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The robotics and manufacturing industries rely on high precision speed reducers to convert low motor torques to high output torque without degrading the tool position. SYNKINETICS Inc. has developed a new and innovative technology that uses flat plate cams with balls at conjugate locations transmitting and amplifying torque. This SYNKdrive technology is then coupled to a special machining operation on the cam set to produce a zero-backlash speed reducer. The analysis and data produced thus far show the zero-backlash SYNKdrive™ technology as a very versatile, low cost alternative to the products generally on the market today. The data have shown low backlash with good repeatability and low hysteresis. The analytical predictions show opportunities for improvement in a follow on program. SYNKdrive zero-backlash systems testing and analysis represent a major step in applying a new precision speed reducer technology to the robotics and manufacturing industry.

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PRECISION SPEED REDUCER FOR ROBOTICS AND MANUFACTURING

FINAL REPORT

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1.0 Background

1.1 Phase I Summary

This is the final report for SYNKINETICS, Inc., contract DASG60-94-C-0098, Phase I study of application of SYNKdrive technology to manufacturing and robotics. Within this Phase I effort we have built a manual SYNKdrive "ZBcam" zero backlash prototype which proves the basic technology necessary for development of a full-scale demonstration of SYNKdrive zero backlash performance.

Looking to the future, the SYNKdrive rolling action speed reducer is useful in many commercial applications. Examples of SYNKdrive precision positioning equipment include precision robotic joints and positioners and high precision positioning tables. SYNKINETICS intends to manufacture such precision products for industry incorporating our zero backlash cam design. These SYNKdrive devices will be less expensive to manufacture and assemble and will feature higher accuracy and longer life than the components being replaced.

1.2 Background to SYNKdrive Technology

1.2.1 Introduction to SYNKINETICS Inc.

SYNKINETICS has developed a new and innovative speed conversion technology aimed at the commercial market. Our basic speed reducer is gearless. It uses balls and cam tracks cut in the faces of flat disks to achieve speed and torque conversion as it simulates a rotating bearing. The cams can be designed to yield any required reduction at constant, varying, intermittent or oscillatory output from a conventional input. We refer to our devices as SYNKdrive cam devices and SYNKdrive speed reducers. We refer to our zero backlash cam design as the ZBcam.

Our SYNKdrive technology is new. We have not sold or offered for sale any products. We have now built a prototype of our zero backlash ZBcam SYNKdrive speed reducer, the latter embodiment being an essential part of this project. This project was the first major opportunity for SYNKINETICS to study and characterize the basic SYNKdrive ZBcam technology discussed herein. SYNKINETICS two issued patents on the SYNKdrive technology, and several other patent applications are pending, U.S. and foreign.

1.2.2 The Basic SYNKdrive Technology

SYNKdrive devices do what gear assemblies do, but better. Gear assemblies are generally considered to be relatively efficient tools for many purposes, but they still suffer inherent limitations because gears interact with each other only one tooth at a time. Meanwhile SYNKdrive devices are highly effective because all of the "cycles" (i.e., "teeth") of the cams work together simultaneously and synchronously, not one at a time.

Furthermore, SYNKdrive devices can obtain high precision, are relatively easy to manufacture, and can reach operating efficiencies perhaps as high as 95%. By comparison, precision worm gear assemblies are relatively expensive to manufacture and operate at about 30-50% efficiency.

As shown in Figure 1.2.2-1, every SYNKdrive speed reducer includes an input drive disk 1 and an output driven disk 2. A drive cam track 3 (having at least one cycle) is formed on the face of the drive disk. A driven cam track 4 (having several cycles; 12 cycles are shown) is formed on the face of the driven disk. Also included is a reaction disk 5 which has slots 6 for receipt of balls 7. Reaction disk 5 is typically grounded to the speed reducer housing 8.

The drive disk 1 is connected to an input shaft 9. In operation, rotation of input shaft 9 rotates drive disk 1 and its cam track 3. Rotation of cam track 3 in turn displaces balls 7 radially in or out in the slots 6. These moving balls then drive the driven cam track 4, which turns the driven disk 2 and output shaft 10 connected thereto.

The balls do not drive when they are in transition, which occurs one ball at a time and instantaneously, so that their change of direction is insignificant. The speed reducer input can be driven either clockwise or counterclockwise, but the basic speed reducer is irreversible from the output. However, in another arrangement of the speed reducer, the reaction disk is selectively ungrounded to provide a clutch function and to enable reversibility from the output.

Kinematic optimization of the speed reducer is achieved by designing the cam curvatures such that the balls synchronously interact with every cycle of the drive and driven cams at optimized contact angles, all at the very same time, with equal effect and with low frictional losses. Thus our speed reducers overcome both the frictional losses and the one-tooth-at-time deficiencies of gears.

Multiple SYNKdrive speed reducer stages can be mated in a single device for very high reduction ratios. In fact, the device of Figure 1.2.2-1 can be expanded by addition of "interim" disks. As shown in Figure 1.2.2-2, a first stage includes the drive cam 2 of disk 1 cooperating with a new driven cam 11 on a first side of interim disk 12. A second stage includes a new drive cam 13 on the other side of the interim disk 12 cooperating with driven cam 4 of disk 3. A second reaction disk 5A is installed for the second stage, and this automatically expands the housing to accommodate the extra interim disk. As the input shaft 9 is rotated, drive cam 3 drives driven cam 11 which turns interim disk 12 and which turns drive cam 13 which turns driven cam 4 of the driven disk 2 and thus drives the output shaft 10.

The speed reduction ratio is simply computed as the number of cycles of the driven cam track compared to the number of cycles of the drive cam track for each stage. In Figure 1.2.2-1, the ratio is 12:1, but it could be any ratio, whole number or fractional. For example, a three cycle drive cam and a ten cycle driven cam would yield a 3 1/3:1 reduction ratio. In a multiple-stage assembly, the reduction ratios for all of the stages are multiplied. For example, a SYNKdrive speed reducer having four stages of 50:1 each has a total reduction of 6,250,000:1. It is expected that this is achievable with zero backlash, based upon the cam design discussed below. At about 95% efficiency for each stage of a speed reducer unit incorporating our zero backlash cam design, a four stage unit would operate at perhaps 75-80% efficiency with zero backlash at any operating speed.

The significant features of the SYNKdrive speed reducer can be appreciated by reviewing the balls 7 and reaction disk 5 configuration shown in Figure 1.2.2-1. The eleven balls are all participating in transmitting the torque from the drive cam to the driven cam. Accordingly, the stiffness of the drive is predicated on the deflection of the eleven balls acting in parallel and sharing the load essentially equally. The deflection of the balls under their normal load is designed to be greater than the precision of the CNC machining tolerances (i.e., 0.0005 deflection versus 0.0002 CNC tolerances).

Accordingly, as the first two or three balls accept load and start deflecting the other balls because of their precise position will unload the first balls and accept loading until total load is dissipated over all of the balls, thereby normalizing the unit load on each ball and each ball should be essentially equally deflected.

The torsional spring constant of the eleven balls in parallel will be exceptionally high and the loads essentially will be equally distributed. There will be an insignificant radial load as they essentially are equally opposing loads. Therefore the central bearings will not experience significant radial loads.

Once in their deformed state and with essentially equal load sharing, a stable condition is achieved and the load will be essentially shared by all the balls. The only variation occurs on each ball individually when the balls reverse direction in the slots. The ball becomes unloaded for such a short duration that it is essentially instantaneously unloaded and then loaded at the peaks of the multi-cycled driven cam.

A zero backlash SYNKdrive speed reducer is shown in Figure 1.2.2-3 having dual sets of concentric cam tracks T1 and T2. We refer to these cams as "ZBcams". The ZBcams are formed on both the drive and driven cam disks, with sets of balls 7A, 7B captured in between. Each cam track is essentially a single flat flank so that each ball is held by two-point contact, once by the flank of the drive cam track and once by the flank of the driven cam track. One of the concentric sets is for driving clockwise and the other is for driving counterclockwise. Axial preloading provided by the compliance member 14 is greater than the separation forces experienced by the balls/cams and assures intimate contact between the cam flanks and the balls for zero backlash. The amount of preloading is determined by the torque throughput and the resulting forces that want to separate the drive and driven cam tracks. An additional 10-20% preload is applied above the calculated separation force to prevent separation. The axial preload accommodates thermal changes, which would be insignificant relative to spring deflection and would therefore not affect the magnitude of the preload.

1.3 The ZBcam and Phase I

The Phase I study was intended to investigate the analytical, design and fabrication requirements for development of a SYNKdrive zero backlash rotary drive unit based on SYNKINETICS' proprietary speed reducer technology. A concurrent engineering program was established to integrate all of the requirements at program start-up such that at the PDR all disciplines were satisfied and efforts proceeded toward the final Phase I design, production of a manual prototype, and preparation of the final report. A Phase II proposal is currently being written.

The manually controlled prototype shown in Figure 1.3-1 was designed and built to illustrate the ZBcam design. To support the foregoing, the following design and analytical efforts were included: (1) Selection of bearings based on torque throughput and output shaft load capability; (2) Analysis of compliance and torsional spring constant; (3) Analysis of Hertz contact stresses based upon the mean compressive stress allowable for the material and hardness; (4) Confirmation of operation of elements of ZBcam drive; (5) Confirmation of ball sizing; and (6) Generation of commands for CNC machining. The cams were cut in stainless steel.

Our concurrent engineering integrated the design, engineering, analysis, manufacturing and testing efforts in a six month program that was supported with our own proprietary analytical software, SYNKcad, and an in-house test program. As a result we have demonstrated the precision and backlash performance of the SYNKdrive ZBcam rotary actuator.

The results of this team effort are most gratifying and are presented in detail below. We believe, as a result of this program, that we have made a significant step forward in the field of high precision rotary actuators. Much has been learned in the process of producing the ZBcam prototype. We are convinced that this Phase I effort has provided Synkinetics with a solid foundation for proceeding with further development as will ultimately produce an ultra-precision rotary actuator. This ZBcam drive unit will enable a higher level of commercial and military precision operation.

2.0 ZBcam Development Program

Initially, a concurrent engineering program was established wherein all the team members submitted requirements, a time line established and the areas of responsibility and specifications were assigned. The following items were identified for study:

- Establish a reduction ratio that would provide significant torque throughput and sensitivity for the test program.
- Parametric analysis to delineate contact stresses, torsional stiffness and pre-load considerations.
- A test program to demonstrate precision and backlash capability. Also, if the budget allows friction and axial pre-load verification.
- Specifications and manufacturing procedures for developing the ZBcam and assembly procedures.
- Design of a precision and robust assembly with special attention to the spindling of the rotating cams and the precision alignment of the reaction disk and the bearings that will define the center of rotation.

2.1 Engineering Effort

The principal factors considered in the design of the test prototype ZBcam SYNKdrive rotary actuator were:

1. Reduction ratio
2. Verification tests
3. Spindle design
4. Pre-load consideration
5. Torque throughput
6. Torsional Stiffness

Obviously, there is considerable inter-relationship of all these factors in the design of the prototype, however, we considered items 1 and 2 perhaps the most important functions for this program. With the cost constraints of this research effort and the sensitivity of test measurements required at this level of precision, a preliminary design was established.

A reduction ration of 37:1 with a ball compliment of 36, 5/32 in. diameter balls was established. This arrangement was decided upon to permit us to study what impact the ball compliment would have using 12, 24, and 36 balls configurations. Factors of contact stress, torsional stiffness, torque throughput and RMS tolerance averaging would benefit from the highest ball compliment. Friction, manufacturing and cost would benefit from the lowest number ball compliment. Finally, the high ratio of 37:1 would be sensitive enough to allow a reasonable measurement test program within the cost constraints of this research grant.

A verification test program was evolved to illustrate both the zero backlash capability and repeatability of the ZBcam design. Within the scope of this research grant a test procedure and test bed were established to demonstrate precision, repeatability and zero backlash of less than 10 arc. seconds. The verification testing included a fixture and dial indicators capable of 0.00005 in. precision with higher theoretical sensitivities than required.

The manufacturing requirements were established for the ZBcam test device and considered in the ultimate design. The spindle, assembly and tolerencing were critical to achieve the run-out precision, low backlash and smooth running. This design is illustrated in Appendix A.

