DRIVE SYSTEM DESIGN METHODOLOGY FOR A SINGLE MAIN ROTOR HEAVY LIFT HELICOPTER

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By

Andrew T. Bellocchio

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DRIVE SYSTEM DESIGN METHODOLOGY FOR A SINGLE MAIN ROTOR HEAVY LIFT HELICOPTER

Approved by:

Dr. Daniel P. Schrage, Advisor School of Aerospace Engineering *Georgia Institute of Technology*

Dr. Dimitri Mavris School of Aerospace Engineering *Georgia Institute of Technology*

Dr. David M. Sanborn School of Mechanical Engineering *Georgia Institute of Technology*

Date Approved: November 21, 2005

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LIST OF SYMBOLS

a, A	Area
А	Distance along an involute tooth profile
Access	Accessory Drive
a _{mr}	Adjustment factor for main rotor
a _{tr}	Adjustment factor for tail rotor
В	Bearing
b	Number of planets
BH	Brinnell Hardness
b_i	Coefficient for first order terms
b _{ii}	Coefficients for pure quadratic terms
b _{ij}	Coefficients for cross-product terms
c	Application factor or radius where stress is calculated
С	Gear constant
C_{f}	Surface condition factor
C_{H}	Hardness ratio factor
C _p	Elastic coefficient
Ср	Specific heat of oil
C _{xc}	Crowning factor for bevel contact stress
d	Pinion pitch diameter or inner shaft diameter
D	Gear pitch diameter or outer shaft diameter
d_{G}	Gear diameter

$d_{\rm P}$	Pinion diameter
ds	Drive system
d_{sh}	Drive shafting
Е	Modulus of elasticity
Eng	Engine
F	Force or net face width
\mathbf{f}_{a}	Axial tension stress
\mathbf{f}_{b}	Vibratory bending stress
F _{en}	Endurance limit stress
fpm	Feet per minute
$\mathbf{f}_{\mathbf{s}}$	Torsional shear stress
F_{sy}	Shear yield stress
F _{ty}	Tensile yield stress
G	Gear
g	gravity acceleration
gb	Gearbox
HP	Horsepower
HP _{rated}	Rated horsepower
HPratedDrive	Drive System Rated Power
HP _{req}	Horsepower Required
HRC	Hardness Rockwell C
Ι	Geometry factor for pitting resistance
Ι	Moment of inertia

in	Inch
J	Geometry factor for bending stress
J	Polar moment of inertia
K	Surface durability factor
k	Total endurance factor
K _b	Bending stress concentration factor
K _B	Rim thickness
K _m	Load distribution factor
Ko	Overload factor
K _R	Reliability factor
Ks	Size factor
Ks	Torsional stress concentration factor
k _t	Configuration factor
K _T	Temperature factor
K _v	Dynamic factor
L	Desired life or shaft length
L _{ab}	Line of action
lb	Pound
L _{dr}	Horizontal distance between rotor hubs
Μ	Bending moment or oil flow
max	Maximum
m _c	Contact ratio
m _G	Gear reduction ratio

m _N	Load sharing ratio
Mo	Overall ratio
MR	Main Rotor
MS	Margin of Safety
m _s	Ratio of planet to sun gear
Ν	Number of teeth
n	Speed (rpm)
N _c	Critical speed
N _{cycles}	Number of cycles
n _{dsh}	Number of drive shafts excluding rotor shaft
ngb	Number of gearboxes
n _P	Pinion speed
OC	Oil Cooler
р	Circular pitch
Р	Diametral pitch or planet gear
P _{nd}	Normal diametral pitch
psi	Pounds per square inch
$p_{\rm x}$	Axial pitch
Q	Heat generated
q	Number of contacts per revolution
$Q_{\rm v}$	Transmission quality number
r	radial distance of the desired stress point on shaft
R	Ring gear or response of interest

