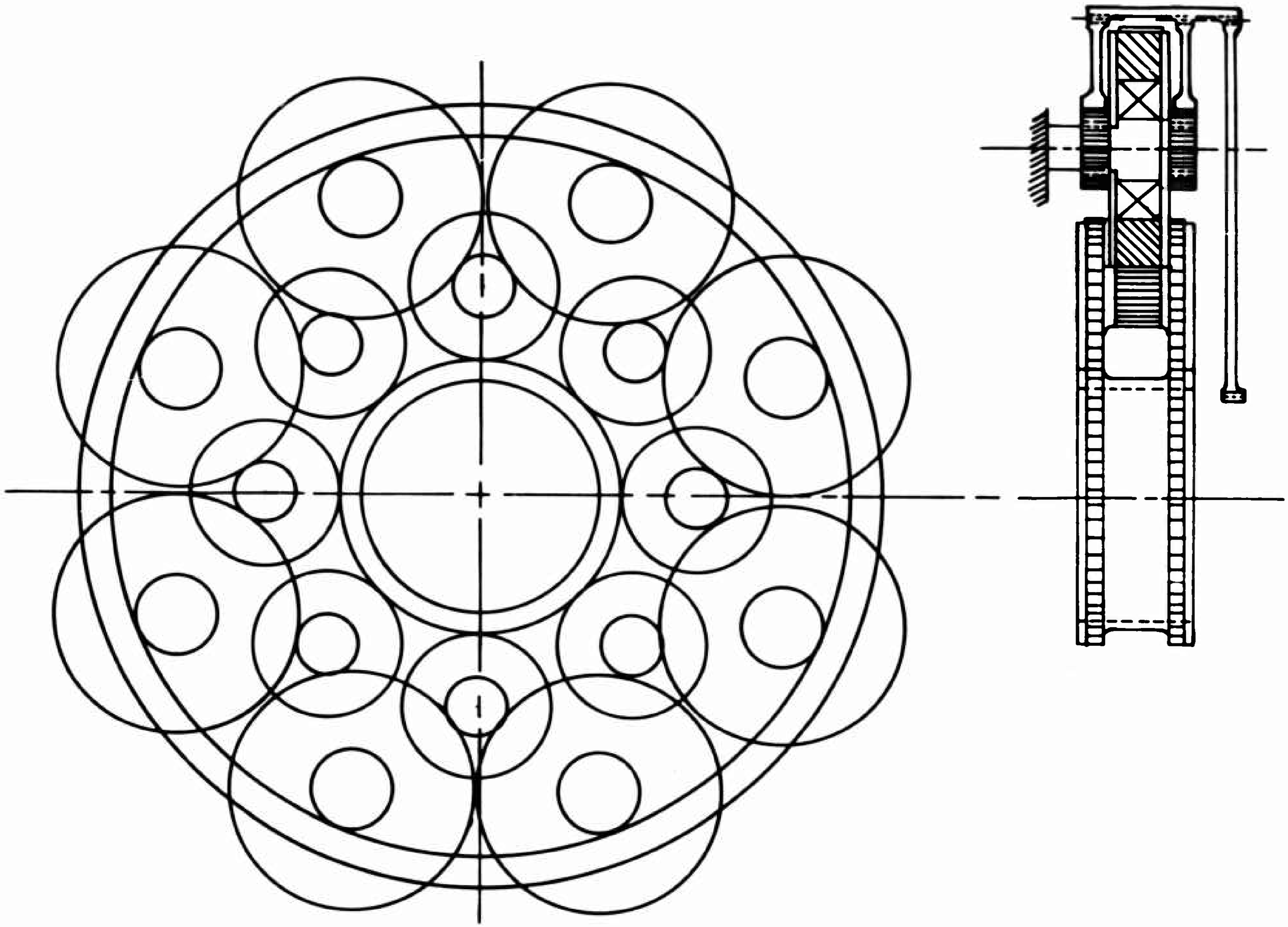


Figure 45. Enlarged Sun Gear Drive; 1000 Horsepower, 20:1 Ratio, 21,000-rpm Input Speed, 8-8 Drive.