The analysis of the design was performed using our proprietary SYNKcad program. This software was developed by Synkinetics and enabled investigating the parameters of the ZBcam configurations. The comprehensive analytical capability of this software provides considerable insight into the kinematics of the prototype including: contact stress, torsional stiffness, torque throughput, and separation forces. The SYNKcad program provided the above kinematic solution for all the balls simultaneously at any given angle. This allowed us snapshots indicating the prevailing conditions for all ball compliments.

The machined parts were outsourced to an independent manufacturing facility where close supervision and due diligence produced the prototype unit. The Computer Numeric Control (CNC) programs were written in -house and made available to the outsource manufacturer. The unit was assembled under direction of the primary investigator and returned to the laboratory for testing.

The test program involved developing fixtures and mechanisms to produce meaningful data. The testing included instrument calibration, backlash and position repeatability measurements and torsional stiffness and separation force analyses.

2.2 ZBcam Analysis

2.2.1 Analysis Modeling Assumptions

The basis of the SYNKdrive analytical studies was completed in mathematical formulations defining the internal kinematics and load transferring capabilities for various configurations. Inherent in the mathematics of contact mechanics, strength of materials approaches and machine kinematics, there exist assumptions of application of the methods as they are applied to various designs. The engineering assumptions for the analytical and numerical analyses performed on the SYNKdrive configuration for rotary table applications are presented as follows.

A-2.2.1.1 The modeling of the SYNKdrive device included quasi-static modeling assumptions. The condition of quasi-static implies that mass effects and damping effects were not modeled.

A-2.2.1.2 The quasi-static assumptions of geometrical constraints were modeled. Therefore, the geometrical constraints were not defined as function of velocities or accelerations.

A-2.2.1.3 The structural dynamics considerations of the SYNKdrive nor of the surrounding mechanical systems were analyzed during the Phase-I study.

A-2.2.1.4 The assumption of nonlinear displacements were modeled for the contact point stiffness calculations. Consistent with the Hertz contact analysis approach, the contact patch develops to finite size for practical levels of applied loads.

A-2.2.1.5 The assumptions of finite rotations or the nonlinear kinematic behavior of the drive configuration was modeled to accurately generate the geometric boundary condition of the drive motion.

A-2.2.1.6 The effects of friction were not modeled in the analysis for the Phase-I analysis. However, the mathematical foundation of the drive was created to consider effects of friction in future investigations.

A-2.2.1.7 The effects of ball sliding, rolling and spinning were not studied. The mathematical foundation of the drive modeling will permit the detailed ball motion kinematics to modeled in future work.

2.2.2 Geometry Definition of SYNKdrive

The geometry definition was approximated consistently with the assumptions used for the detailed stress analysis. A list of the assumptions is presented as follows.

A-2.2.2.1 The geometry of the SYNKdrive configuration was defined by a parametric definition of geometry.

A-2.2.2.2 A complete and unambiguous list of the design parameters were recorded for the disks and ball assembly. Examples of the design parameters include:

- overall dimensions of primary, secondary and intermediate disks,
- detailed geometric definition of the race section profiles for the primary and secondary disks,
- complete and unambiguous mathematical description of the race profiles for the primary and secondary disks,
- conforming percentage defining conformity of ball to all races,
- race flank angles for each of the three disks,
- calculation of the ball contact terminology geometry-parameters used in the stress and deflection analysis,
- contact mechanics terminology for Hertz contact based analysis.

A-2.2.2.3 For a given set of design parameters, the device was represented as a practical and physical device. For example, practical tolerances were maintained for the axial gaps between the disks. The geometry information was post-processed to indicate the nearness of the solution features such as the contact patch edge to geometry features such the corners in the disk race profiles.

A-2.2.2.4 A mathematical description of the cam profile geometries were used within the confines of the drive, driven and reaction disks. The original closed form mathematical expressions were expanded throughout the kinematic and load path calculations to relate directly to the underlying mathematical basis of how the SYNKdrive produces the speed conversion.

A-2.2.2.5 The dimensions of the reaction disk was represented with realistic dimensions. This disk was modeled for suitably carrying the internal loads.

A-2.2.2.6 The prescribed design parameters were displayed in a graphics package to verify the orientation and definition of the drive configuration. This provided the visual verification of the appropriateness of selected design parameters of the SYNKdrive model.

A-2.2.2.7 The conformity modeling of the balls and races were defined as a superposition on flat race descriptions. Therefore any nonlinear effects of the development of the contact surface on an inclining conforming race were not modeled during the Phase-I study. However, the mathematical foundation was established for considering this nonlinear physical modeling in future work.

2.2.3 Analysis Model Definition

Nonlinear mathematical models were constructed based on the design parameters, the geometry and the drive assembly conditions. The analysis model was defined in terms of a prescribed SYNKdrive configuration based on a complex definition of the following quantities:

- design parameters of the disks,
- ball race section profiles,
- ball race profiles,
- activity status of the balls used in the analysis,
- kinematical constraints of the drive configuration,
- applied torque to the drive,

- Hertz based solutions for the compliance of the disk and balls,
- three-dimensional load transfer due to the kinematic constraints,
- a mathematical nonlinear finite element calculation for the three-point contact of each ball.

The features of the analysis model definition are outlined in the following.

A-2.2.3.1 The internal loads are calculated based on quasi-static force-deflection analysis. The analysis is completed at each position during the ball excursions to provide the internal loads experienced at the ball contact locations within the SYNKdrive.

A-2.2.3.2 A nonlinear set of equilibrium equations for analysis were formed for each ball. The force-system solved a dominant three-point loading condition for each ball in the drive. These equations provided the following results set at each contact point throughout the drive:

- normal deflections at the contact points,
- total contact force vectors,
- maximum compressive stress,
- semi-major and semi-minor axis of the elliptical contact patch,
- contact patch area,
- distance of the contact patch to geometric features in the drive,
- cross-over angles between the cam profiles on the primary and secondary disks,
- ball center locations based on the drive kinematics and ball deformations,
- ball speed and load factor for load-life estimations,
- numerical accuracy of the deformations and forces,
- overall linear and moment loads on the drive disks,
- average torsional drive stiffness for a given kinematic orientation,
- average load-life factors in support of the life estimates,
- load-life estimates based on the drive load transfer and speeds.

A-2.2.3.3 The localized compliance effects of the primary disk, secondary disk, intermediate disk, and balls were modeled. This was based on well-known and classical assumptions and the existing solutions (text book and literature) of contact mechanics. The effective model generated represented the SYNKdrive as a system of interacting nonlinear springs. The geometric effects of the races and ball material parameters and geometric configurations were modeled [Ref.-1].

A-2.2.3.4 The system of equations representing the system of spring models were solved given a prescribed applied torque to the SYNKdrive. The solution produced includes the force state on each individual ball [Ref. 1,2].

A-2.2.3.5 Analysis of a given single stage SYNKdrive design was completed in terms of the design parameters of the disks and ball races. The geometric definition of the drive was used directly to parameterize the force and deflection analysis of the drive.

A-2.2.3.6 A first-order load-life estimate was completed by consistently modifying empirical radial bearing load-life histories. An appropriate internal ball speed and effective ball loading for each contact point was applied to the SYNKdrive analysis to provide an estimated load-life curves for the overall drive.

A-2.2.3.7 The three-dimensional nature of the load transfer was calculated due to the kinematic constraints on the internal of the drive and defined in terms of the design parameters.

A-2.2.3.8 Design curves were automatically generated in a form that coalesces the results of several mathematical analyses on a prescribed SYNKdrive configuration.

2.2.4 Ball Loading Analysis

The ball contact modeling was completed with a Hertz contact analysis as described by Harris [Ref. 1]. The models were used to show the Hertz contact stresses and deflections for the complexity of the internal workings of the SYNKdrive. The Hertz solutions were used as the basis for calculating the contact point compliance. The compliance analysis involved analytical methods and, in addition, a force-deflection analysis built upon the analytical methods of contact mechanics, the geometric definitions of the drive components, and the strength of materials approach.

2.2.5 Distribution of Force Analysis

The distribution of forces throughout the SYNKdrive model was comprised of all of the ball interacting with the three disks. Due to the approximation of modeling the ball contact as a dominate three-point contact problem for each ball, this therefore leads to a system of three balancing force vectors for each ball in the drive. The reacting forces on each ball are balanced by the associated contact point on the primary, intermediate and secondary disks.

The instantaneous combination of forces on the given ball varies during the balls' excursion through the drive. The details of which flanks of the primary and secondary disk races are used during the ball radial descent and assent were modeled. All internal load paths for contact combinations were considered based on all poseable mathematical solutions. From these poseable solutions, a combination of load paths were chosen based on the feasibility condition for each ball in the drive.

An analytical nonlinear finite element truss-element formulation was considered to evaluate the three-point contact solution for all possible contact combinations. A nonlinear geometric boundary condition was applied analytically from kinematic solution to improve modeling accuracy.

A state-of-the-art nonlinear solver for limiting the numerical complexity in the load path calculations was exercised to perform the extensive calculations to thus generate the design curves.

2.2.6 Axial Load Calculation for Pre-load

An internal axial load was generated due to the three-dimensional aspect of the geometric constraints and the load transfer due to the compliance of the balls and disk races. The force vectors on each ball contribute to the circumferential direction which generates the torque balance in the SYNKdrive device. A small component of the ball force vectors are aligned in the axial direction of the drive to generate a magnitude proportional to the applied torque to the drive.

The magnitude of the internally generated axial load is used as the estimate for the amount of load needed to be reacted from the exterior viewpoint of the drive. This axial load magnitude is a minimum bounded value for the application of external forces for pre-loading calibration of the drive. The pre-loading feature of the SYNKdrive device is exploited to produce the zero-backlash characteristic. The internally generated axial load in the minimum amount of externally applied load required to provide zero-backlash.

The analytical and numerical calculation was completed to provide the detailed internally generated axial load to be used in estimates of circumferential and radial placement of the axial pre-loading device.

2.2.7 Rotary Compliance

The rotary compliance of the SYNKdrive configuration was generated from a strength of materials approach from the overall drive performance. The detailed ball level stress analysis provided the localized stiffness of the balls interacting on the conforming cam race profiles. The overall stiffness of the drive disks were considered extremely stiff when compared to the localized ball stiffness. The detailed stress analysis was computed for a given kinematic configuration of the drive. A small variation in the nonlinear-kinematically-admissible geometric-boundary-condition was applied to a given kinematic orientation. The resulting axial twist for a computed twisting torque was used to compute the tangent stiffness constant of the SYNKdrive. This first-order compliance for the SYNKdrive was completed for several designs and design curves were generated to aid in the sizing of the drive for particular stiffness requirements and applications especially useful for rotary drive situations.

2.2.8 Discussion of Results

An extensive set of analytical and numerical calculations were completed for a flat race, 51%, 52% and 53% conformities for the primary and secondary disks race profiles. For each conformity, a range of flank angles: 25, 30, 35, 40, 45, 50, and 55 degrees were considered on the primary and secondary disks. This study lead to twenty-eight (28) detailed reports of results [see Appendix-B]. A detailed summary of the calculations are provided in a supplementary document. The essence of the detailed studies are presented in a series of design curves in this section. A brief discussion of the results for the drive are presented as follows.

Figure 2.2.8-1: Separating Force versus Flank Angle.

The separating force calculation was produced directly from the force equilibrium on the drive disks. Due to the fact that a prescribed output torque of 1200 in-lbf was required of all of the SYNKdrive analyses, the internally generated separating force for all of the models was observed to be identical. The nonlinear variation of the separating force based on the flank angle was observed and would be used to design the amount of pre-load required to sustain the zero-backlash feature of the drive.

Figure 2.2.8-2: Torsional Stiffness versus Flank Angle.

To satisfy the torsional stiffness requirements of rotary positioning devices, the smaller percentages of race conformity is desirable. The case of flat races was found to be the most compliant model and should be avoided.

Figure 2.2.8-3: Primary Disk (Dp): Maximum Contact Stress.

Figure 2.2.8-4: Intermediate Disk (Di): Maximum Contact Stress.

Figure 2.2.8-5: Secondary Disk (Ds): Maximum Contact Stress.

The maximum contact stress plots illustrated the importance of designing with race conformity. With the use of conformities of 51%, 52%, and 53% the maximum compressive stress at all contact points are less than 300 ksi in magnitude. The case of flat races exceeded 500 ksi and thus would not provide a durable design for the expected load transfer levels indicated by 1200 in-lbf torque output.

Figure 2.2.8-6: Primary Disk (Dp): Distance of Contact Edge to Corner.

Figure 2.2.8-7: Intermediate Disk (Di): Distance of Contact Edge to Corner.