r, R	Pinion and gear pitch radii
R99	Reliability constant for 1 failure in 100
R _c	Rockwell Hardness (C)
Redux	Stage Reduction Ratio
R _{el}	Reliability constant
rpm	Rotations per minute
S	Sun gear
S _{ac}	Allowable contact stress
S _{at}	Allowable bending stress
Sc	Contact stress
$\mathbf{S}_{\mathbf{F}}$	Safety factor for bending stress
S_{H}	Safety factor for contact stress
st	Bending stress
Sut	Ultimate tensile strength
Т	Torque or kinetic energy
t _c	Contact temperature
$t_{\rm fl}$	Flash temperature
Th	Thrust
t _M	Bulk temperature
T _{mrgb}	Ratio of transmission hp to main rotor rpm
t _{oil}	Oil temperature
T_{oilavg}	Average oil temperature
T _{oilin}	Oil in temperature

T _{oilout}	Oil out temperature
TR	Tail Rotor
T _{trgb}	Ratio of tail rotor power to tail rotor rpm
V	Potential energy
v	Volume
W	Weight or total force
Wa	Axial force or thrust load
wi	Lumped weight
Wr	Radial force
\mathbf{W}_{t}	Tangential force or transmitted load
\mathbf{x}_{ij}	Independent variable
у	Shaft deflection
y _i	Element deflection
Y_N	Stress cycle factor for bending stress
Z _{mr}	Number of stages in main rotor drive
Z_N	Stress cycle factor
ΔΤ	Temperature rise across gearbox
μ	Poisson's ratio
ν	Coefficient of variation
φ	Pressure angle
ϕ_t	Transverse pressure angle
ψ	Helix angle
ΔT	Temperature rise

- ε Error term
- η Efficiency
- ω shaft frequency

SUMMARY

The transformation of Joint forces to be lighter, more lethal, and capable of deploying from multiple dispersed locations free of prepared landing zones will widen a capability gap that cannot be met by the current fleet of Joint aircraft. In order to close this capability gap, a dedicated heavy lift VTOL aircraft must rapidly deliver large payloads, such as the 20 to 26 ton Future Combat System, at extended ranges in demanding terrain and environmental conditions.

Current estimates for a single main rotor configuration place the design weight over 130,000 pounds with an installed power of approximately 30,000 horsepower. Helicopter drive systems capable of delivering torque of this magnitude succeeded in the Russian Mi-26 helicopter's split-torque design and the Boeing VERTOL Heavy Lift Helicopter (HLH) prototype's traditional multi-stage planetary design. The square-cube law and historical trends show that the transmission stage weight varies approximately as the two-thirds power of torque; hence, as the size and weight of the vehicle grows, the transmission's weight becomes an ever-increasing portion of total gross weight. At this scale, optimal gearbox configuration and component design holds great potential to save significant weight and reduce the required installed power.

The presented drive system design methodology creates a set of integrated tools to estimate system weight and rapidly model the preliminary design of drives system components. Tools are provided for gearbox weight estimation and efficiency, gearing, shafting, and lubrication and cooling. Within the same architecture, the designer may add

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similar tools to model subcomponents such as support bearings, gearbox housing, freewheeling units, and rotor brakes.

Measuring the relationships between key design variables of these components and system performance metrics reveals insight into the performance and behavior of a heavy lift drive system. A parametric study of select design variables is accomplished through an intelligent Design of Experiments that utilizes Response Surface Methodology to build a multivariate regression model. The model permits visualization of the design space and assists in optimization of the drive system preliminary design.

This methodology is applied to both the Boeing HLH and the Russian Mi-26. Both designed in the late 1970's, the tandem rotor HLH fails to take full advantage of the tremendous benefits gained by dividing the input torque into multiple, high speeds paths and then recombining the split paths at the final stage. The Mi-26 has successfully employed a split torque gearbox in the field for over 20 years. This study applies the drive system design methodology to compare the split-torque gearbox over a multi-stage planetary gearbox in a single main rotor heavy lift helicopter.