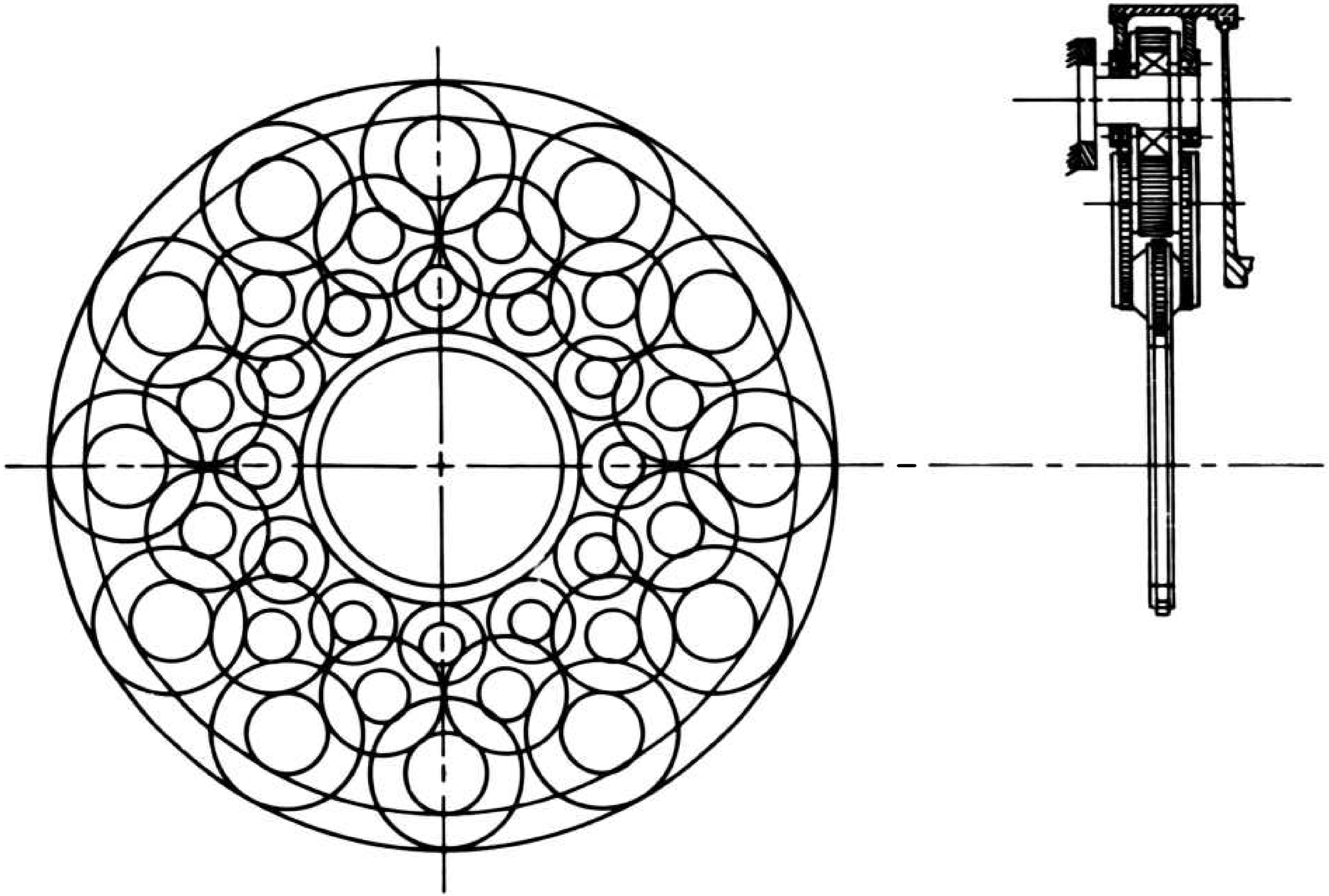


Figure 46. Enlarged Sun Gear Drive; 1000 Horsepower, 20:1 Ratio, 21,000-rpm Input Speed, 12-12-12 Drive.

TABLE 6

DRIVE SYSTEM COMPARISON WEIGHTS

	4-4	8-8	12-12-12
Sun Gear	1.00 lb	0.35 lb.	0.35 lb.
Intermediate Gears	41.47	22.00	15.46
Ring Gear Without Spider	15.18	4.24	2.16
Bearings (No. of)	13.60(4)	7.60(8)	6.60(12) [5.28(24)]
Totals	71.25	34.19	24.57 or [23.25]

The 12-12-12 drive has one more gear contact than the 4-4 drive. This is due to the extra row of planets. The losses on one contact of the roller gear drive are .3 to .4 percent. For 1,000 horsepower they are then 3 to 4 horsepower. Since each horsepower loss is equivalent to 8 pounds' lifting weight, the added loss due to the extra row of rollers is from 24 to 32 pounds. Therefore, it appears that the optimum drive in this case is the 8-8 drive with about 50 percent apparent weight savings over the 12-12-12 drive.

These preliminary comparisons look hypothetical because the first row of rollers in the 12-12-12 would rotate faster than 60,000 rpm. But all three drives would have the same bevel gear reduction, so the comparison is still valid although the values given in Table 6 are not. With the bevel gear reduction prior to the roller gear drive in the drive train, the 2 to 3 times speed increase in the first row planets would not increase the rpm and surface velocity beyond practical limits.