Figure 2.2.8-8: Secondary Disk (Ds): Distance of Contact Edge to Corner.

The correlation of the geometry description with the stress and deflection analysis of a given ball and ball race profile defines an essential design requirement. The interactions of the developed contact patch size with geometry features noted as ***corners*** in the race profiles define a limit to amount of contact development allowed for a specific design. The critical locations are the corners on the primary and secondary disks. It is noted that for the maximum stiffness case, with the requirement of non-overlapping contact patches with corners, the case of 40% flank angle at the 52% conformity level will suffice. The case of 40% flank angle with 51% is marginal on the primary disk, but is over-lapping on the secondary disk. The case of 35% flank angle with 53% could be acceptable on the primary disk, but is marginally acceptable on the secondary disk.

A detailed numerical listing of the drive with 40% flank angles and 52% conformity is provided in Appendix-A. A summary of the performance values and extreme values observed for this SYNKdrive configuration are presented in Tables 1 and 2.

Figure 2.2.8-9: Load-Life Calculation for 52% Conformity and 40% Flank Angle.

A load-life calculation for the recommended design is presented in Figure 2.2.8-9. The specific numerical values are presented in Table-3. The load-life calculations for the low speeds experienced by rotary positioning applications provide an extremely long lifetime estimates. The higher drive speed magnitudes predict device lifetimes that represent suitable application to higher speed positioning for the same torque throughput levels.

Table 2.2.8-1: Summary of Performance for 40 degree Flank Angle and 52% Conformity.

Applied Angular Twist (degrees) :	0.395194
Torque Output:	1200.00
Separating Force:	531.859
Torsional Stiffness (in-lb/rad) :	173977.0

Table 2.2.8-2: Summary of Extreme Values for 40 degree Flank Angle and 52% Conformity.

Maximum Compressive Contact Stress Dp :	-230767.00	(ball 35)
Maximum Compressive Contact Stress Di :	-201066.00	(ball 36)
Maximum Compressive Contact Stress Ds :	-238991.00	(ball 36)
Maximum Ball Deflection Dp:	0.000125153	(ball 35)
Maximum Ball Deflection Di:	0.0000950156	(ball 36)
Maximum Ball Deflection Ds:	0.000141442	(ball 35)

Maximum Ball Contact Area Dp:	0.00014225	(ball 35)
Maximum Ball Contact Area Di:	0.000107999	(ball 36
Maximum Ball Contact Area Ds:	0.000160781	(ball 35)
Maximum Ball a-patch Dp:	0.0192974 (ball 35)	
Maximum Ball a-patch Di:	0.0168142 (ball 36)	
Maximum Ball a-patch Ds:	0.0205148 (ball 35)	
Maximum Ball b-patch Dp:	0.0023464 (ball 35)	
Maximum Ball b-patch Di:	0.00204453	(ball 36)
Maximum Ball b-patch Ds:	0.0024947 (ball 35)	
Minimum Distance to Dp Corner :	0.00546327	(ball 35)
Minimum Distance to Di Corner: Dp:	0.00943585	(ball 36)
Minimum Distance to Di Corner: Ds:	0.00943585	(ball 36)
Minimum Distance to Ds Corner:	0.00424591	(ball 35)

Table 2.2.8-3: Summary of Load-Life Estimates.

Dp speed	Ds speed	Dp LLF	Di LLF	Ds LLF	Life (hours)	Life (years)
10	0.2703	10.5036	15.3547	8.6056	34028100	3884.49
40	1.0811	7.4264	10.8563	6.0844	8503470	970.715
400	10.8108	4.1754	6.1038	3.4209	849698	96.9975
800	21.6216	3.5121	5.1342	2.8775	425370	48.5582
1200	32.4324	3.1736	4.6393	2.6001	283591	32.3734
1600	43.2432	2.9536	4.3177	2.4199	212752	24.2868
2000	54.0541	2.7932	4.0832	2.2885	170172	19.426
2400	64.8649	2.6688	3.9013	2.1865	141815	16.189
2800	75.6757	2.5680	3.7541	2.1040	121586	13.8797
3000	81.0811	2.5238	3.6894	2.0678	113422	12.9477
4000	108.1080	2.3488	3.4336	1.9244	85087	9.71315

2.2.9 Preliminary Design Methodologies

The ***preliminary*** nature of design methodologies are believed to be the application analytical methods to ***front load*** the design process of implementing SYNKdrive devices. The basis of this approach could include the search for appropriate magnitudes of geometry and design parameters used in specifying the drive. The search for appropriate SYNKdrive sizing could begin with design curves that have been provided in this final report. The overall behavior of the SYNKdrive was seen to be modeled with consistent behavior throughout the variations of the design variables. This mechanical feature is especially useful and essential for robust and critically design applications using the SYNKdrive.

From the presented study, it is believed that the design of future applications of the SYNKdrive can realistically begin with investigations through the design curves for parameter selection for the device sizing. This level of analytical and technical understanding of the SYNKdrive was brought to a significant level during Phase-I. However, additional analytical modeling is recommended for issues regarding the detailed ball motion and dynamics to better understand the ball performance. Each new analytical modeling produced could directly lead to new design curves as presented in this first study.

2.2.10 Recommendations for Rotary Table Design

The essential recommendations regarding rotary table design are dominated by the stiffness requirements, zero back-lash, and contact stress requirements of the particular application.

For the design of the SYNKdrive device for high stiffness requirements, the designer has control by exploiting the number of balls in the drive, and for a given number of balls, the flank angle of the ball race profiles can be adjusted to generate the tangent stiffness required.

In combination with the number of balls and the flank angles, the maximum contact stress must be limited to provide the durability of the device for the particular speed application. In concert with the balls and flank angles, the maximum contact stress magnitude can be tuned by adjusting the amount of conformity between the balls the ball races.

These above mentioned design parameters act a starting point to provide the general features of the external performance of the drive. The secondary aspects of the drive design involve the size of the contact patch semi-major axis at each contact point on each ball. The geometric limitations of the conforming ball and ball-race are significantly less than what is usually seen in conventional radial bearings. A limit exits to what extent that conformity can be used to control the maximum contact stress without creating over-hanging contact patches with the internal geometry of the SYNKdrive. This is not a limiting feature to the drive design and performance, but is a design requirement that must be satisfied.

2.2.11 References

1. Tedric A. Harris, "Rolling Bearing Analysis", John Wiley & Sons, Inc., Third Edition, 1991.

2. Klaus-Jurgen Bathe, "Finite Element Procedures in Engineering Analysis", Prentice-Hall, Inc., 1982.

2.3 Fabrication Effort

A standard SYNKdrive uses a single drive cam and a single driven cam. A gap is set between the cam plates so that the ball rides on two flanks and the reaction disk. When reversing direction, the ball becomes free in the preset gap until the opposite flank is contacted. This gap is related to backlash.

ZBcams are fabricated such that the gap may be set to zero without squeezing the ball and forming five points of contact. If one flank is removed from the cam path, the ball can only ride on one flank. To hold this ball position, the drive and driven flanks must be fabricated in the same way, wedging the ball into the reaction disk. This is fine for a single direction of rotation, but if the drive reverses, there is no flank left on which to drive. Therefore, a second concentric cam must be cut opposite to that of the original cam which will then allow for reverse direction rotations.

To remove the proper flank in a controlled fashion a program was written which allows for tool widths and ball contact points as it moves around the cam path removing material. The tool moves around the cam lobe tips within the original cam path. When the tool reaches a point where the flank is no longer needed, the tool is offset a prescribed amount into the flank to be relieved. The tool then follows this offset path until another lobe tip has been encountered and moves back to the original cam path until, again, the tool is offset into the offending flank. This way the cams may be modified without affecting the system performance. See figure 2.3-1 for a section view of the SBIR phase I prototype and Appendix A for the drawing details.

2.4 Testing the ZBcam

To fully appreciate the quality of SynkDrive™ technology as it pertains to high precision positioning systems, extensive testing is required. Phase I testing, as stated in the proposal, includes: (1) Position repeatability; (2) Torsional stiffness; (3) Backlash; (4) Separation forces.

The testing requirements stated above require equipment particular to this type of device. Needed are position measuring instruments on the input and the output of the mechanism, torque and load sensors and a device to apply loads.

Required Equipment:
2-Dial indicators @ 0.00005" grad, 0.015" range
2-Dial test indicators @ 0.0005" grad, 0. 030" range
4-Magnetic bases for indicators
1-2 lb Spring scale
1-50 lb Spring scale
50 lb sand
bucket
assorted curved spring washers

2.4.1 Test Procedures

2.4.1.1 Assembly and Handling

The mechanism will be assembled by the test engineer under the supervision of the principal investigator. The races will be cleaned and greased with Dow Corning BG 20 synthetic grease.

The drive will be mounted vertically in a frame designed to incorporate the test apparatus. See Figure 2.4.1.1-1 for the layout of the mechanism and test assembly.

Prior to mounting in the test stand, the drive will be cleaned and be kept free of dirt and dust. The mechanism will be covered when not in use.

NOTICE: Once the drive is handed over to the test engineer, the drive becomes the responsibility of the test engineer. **Nobody**, shall touch, move or operate the drive without expressed permission from the test engineer or principal investigator.

2.4.1.2 Data Acquisition and reporting

The measurement equipment will be calibrated and sequestered for the duration of this experiment. Measurements will be made from manual observations of the test equipment and the data entered in a log book with the date and time of observation.

The log will contain a description of the experiment, a sketch of the setup, clearly tabulated data and a detailed discussion of thoughts, ideas and/or observations witnessed. These discussions need not be related to the goals of the measurements. The data from the log will then be transferred to a computer spreadsheet in a format ready for reporting.

2.4.2 Test Equipment and Fixtures

2.4.2.1 Position Sensors

Angular position will be determined using linear dial gages and a bar attached to the rotational axis. The bar will be calibrated for deflection under load. Measurements will be limited to small linear travel for accurate angular relationships.

With all of the errors taken into account, the accuracy of the drive cam position is 10 arc seconds and the accuracy of the driven cam position is 10 arcseconds.

Linear measurements will be made with direct readings from the linear dial gages at a resolution of 0.00005 inches and an accuracy of ±0.000025.

2.4.2.2 Torque Sensor

Torque will be measured using linear spring scales positioned on a bar attached to the rotation axis. Measurements will be limited to small linear travel for accurate torque relationships.

Accuracy of the torque measurement is ±7.0 in-lb using the 50 pound scale and ±1.0 in-lb using the two pound scale.

2.4.2.3 Load Application

Loads will be applied using a container with sand hanging from a spring scale. Measurements will be limited to small linear travel for accurate load relationships.

Accuracy of the load measurement is ±0.5 pounds for the 50 pound scale and ±0.03 pounds for the two pound scale.

2.4.3 Measurements

2.4.3.1 Instrument Calibration

To increase the precision of the measurements made in this experiment, the instruments will be calibrated. The instruments to be calibrated are the load/position bars, spring scales and dial gages.

2.4.3.1.1 Spring scale

The scales are purchased with an accuracy of ±0.5 pounds for the 50 pound scale and ±0.03 pounds for the two pound scale. These numbers will be verified at the mid range of the scale with a single known weight of 25 pounds for the 50 pound scale and one pound for the two pound scale.

2.4.3.1.2 Dial gages

The dial indicators are purchased with an accuracy of ±0.000025 inches for the 0.015 inch scale and ±0.00025 inches for the 0.030 scale. These numbers will be verified at the mid range of the dial indicator with a single known gage block of 0.007 inches for the 0.015 inch scale and 0.015 inches for the 0.030 inch scale.

2.4.3.1.3 Load/position bar

To use the load/position bars for measuring position, the deflection of each bar must be calibrated This quantity is used to remove measurement deflections from the actual system deflection by subtracting the calibration data from the recorded data.

Using the calibrated spring scale and dial gage, loads will be applied to the bar which has been clamped in a very stiff vise. A plot of deflection versus load would then be determined.

Each beam will be clamped in a vise and a dial gage located at the end of the bar will be zeroed. The beam will then be loaded in five pound increments to 50 pounds total. At each load a dial gage reading will be recorded. The data will then be fit to a polynomial such that the fit error is no more than 0.1% at any point.

2.4.3.2 Backlash

Backlash will be measured by two means. The first method looks at the play in the driven disk with the drive disk held stationary and the second method involves moving the drive disk back and forth to known locations.