SCOPE

The scope of this thesis includes the furnished single main rotor baseline design and drive system requirements, drive system configuration for a multi-stage planetary main gearbox and a split-torque main gearbox, weight estimation, and preliminary design tools for the following drive components:

- 1. Gearboxes
- 2. Shafting
- 3. Gearbox cooling
- 4. Accessory accommodation

A designer has the freedom to incorporate additional analysis into the model. Some recommended elements that should be included for a complete preliminary design include:

- 1. Bearings
- 2. Housing
- 3. Keyways
- 4. Seals
- 5. Couplings
- 6. Structural integration

7. Failure warning and health systems The thesis obtains the impact of excursions through a Design of Experiments (DoE) and utilizes Response Surface Methodology (RSM) to explore the response behavior of the drive system through the

design space defined by excursions (E1, E2, E4, E5, and E6). The thesis limits the study to:

- 1. Single main rotor helicopter
- 2. Specified design space
- 3. Simplified rotor loads
- 4. Accessory accommodation

MOTIVATION

Need for a Joint Heavy Lift Helicopter

The military and commercial industries have great need for a heavy lift VTOL aircraft capable of transporting large payloads over longer distances and at a high rate of speed. Current transport fixed wing aircraft require large runways and support facilities, thus severely restricting a military commander's or commercial operations officer's freedom. The heavy lift VTOL aircraft fills a capability gap by providing heavy lift capability at high speeds independent of large runways.

For the military, the transport of large payloads is possible with fixed wing aircraft such as the C-130, C-141, or C-5. These assets, however, have their limitations as they are tied to fixed-base operations and large runways. A commercial airline experiences the same trade off within its fleet of commercial transport airplanes. The modern V-22 achieves a new degree of freedom through its vertical take-off capability and high-speed cruise though it cannot lift large payloads. The largest heavy lift helicopter in the world, the Mi-26, is capable of transporting the requisite 20-ton payload but is restricted from shipboard operations and reaches only 150 knots. There is not an aircraft in existence that can couple the freedom gained by VTOL, a high cruise speed, and a high payload capacity.

This mission is further detailed in the Draft Version 6.1 of the Initial Capabilities Document of the Joint Heavy Lift (JHL) Supplemental Package and is summarized by the associated Statement of Objectives:

"The JHL is expected to overcome enemy anti-access strategies, execute joint-enabled operational maneuver, and leverage sea basing in order to expand Expeditionary Maneuver Warfare capabilities. It [JHL] is also expected to achieve positional advantage to maintain operational momentum; enable SOF infiltration & exfiltration operations; conduct mounted and dismounted vertical envelopment; and perform aerial delivery operations."¹

The military is searching for new ways to provide a rapid, flexible response in a hostile environment without being dependent upon runways. Furthermore, the military is looking for a way to bypass traditional points of embarkation in exchange for the flexibility to operate directly from ships. The overall goal is to transport light combat vehicles at payloads, ranges, and speeds beyond what is considered feasible for traditional helicopters. The heaviest payload includes the Future Combat System (FCS) at a weight of 20-26 tons (see Figure 1). Despite the Mi-26's success, there exists no shipboard capable rotorcraft with the payload capacity, cruise speed and range to accomplish this mission.



Figure 1: The Future Combat System

¹ Aviation Applied Technology Directorate, "Joint Heavy Lift Supplemental Package: Statement of Objectives" (Fort Eustis: AATD, 4 November, 2004), 1.

Square Cube Law

The square-cube is a mathematic principle of proportion that is often applied to engineering and biomechanics. The square-cube law states "when an object undergoes a proportional increase in size, its new volume is proportional to the cube of the multiplier and its new surface area is proportional to the square of the multiplier."² This proportional increase is illustrated in Figure 2.