2.4.3.2.1 Driven disk play

The drive disk will be locked and a 2 pound spring scale attached to the driven disk. A dial indicator with 0.00005 inch graduations will be set up to indicate motion on the driven disk.

The spring scale will then be pulled to no more than eight ounces perpendicular to the bar axis and the change in the dial indicator readout will be recorded.

2.4.3.2.2 Return displacement measurement

Two hard stops will be placed in the rotational path of the drive disk bar to create two repeatable input positions. The driven disk will be placed at an arbitrary starting location and a dial indicator with 0.00005 inch graduations will be set up to indicate the starting position of the driven disk.

The drive disk will be rotated form the start position to the second position and then back to the start position. The driven disk dial indicator readout will be recorded.

2.4.3.3 Position Repeatability

Two hard stops will be placed in the rotational path of the drive disk bar to create two repeatable input positions. The driven disk will be placed at an arbitrary starting location and two dial indicators with 0.00005 inch graduations will be set up to indicate the starting position and stopping position of the driven disk.

The drive disk will be rotated form the start position to the second position and the driven disk stop position dial indicator readout will be recorded. The drive disk will then be rotated form the second position back to the start position and the driven disk start position dial indicator readout will be recorded.

The test will be repeated for eight random and arbitrary driven disk start and stop positions.

2.4.3.4 Torsional Stiffness

The drive disk will be locked and a 50 pound spring scale with a bucket will be attached to the driven disk. A dial indicator with 0.00005 inch graduations will be set up to indicate the position of the driven disk.

Sand will be added to the bucket in five pound increments up to 45 pounds and the dial indicator readout will be recorded. The readings will be corrected as per the bar calibration.

2.4.3.5 Cam Separation Force

The curved spring washer will be set to a predetermined preload of the cam system. The drive disk will be locked and a 50 pound spring scale with a bucket will be attached to the driven disk. Two dial indicators with 0.00005 inch graduations will be set up to indicate the axial deflection of the drive and driven disks. Sand will be added to the bucket in five pound increments up to 45 pounds and the dial indicator readouts will be recorded.

2.4.4 Test Results

The testing was conducted in the SYNKINETICS' laboratory as per described above. The results have been detailed below.

2.4.4.2 Backlash

As was detailed above, backlash will be measured by two means. The first method looks at the play in the driven disk with the drive disk held stationary and the second method involves moving the drive disk back and forth to known locations.

2.4.4.2.1 Driven disk play

The spring scale will be pulled to no more than eight ounces perpendicular to the bar axis and the change in the dial indicator readout will be recorded. Ten trials with a clockwise pull and ten trials with a counter-clockwise pull were registered. The results show about 15 ± 10 arcseconds of backlash in the clockwise pull direction with a 3 ± 1 arcsecond of hysteresis. The results also show about 10 ± 3.5 arcseconds of backlash in the counter-clockwise pull direction with a 2 ± 1 arcsecond of hysteresis.

2.4.4.2.2 Return displacement measurement

The drive disk will be rotated form the start position to the second position and then back to the start position. The driven disk dial indicator readout will be recorded. The backlash for three random positions of the output cam are 13 ± 4 arcseconds, 6 ± 2.5 arcseconds and 7 ± 2.5 arcseconds. Hysteresis is included in these numbers as this experiment cannot differentiate between the errors.

2.4.4.3 Position Repeatability

The drive disk will be rotated form the start position to the second position and the driven disk stop position dial indicator readout will be recorded. The drive disk will then be rotated form the second position back to the start position and the driven disk start position dial indicator readout will be recorded. The results show a good average return to position ranging from -3 arcseconds to +2 arcseconds of repeatability. The standard deviation shows a wild scattering of points ranging from 5.5 arcseconds to 13 arcseconds.

2.4.4.4 Torsional Stiffness

Sand will be added to a bucket in five pound increments up to 45 pounds and a dial indicator on the beam will readout and recorded. The readings will be corrected as per the bar calibration. Three trials at three different output cam positions were tested. The results show stiffness in the range of 65,000 ± 3000 in-lb/rad.

2.4.4.5 Cam Separation Force

In monitoring of the cam plate separation forces, two trials at two random output cam positions were developed. The belville washer preload was calculated to be 32 pounds. The forces of separation ranged linearly from 0 to 62 pounds as the input was varied from 0 to 450 in-lb of torque.

2.4.5 Results Discussion

The data show results at least as good as the measurements and stand up to the analytical results reviewed above. The results detailed above are collected in the following table.

Table 2.4.5: Test Results of ZBcam Prototype

Test	Measured Range	Predicted	Units
Backlash	3 to 15	-	arcseconds
Position Repeatability	±3	-	arcseconds
Torsional Stiffness	65,000	81,000	in-lb/rad
Separation Forces	62 @ 450 in-lb	500 @ 1200 in-lb	pounds

Backlash and repeatability are both within the reading error of ±10 arcseconds. The scatter of the repeatability data suggests a nonrepeatable hysteresis effect that needs more discovery. The torsional stiffness measurement accuracy is ±10% and the result was within 20% of predicted. The predicted stiffness assumes no hysteresis and no friction which will affect the overall system stiffness. The biggest difference is in the predicted separation forces versus the measurement. Predictions were calculated at 1200 in-lb output torques while the measurements were limited to 450 in-lb at the output cam. The preload washer was not calibrated to a reliable extent. Therefore, at this time no conclusion can be drawn as to the difference between the predicted and measured separation forces.

3.0 Conclusions

This study of SYNKdrive, ZBcam technology and how it relates to manufacturing and robotics was most beneficial for not only developing and proving the know-how but also justifying the commercial viability of ZBcam products. This has been realized by the development of a corporate partner interested in improving and expanding their current product line. The following are the SYNKINETICS Inc. accomplishments achieved during the phase I program:

- Analyzed the ZBcam technology studying the effects of the various cam parameters,
- Applied the SYNKdrive technology, via parametric equations, to manufacturing processes,
- Developed a working prototype to demonstrate the ZBcam technology,
- Developing a corporate relationship with leaders in the field of precision rotary tables.

As a result of our phase I progress, SYNKINETICS Inc. is poised to further the development of ZBcam technology in manufacturing and robotics by way of product design and implementation, with the help of our corporate partners and the Small Business Innovative Research Program, phase II process.

The next phase of work will entail improved testing facility, higher precision machining and a comprehensive approach toward developing product. With these factors in mind, the future of the ZBcam technology may be explored and exploited such that enhancements in robotics and manufacturing are realized.

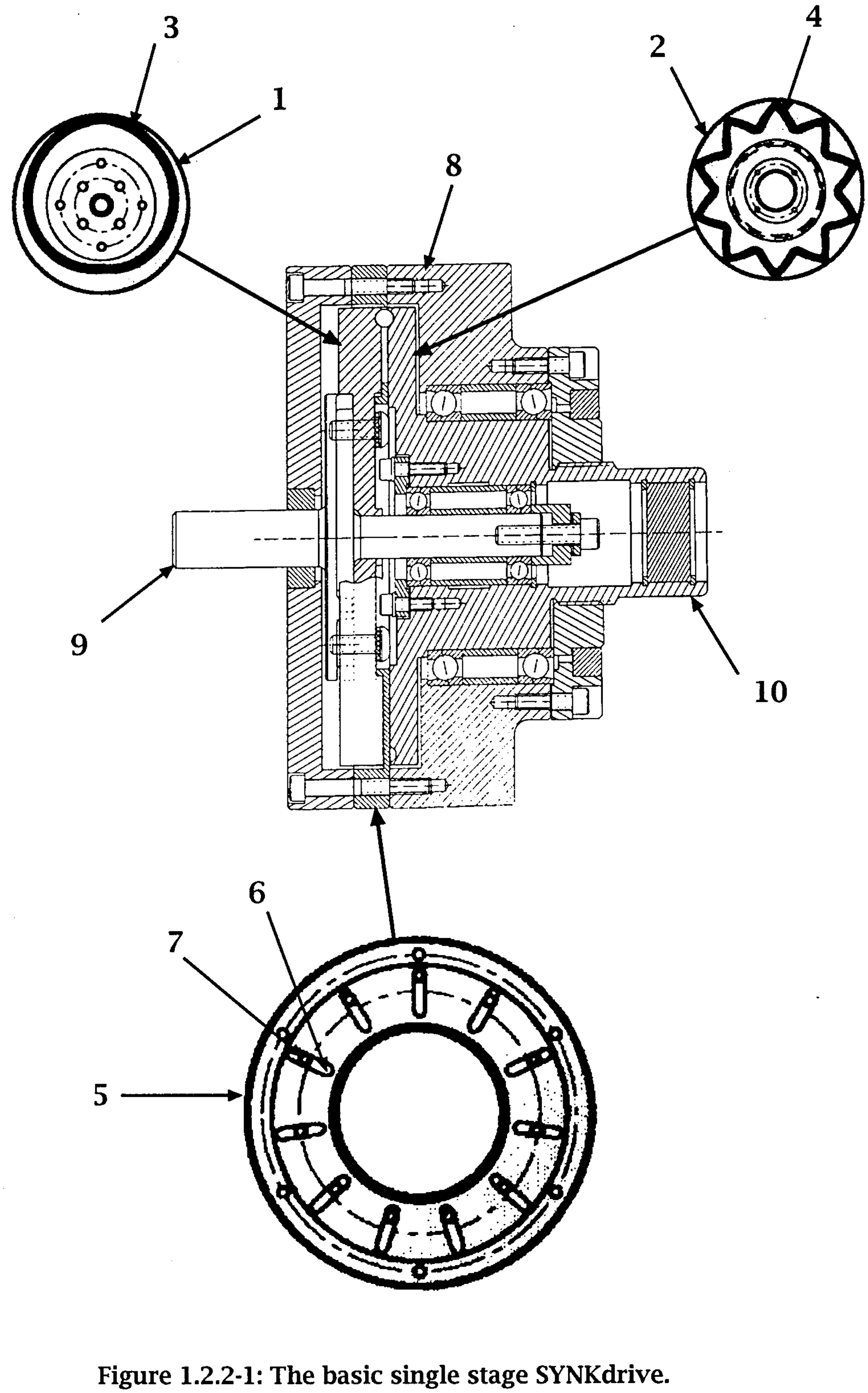


Figure 1.2.2-1: The basic single stage SYNKdrive.

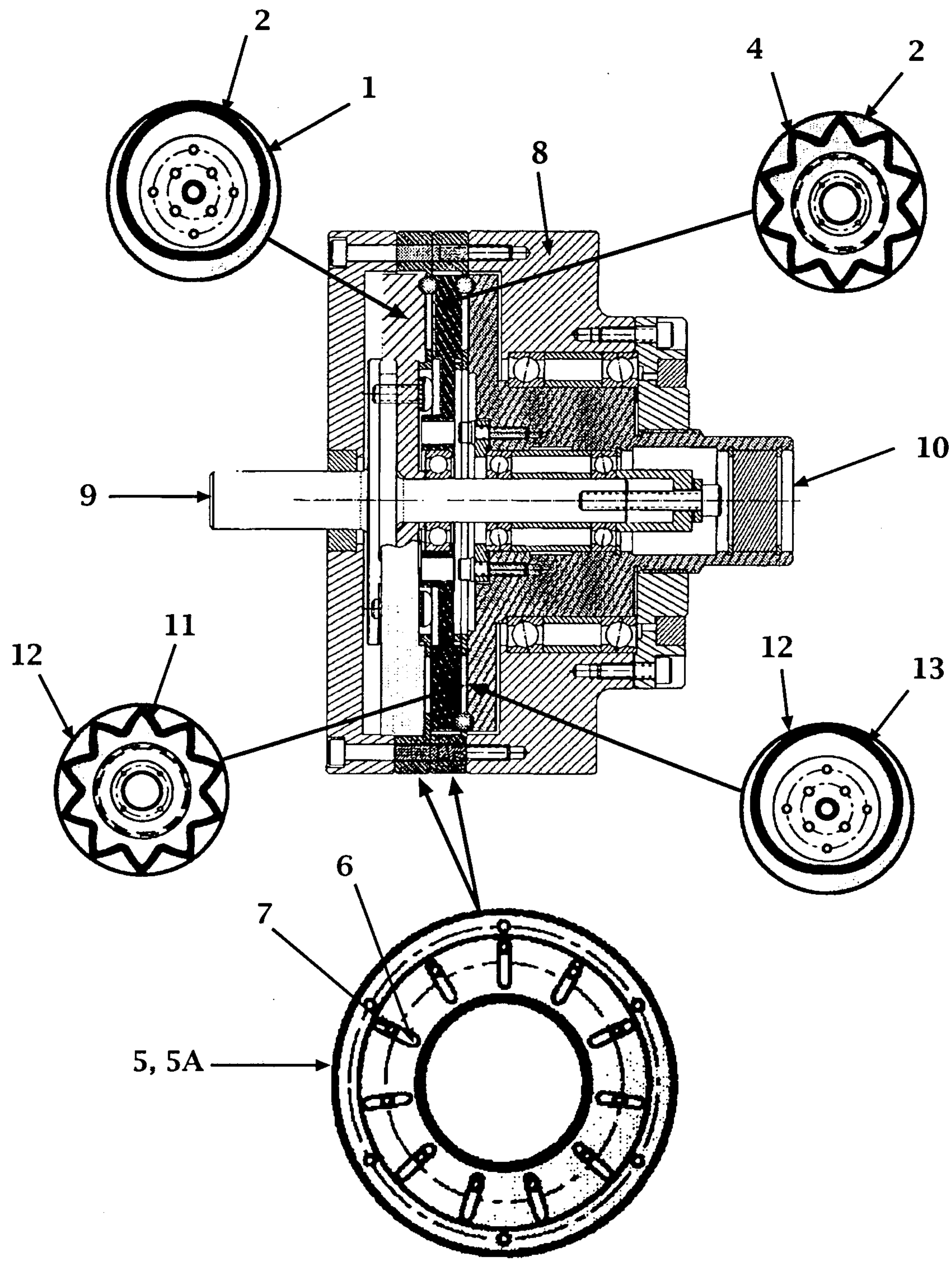


Figure 1.2.2-2: The basic multi-stage SYNKdrive.

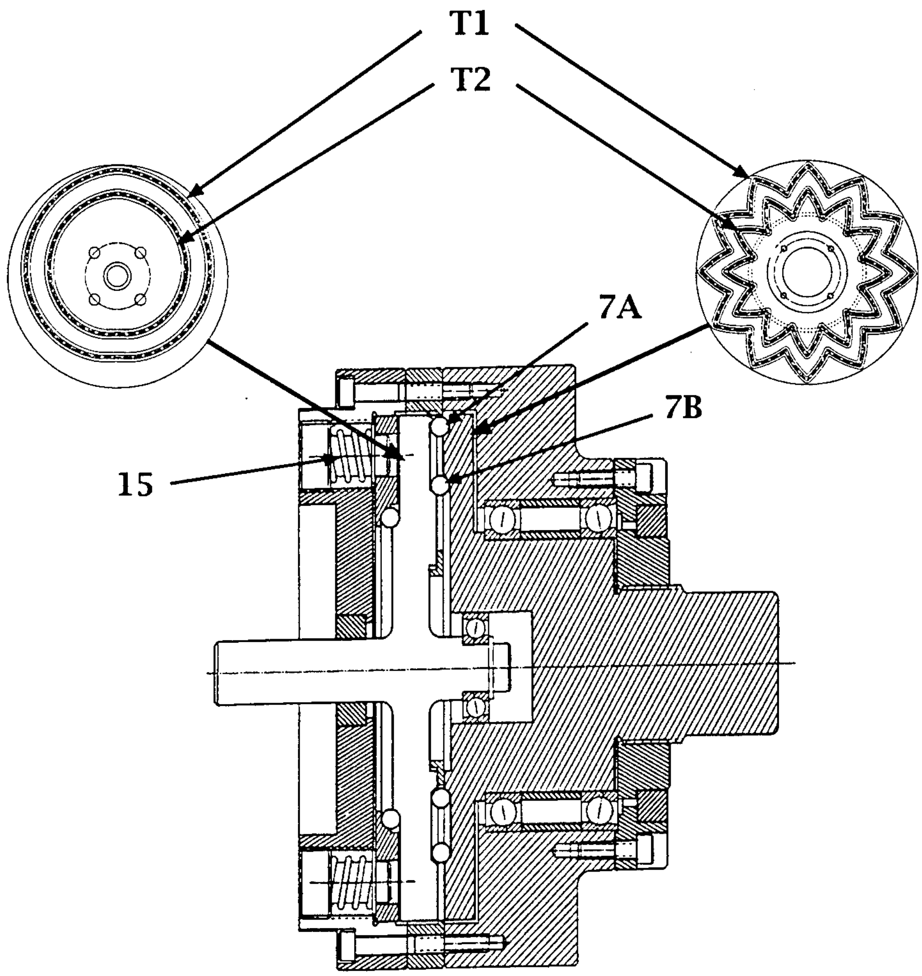


Figure 1.2.2-3: The zero backlash SYNKdrive.

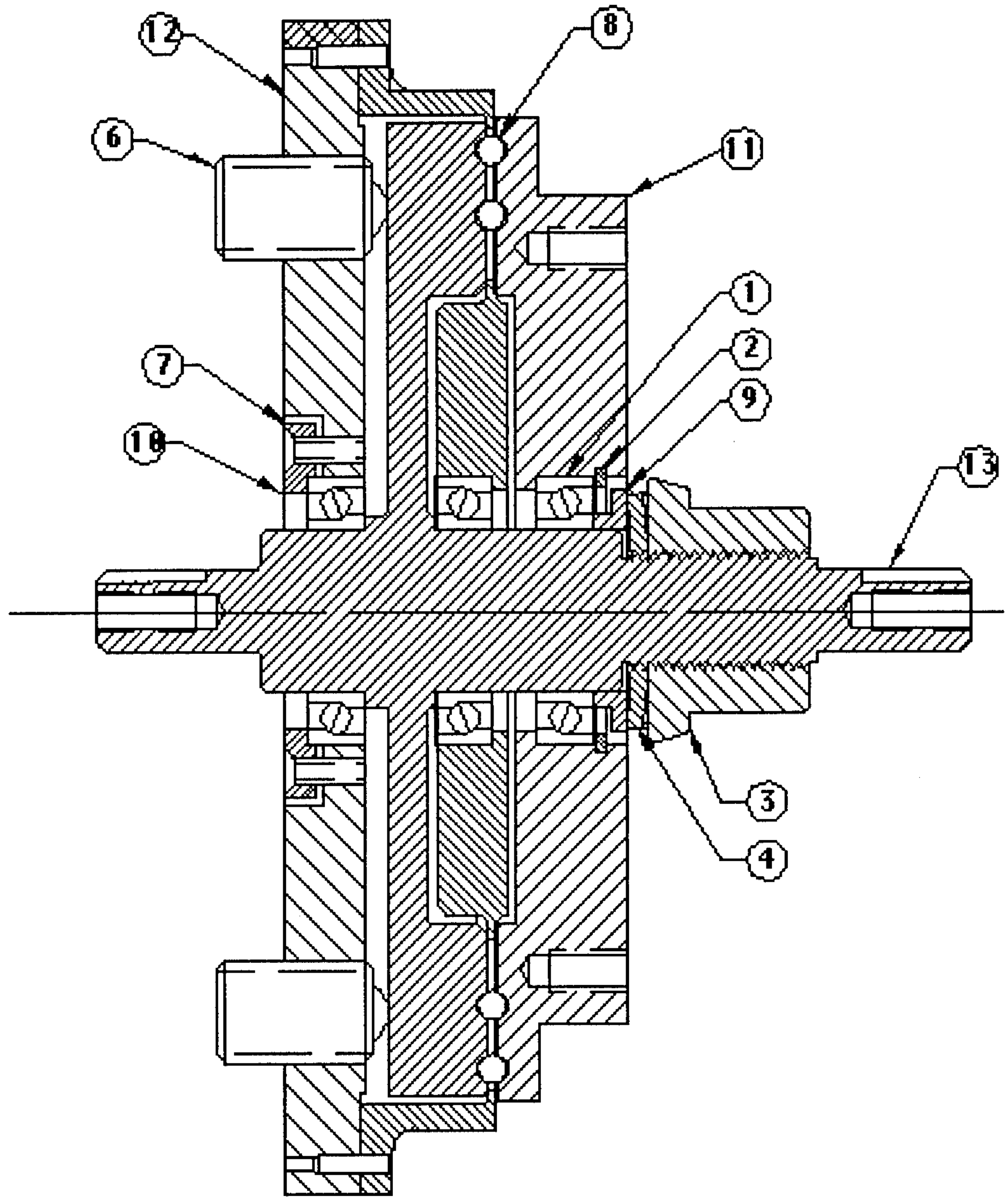


Figure 1.3-1: The zero backlash prototype for the phase I development.

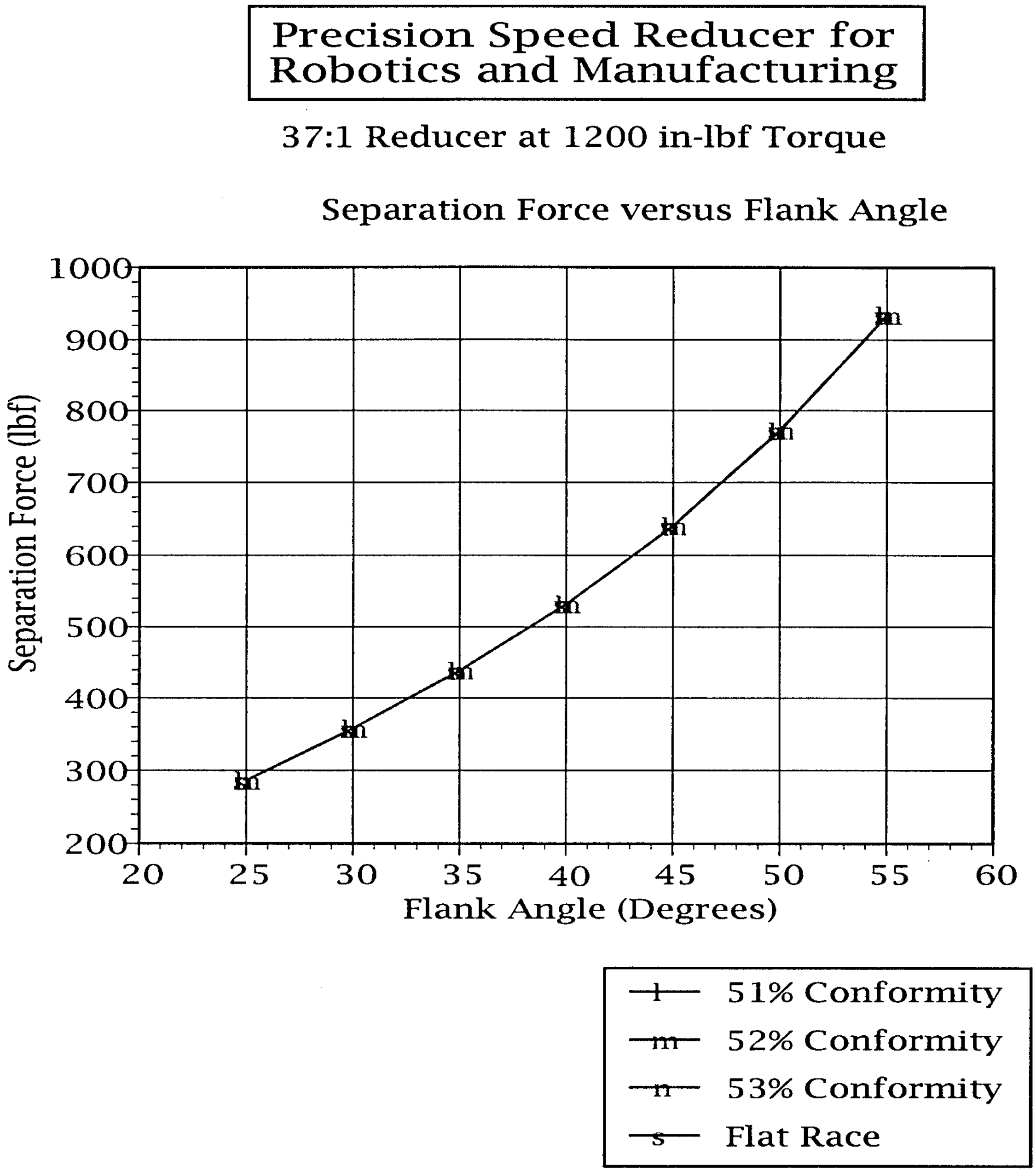


Figure 2.2.8-1: Separating force versus flank angle.