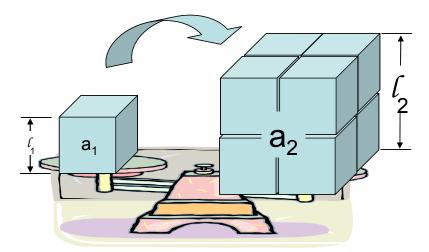


Figure 2: Square Cube Law Block

Relating volume to area will mathematically express this law as:

$$v_2 = v_1 \left(\frac{\lambda_2}{\lambda_1}\right)^3$$
 and $a_2 = a_1 \left(\frac{\lambda_2}{\lambda_1}\right)$

Equation 1

where

 l_1 is the original length and l_2 is the new length v_1 is the original volume and v_2 is the new volume

² Wikipedia website, http://en.wikipedia.org/wiki/Square-cube_law, 1 August, 2005.

 a_1 is the original surface area and a_2 is the new surface area

For example, if the length, l_1 , of a cube shown in Figure 2 is doubled in length to l_2 , the cube's surface area is increased by 2^2 or by a factor of 4. For volume, if l_2 is doubled, then volume increases by 2^3 or 8 times the cube's original volume.

By equating the expressions in Equation 1 and considering constant material, uniform density, and constant speed (for rotating components such as gears, shafts, and bearings) the square-cube law may be expressed as:

$$W_2 = W_1 \left(\frac{T_2}{T_1}\right)^{3/2}$$

Equation 2

where

 W_1 is the original weight and W_2 is the new weight T_1 is the original torque and T_2 is the new torque required

This relationship has a tremendous impact on the engineering of structures and mechanical systems such as mechanical power transmitting drive trains. As the size or torque transmitted of the drive system increases, the system mass will outpace the torque change at an exponential rate. This application of the square-cube law provides a general guideline to the resizing of mechanical systems as a function of the torque of 1.50 (Equation 2). In fact, the AMCP 706-201 predicts new gear stage weight as a power of 1.43.³

³ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design Handbook:</u> <u>Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 4-66.

The AMCP approximates gear stage weight as:

$$W_{stage_2} = W_{stage_2} \left(\frac{T_2}{T_1}\right)^{1.43}$$

Equation 3

According to the AMCP, the change in shafting weight is an even more dramatic power of 2.63:

$$W_{shaft_2} = W_{shaft_2} \left(\frac{T_2}{T_1}\right)^{2.63}$$

Equation 4

An increase in weight against the change in torque required is shown graphically for Equation 2, Equation 3, and Equation 4 in Figure 3.

At higher and higher relative torque, a drive system grows enormous in size and especially weight. If carried to the extreme, the size of a system will reach a practical limit where the system will be unable to support itself and buckle under its own massive weight. Such limits are common in the civil engineering of a skyscraper's structure. For aerospace engineering, the aircraft structure or drive system may never reach such an extreme point, but the design will certainly reach a practical limit where the generated lift cannot sustain flight of the aircraft's large gross weight.

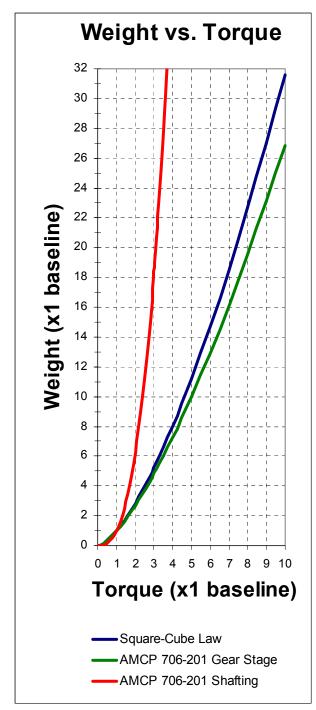


Figure 3: Square-Cube Law Predictions of Weight verse Torque

<u>The Importance of Heavy Lift Drive Systems</u>

Current estimates for a single main rotor heavy lift configuration place the design weight over 130,000 pounds with an installed power of approximately 30,000 horsepower. At such high gross weights and loads, the drive system becomes an everincreasing percentage of the aircraft gross weight.