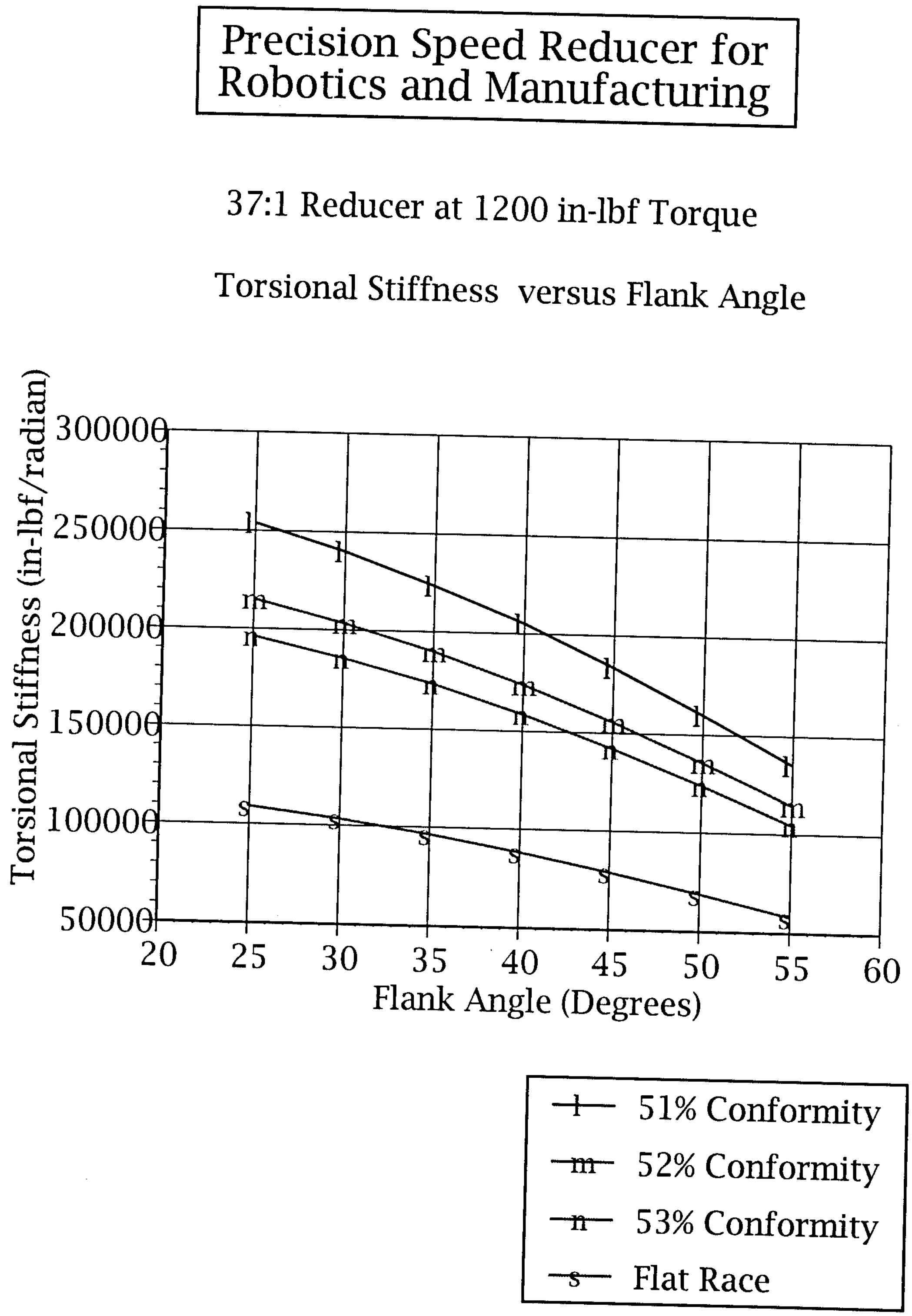


Figure 2.2.8-2: Torsional stiffness versus flank angle.

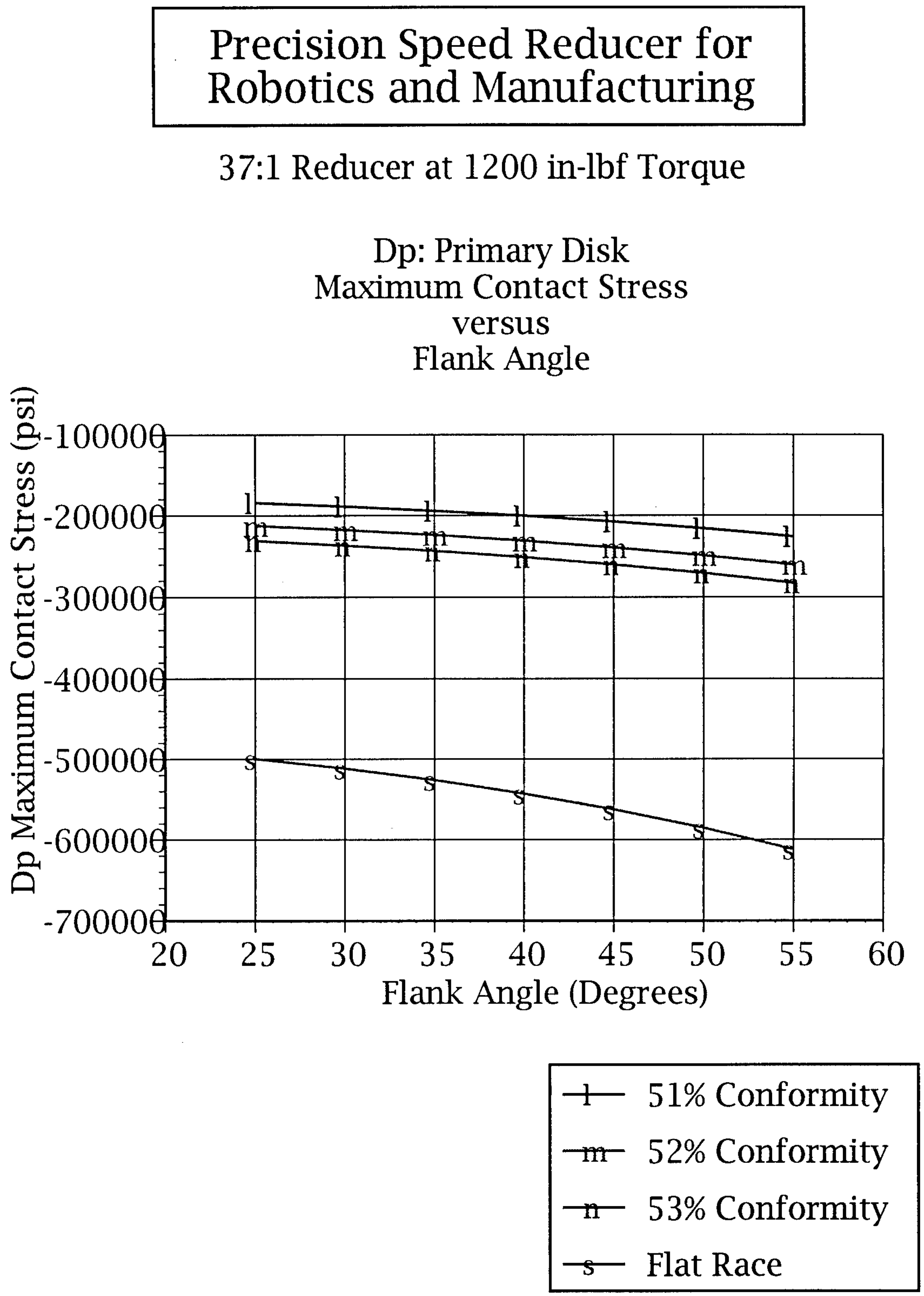


Figure 2.2.8-3: Primary disk maximum contact stress.

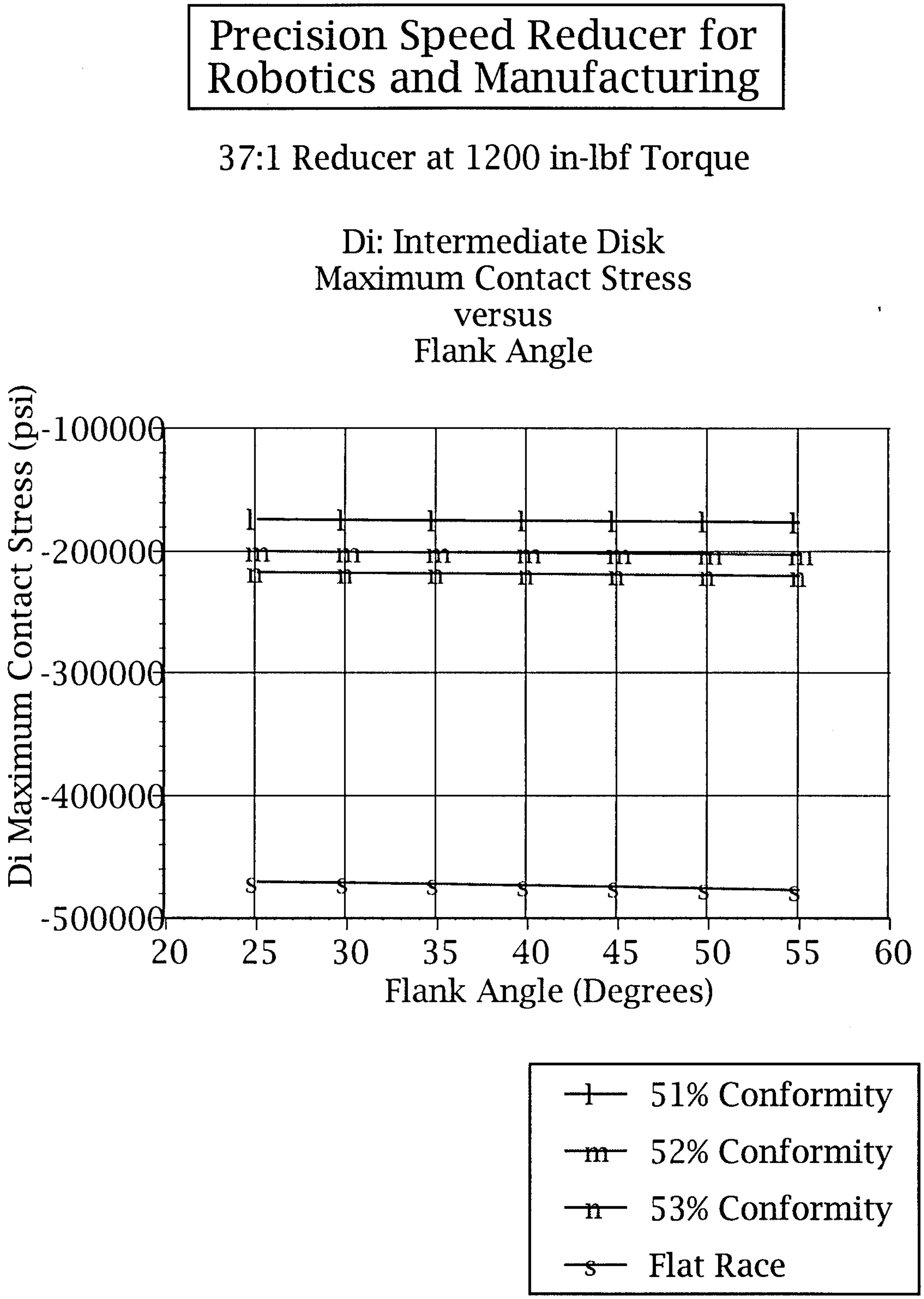


Figure 2.2.8-4: Intermediate disk maximum contact stress.

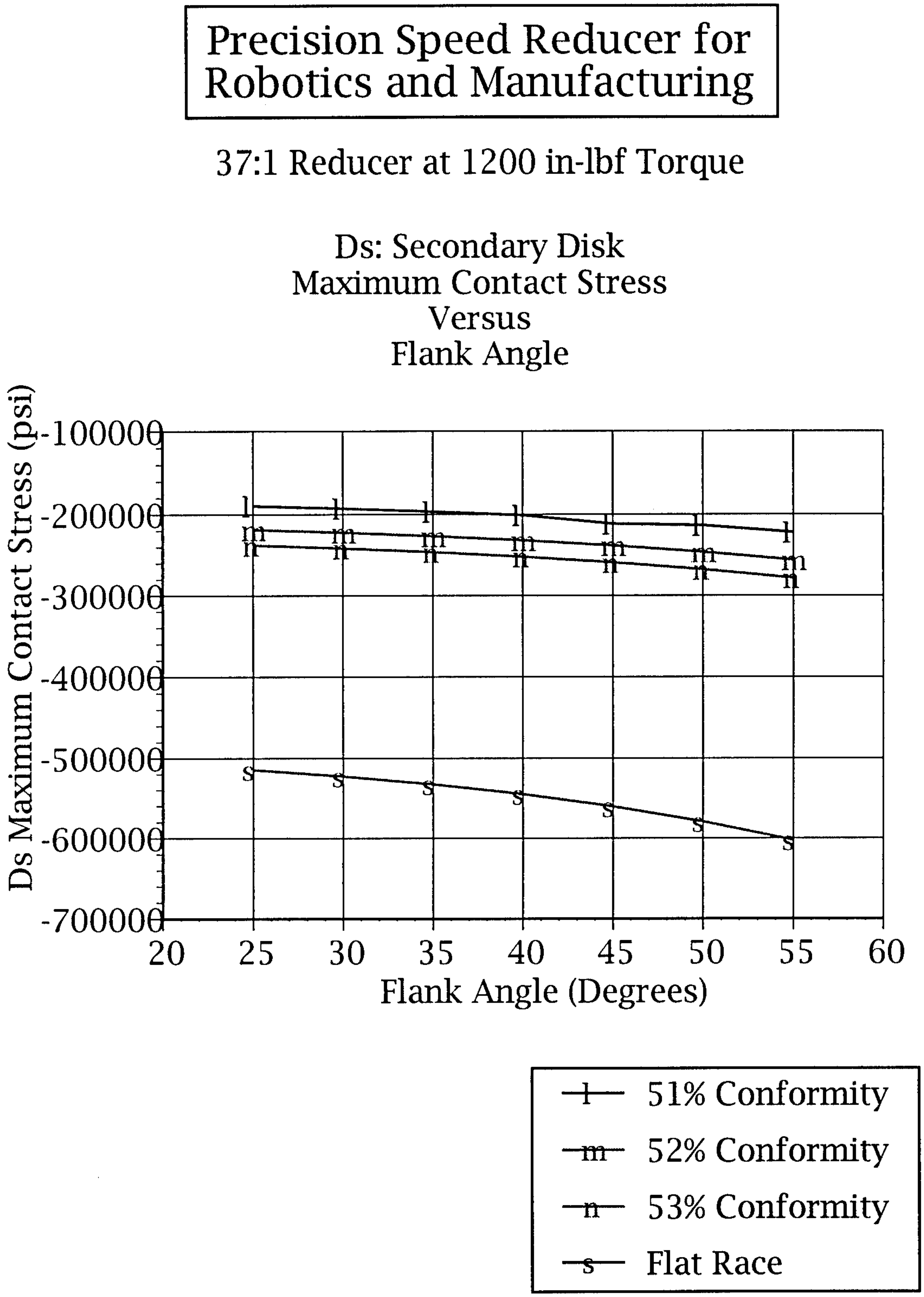


Figure 2.2.8-5: Secondary disk maximum contact stress.

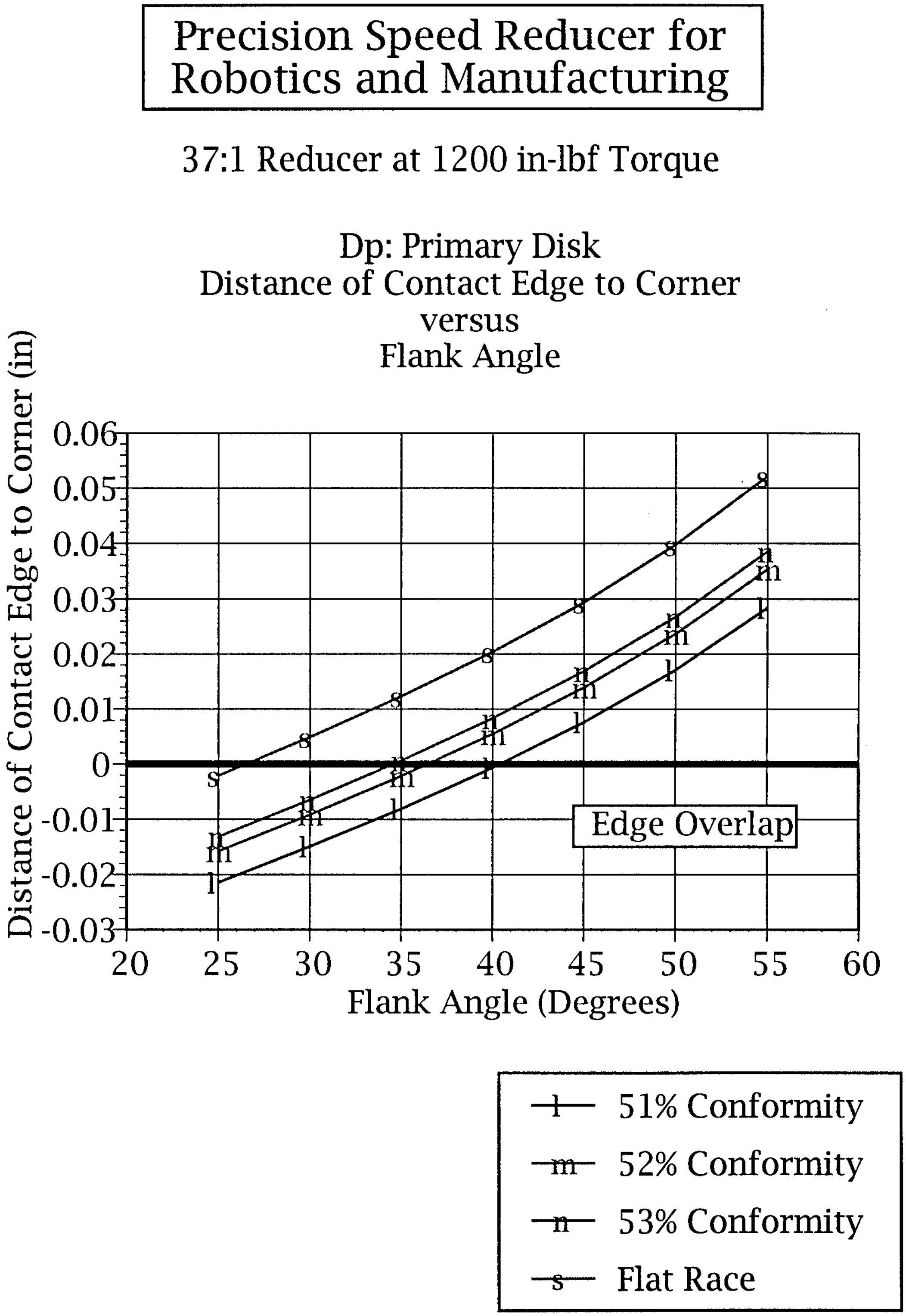


Figure 2.2.8-6: Primary disk distance of contact edge to corner.

Precision Speed Reducer for Robotics and Manufacturing

37:1 Reducer at 1200 in-lbf Torque

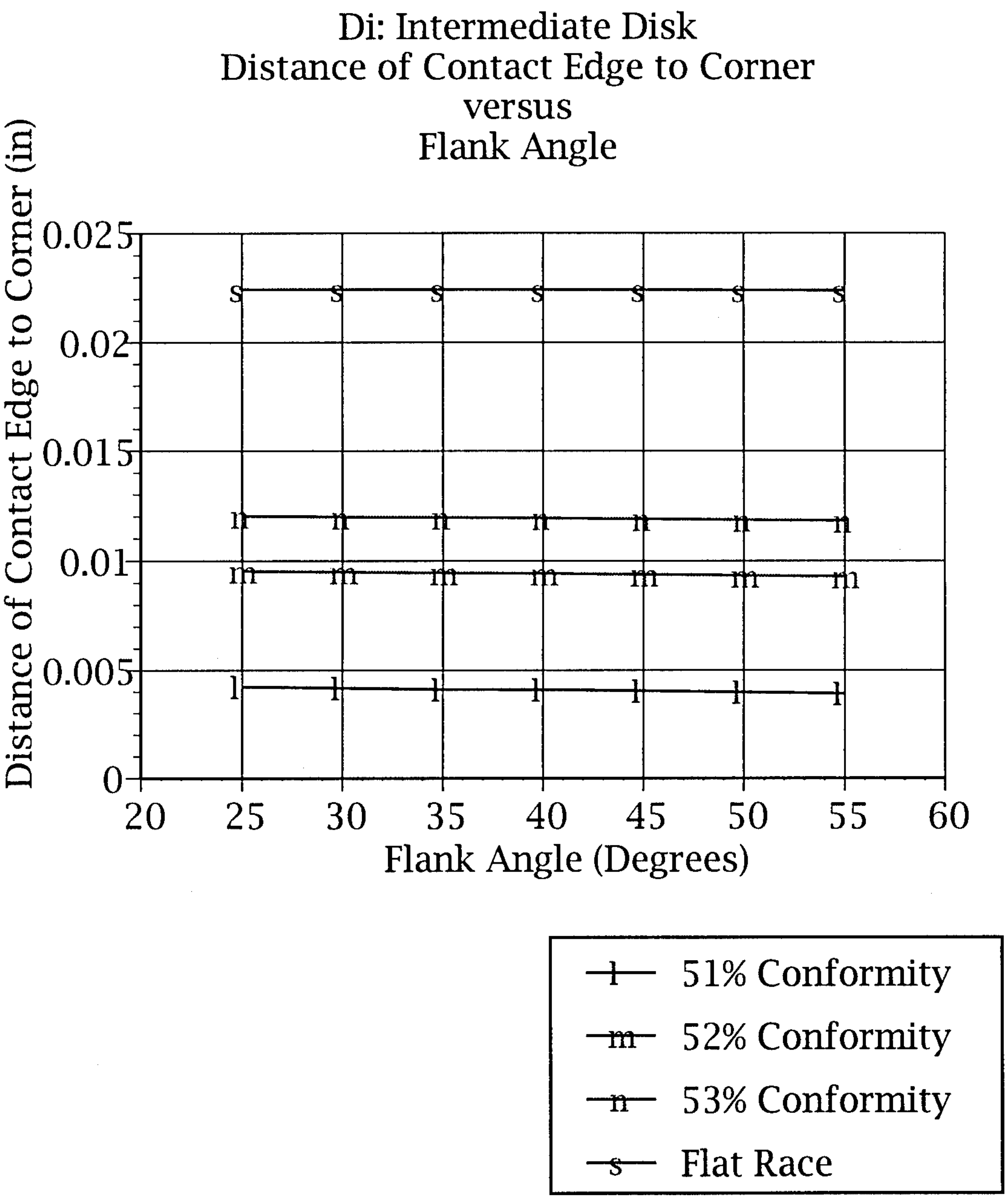


Figure 2.2.8-7: Intermediate disk distance of contact edge to corner.