For heavy lift rotorcraft and drive systems, increasing total lift exacerbates this exponential weight increase. To generate sufficient lift, the rotorcraft requires a larger power plant and either more lifting rotors or larger rotor blades. Either condition increases drive system weight. Adding more rotors is an alternative method to avoiding high single torque values and excessively heavy components; however, much of this benefit is lost by duplicate gearing and heavy cross shafting.

Increasing the rotor blade radius presents a different set of problems in the quest for additional lift. With the tip speed of rotor blades fixed constant by compressible flow limits, a larger rotor demands slower rotor rotational speed (rpm). Slower rotational speed requires a greater drive system reduction ratio and results in higher torque values at the end of the drive train. Drive systems capable of higher torque values demand larger and heavier gears, shafting, and bearings. Dividing the torque into multiple paths and combing the torque at the end of the drive train may avoid a portion of this gear and shafting weight gain. Such a system is called a split-torque design and is explored in this study as an alternative to traditional planetary gearing. Despite the benefits of a splittorque, a final reduction to the slower rotation speed is inevitable. Split-torque designs cannot eliminate the heavy gearing and support bearings at the last stage but may lessen the severity of the total system weight gain.

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The use of face gearing to combine the split torque paths, however, does hold promise to further mitigate the weight gain in the last stage. Unfortunately, the gearing community has yet to standardize face gear stress formulas⁴—removing face gear drives from consideration for this preliminary design model. Despite the current immaturity of face gear standardization, an optimized split-torque drive utilizing spur and bevel gears still holds great potential to save significant weight and reduce the required installed power.

⁴ F.L. Litvin et al, <u>NASA/CR-2000-209909 Handbook on Face Gear Drives With a Spur Involute</u> <u>Pinion</u> (NASA, March 2000), 48.

JHL BASELINE

As part of the JHL's Concept Design and Analysis (CDA) effort, the government has solicited industry to assist in "the conceptual design of a baseline VTOL platform configuration and the exploration of the technical trade space associated with that concept."⁵ The CDA's focus is on "the substantiation of viable design concepts" within the desired trade space that "have a reasonable chance of achieving TRL 6 by 2012."⁶ The aircraft's trade space shall meet the desired capabilities described in the Draft Initial Capabilities Document (ICD) and the design flight profile as defined in the Model Performance Specification (MPS). Additionally, the CDA will assess the impact of a set of excursions from the design baseline. This thesis strives to provide insight into the performance and behavior of a heavy lift drive system within the CDA bounds.

Government Furnished Baseline

To assist in the CDA development and to provide a common starting point, the government has furnished conceptual sizing and performance data for a single main rotor helicopter configuration (extracted information relative to the thesis is included in APPENDIX A: JHL SUPPLEMENTAL PACKAGE EXTRACTS). The work of this

⁵ Aviation Applied Technology Directorate, "Joint Heavy Lift Supplemental Package: Statement of Objectives," (Fort Eustis: AATD, 4 November, 2004).

⁶ Aviation Applied Technology Directorate, "Joint Heavy Lift Supplemental Package: Statement of Objectives," (Fort Eustis: AATD, 4 November, 2004).

study is to provide a preliminary design and optimization of the drive system as defined by the furnished single main rotor baseline. This baseline aircraft is a traditional single main rotor helicopter design with a five-bladed main rotor and classic anti-torque tail rotor as shown in Figure 4.



Figure 4: Single Main Rotor JHL⁷

The power plant consists of three scaleable turbine engines with the drive system combining engine torque at the main gearbox. A long, segmented shaft transmits torque to the tail rotor intermediate gearbox that supplies a direction change at the base of the

⁷ Wayne Johnson, "Heavy Lift Rotorcraft Plans and Status" (Ames Research Center: NASA, 8 June, 2004).

vertical tail. A shorter shaft connects the intermediate gearbox with the tail rotor gearbox. Figure 5 on the next page shows the 3-view drawing of the furnished, single main rotor baseline aircraft.

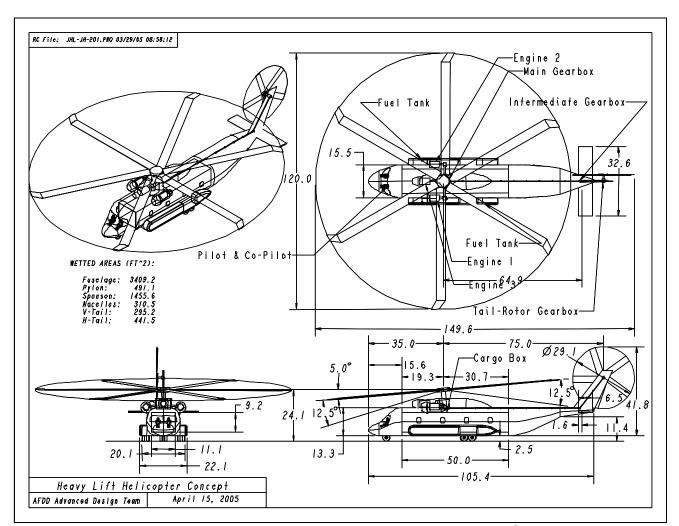


Figure 5: Single Main Rotor JHL 3-View Drawing⁸

⁸ Aviation Applied Technology Directorate, "Joint Heavy Lift Supplemental Package: Government Furnished Information Listing, Sample Data Formats, JHL Baseline Drawings," (Fort Eustis: AATD, November 4, 2004).

Design Excursions

Seven design excursions or alternate sizes from the baseline shall assist in the understanding of varying user design requirements. The purpose of the excursions is to provide trade space information to assess the impact of changing user requirements. The design excursions vary one mission parameter at a time (underline variables represent the baseline design values):⁹

- 1. Design Payload Excursion: 16 ton, 20 ton, 26 ton
- 2. Design Radius Excursion, 210 nm, 250 nm, 400 nm, 500 nm
- 3. Design Hi/Hot Excursion: <u>4,000'/95°F</u>, 6,000'/95°F

These excursions serve as the basis to the formulation of the design space for the drive system design in this thesis. The seven excursions are summarized in Table 1 and Table 2 below. Further information is in found throughout APPENDIX A: JHL SUPPLEMENTAL PACKAGE EXTRACTS and especially in Table 23 and Table 24.

⁹ Aviation Applied Technology Directorate, "Joint Heavy Lift Supplemental Package: Contract Data Requirements List A005 Block 16 Remarks Continuation Sheet," (Fort Eustis: AATD, November 4, 2004).

Design	Baseline	E1	E2	E2A*	E3
Payload	20	16	26	26	20
Radius	250	250	250	250	210
High/Hot, 1000ft/deg F	4k/95	4k/95	4k/95	4k/95	4k/95
Design Gross Weight	138,868	114,035	177,392	xxx,xxx	xxx,xxx
Engine Size, shp	10,985	9,114	13,912	xxx,xxx	xxx,xxx
Drive Rating	25,964	21,420	33,011	xx,xxx	xx,xxx
(TO rpm), shp					

Table 1: Design Excursions E1-E3¹⁰

*E2A has larger cargo box size to fit FCS Full Combat Configuration. All others have standard cargo box size to fit FCS Essential Combat Configuration.

Italics are variation from baseline design variable. Blue is low variation. Red is high variation.

Design	Baseline	E4	E5	E6	E7*
Payload	20	20	20	20	20
Radius	250	400	500	250	250
High/Hot, 1000ft/deg F	4k/95	4k/95	4k/95	6k/95	4k/95
Design Gross Weight	138,868	172,197	206,501	148,606	xxx,xxx
Engine Size, shp	10,985	13,495	16,066	13,113	xx,xxx
Drive Rating (TO rpm), shp	25,964	30,642	35,469	27,567	xx,xxx

 Table 2: Design Excursions E4-E7

*E7 must be shipboard compatible. All others are shipboard capable.