Precision Speed Reducer for Robotics and Manufacturing

37:1 Reducer at 1200 in-lbf Torque

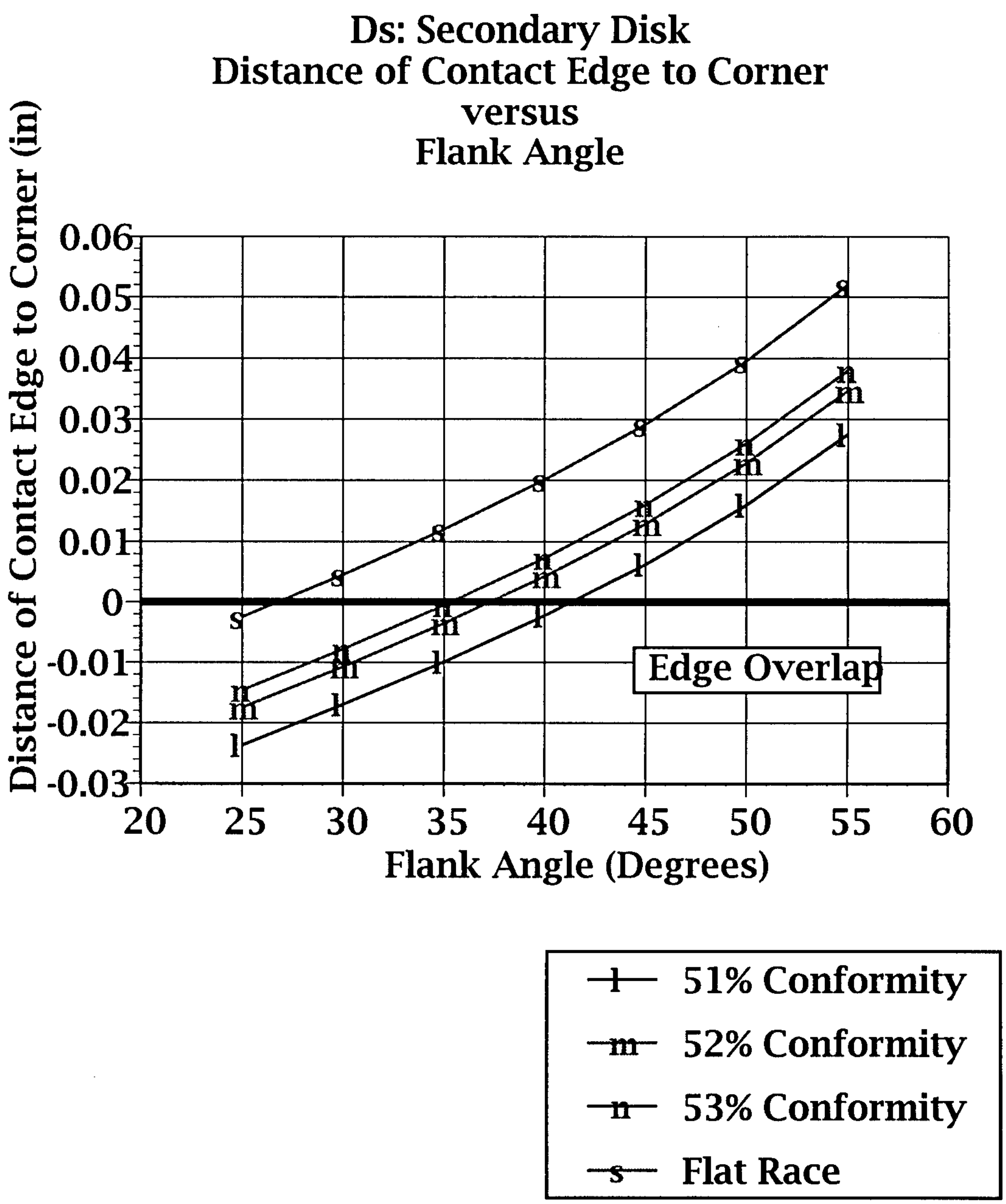


Figure 2.2.8-8: Secondary disk distance of contact edge to corner.

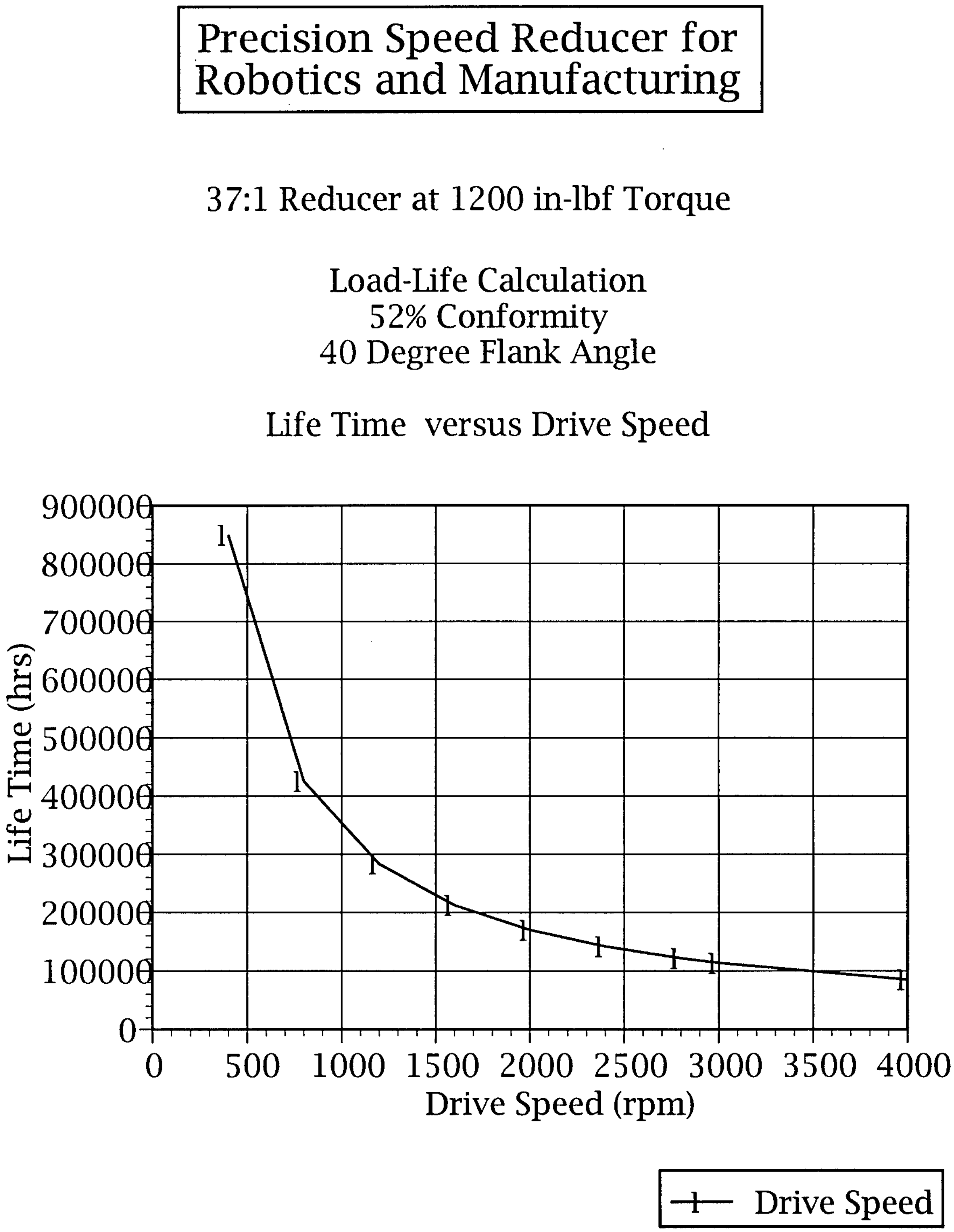
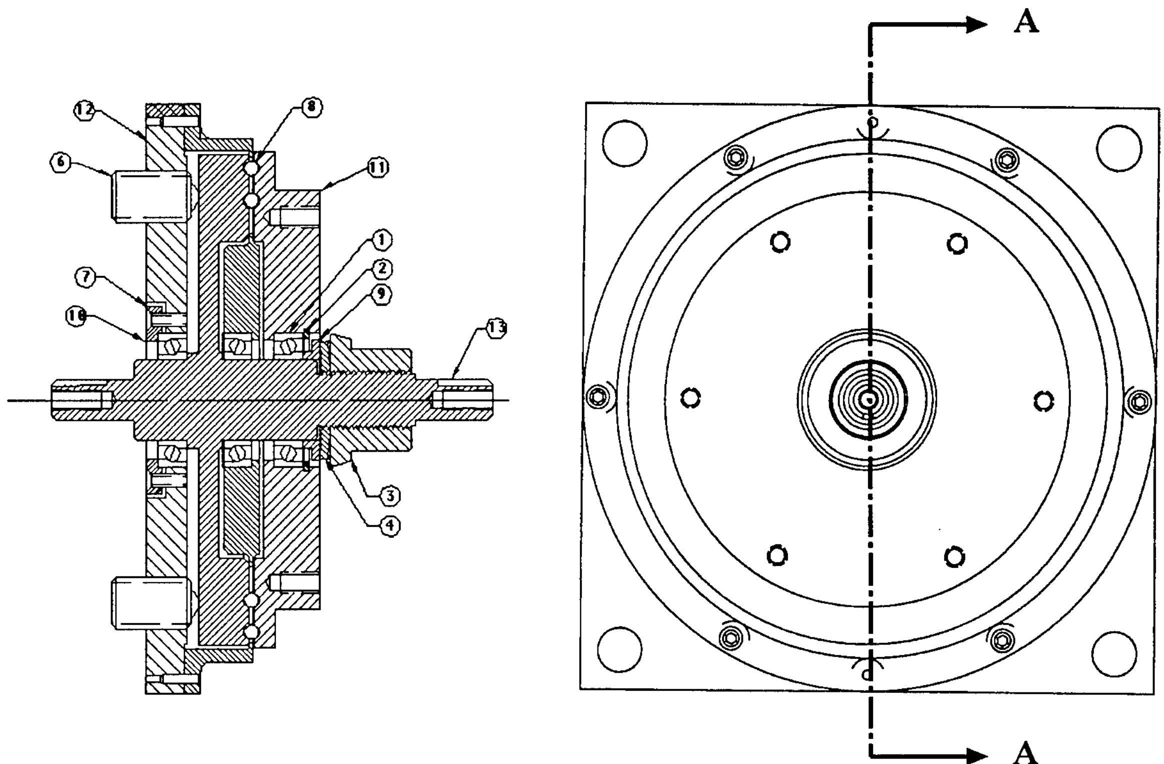


Figure 2.2.8-9: Load-life calculation for 52% conformity and 40% flank angle.



Section A-A

Front View

Figure 2.3-1: Section view of prototype device for fabrication.

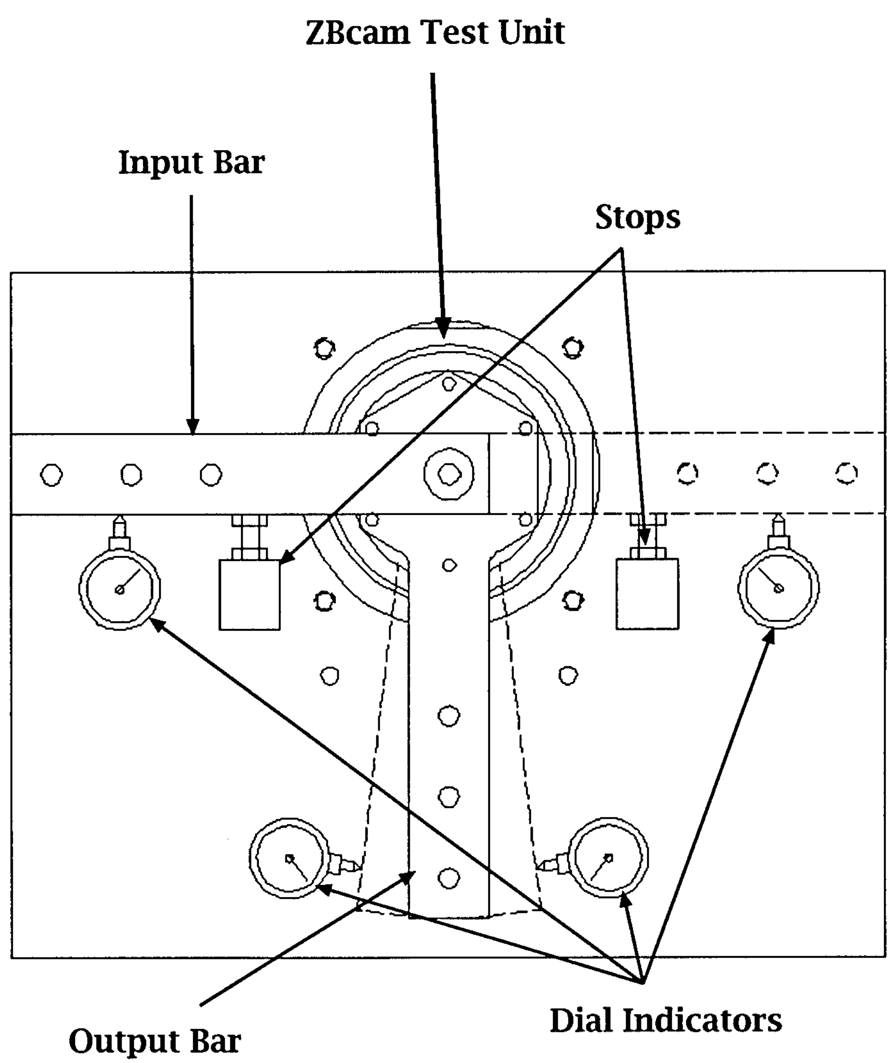


Figure 2.4.1.1-1: Prototype test assembly.

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