Italics are variation from baseline design variable. Blue is low variation. Red is high variation.

¹⁰ Aviation Applied Technology Directorate, "Joint Heavy Lift Supplemental Package: Government Furnished Information Listing, Sample Data Formats, JHL-Design Excursions Description Capabilities," (Fort Eustis: AATD, November 4, 2004).

The baseline, E1, E2, E3, E4, E5, and E6 provide valid trade space on changing payload, radius, and hi/hot atmosphere. For simplicity, the study will omit excursions E2A (larger cargo box) and E7 (shipboard capable), as it does not measure payload, radius, or atmospheric performance. E3 is also excluded, as sufficient data is unavailable.

Design Space

The Joint Heavy Lift Supplemental Package provides sufficient information to establish a baseline drive system specific design space. Data arises from the tables in APPENDIX A: JHL SUPPLEMENTAL PACKAGE EXTRACTS or from calculations from the same data. A few calculations are:

- 1. A constant tip speed of 725 ft/s and rotor diameter allowed the calculation of the main rotor speed.
- Main rotor thrust is the product of the constant disc loading at 12.3 lb/ft² and the provided effective disk area.
- 3. An examination of the scaleable NASA high tech turbine engine for heavy lift gave an approximate speed range from 10,000 to 20,000 rpm.

Symbol	Metric	Units	Low	Baseline	High
HP _{reqMR}	Main Rotor Power Required	hp	14,000	22,247	27,000
HP _{reqTR}	Tail Rotor Power Required	hp	1,500	1,989	3,200
HP _{engMRP}	Engine Maximum Rated Power	hp	9,114	10,985	16,066
HP _{reqAccess}	Accessory Power Required	hp	60	120	180
n _{MR}	Main Rotor Speed	Rpm	90	115	130
n _{TR}	Tail Rotor Speed	Rpm	375	476	530
n _{eng}	Engine Speed	Rpm	10,000	15,000	20,000
Th _{MR}	Main Rotor Thrust	lb	111,156	139,113	206,480

Table 3: Design Space

Italics indicate estimates.

When data was lacking, estimated values were selected to best represent a feasible range. Estimates originate from a comparison to other aircraft (mainly the Boeing HLH and Mi-26), extrapolation using the square cube law, and engineering judgment. The design space as shown in Table 3 established the bounds for the analysis conducted in the remaining sections.

ASSUMPTIONS

Major assumptions for the thesis include:

- 1. Scaling for aircraft excursions follows square-cube law
- 2. Design space extremes are in appropriate and valid ranges
- Reference Heavy Lift NASA High Tech Engine deck for engine output speed
- 4. Tail rotor tip speed equals main rotor tip speed (725 ft/s)
- Design of excluded components (bearings, splines, keyways, seals, etc) does not significantly impact the transmission design
- 6. Estimate accessory power requirements
- 7. Simplified model for rotor loads on main and tail rotor drive shafts
- 8. Detailed geometry complies with baseline drawings
- 9. Shipboard compatible and folding/stowage shall not impact drive system design with the exception of a rotor brake
- 10. Main rotor and tail rotor shafts are nonuniform while all others are uniform with bearings to absorb loading

METHODOLOGY

Design Process

The preliminary design and optimization is a three step process (Figure 6) of:

- 1. Drive System Configuration
- 2. Drive System Weight Estimate
- 3. Capture Responses and Conclusions

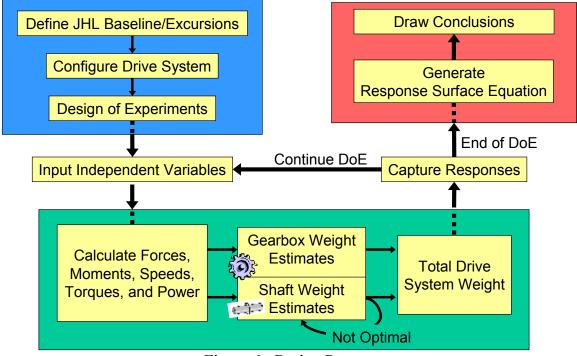


Figure 6: Design Process

The first step, Drive System Configuration (upper left box), involves defining the supplied Single Main Rotor (SMR) JHL configuration. Associated with this is a

component layout to include selection of gearboxes and shafting connecting gearboxes. Next, a screening test selected the major design parameters that most influence the drive system behavior. Once identified, the combinations of key design parameters constitute a Design of Experiments (DoE). Each independent design parameter is varied within the defined design space and in accordance with the DoE.

The middle step (shown as a bottom box), models the actual drive system response for a particular set of inputs. The Weight Estimates section contains three drive system weight estimate methods (modified solid rotor volume, Boeing-Vertol, and Research Technology Laboratories). Here, a designer has the option to model all drive system components similar to the gearing, shafting, and gearbox cooling tools in the Appendices. The accuracy of the weight estimates and the integration of each toolbox to provide a functional, weight-optimized design is the most crucial step in generating realistic solutions.

The final step, Capture Responses and Conclusions (upper right box), captures the output responses for each configuration and uses Response Surface Methodology to determine system behavior. For every set of design parameters, output responses for the solution shall be captured and used to generate a Response Surface Equation (RSE). With the RSE constructed, a family of optimized solutions maybe provided for the drive system. The RSE shall also enable the design to quickly evaluate the impact of changing design parameters through the design process. The accuracy of the weight RSE maybe judged against the two well establish weight estimates as well as compared to similar aircraft like the Mi-26 or CH-53.

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<u>Methodology for the JHL</u>

According to Lynwander, "the function of a gearbox is to transmit rotational motion from a driving prime mover to a driven machine."¹¹ In the case of a helicopter, the driver is one or more turbine engines and the driven members include lifting rotors, anti-torque device, and any mechanically driven accessories. A drive system is composed of a series of gears linked together by shafting, supported by bearings, enclosed and mounted by housing, and lubricated and cooled by oil.

The following key principles support the drive system design:

- 1. Transmission loads are a function of power and speed: (T = HP/rpm)
- 2. Input rpm is fixed by output speed of engine
- 3. Rotor speed is a function of blade tip speed and rotor diameter
- 4. Best to take largest reductions in the final stage

Most of the transmission elements can be broken down into:

- 1. Gears
- 2. Shafting
- 3. Bearings 🥥
- 4. Freewheel units (clutches) and Rotor Brake
- 5. Lubrication systems
- 6. Housing

¹¹ Peter Lynwander, <u>Gear Drive Systems Design and Application</u> (New York: Marcel Dekker, 1983), iii.

Each of these components can be designed into toolboxes, represented by an icon, that will analyze forces, size the component, and select the correct configuration. For the demonstration of this methodology in the planetary and split torque gearboxes, only gearing, shafting, and gearbox cooling are included. The designer has the freedom to integrate toolboxes as needed. The above list is the suggested minimum for a complete preliminary design.

For the included toolboxes, gear capacity is derived from American Gear Manufacturer's Association (AGMA) standards and includes gear type selection in a variety of external and internal arrangements, evaluation of axial and radial loading, sizing for bending and compressive stress (pitting), and reduced scoring hazard. Shafting withstands simultaneous axial tension stress, vibratory bending stress, and torsional shear stress while ensuring appropriate operation away from critical speeds. Lubrication absorbs gearbox-generated heat through a series of independent splash and force feed oil systems.

The overall system configuration is defined in the government furnished JHL baseline. Figure 7 shows the drive system layout as derived from the furnished 3-view drawings. Included are suggested toolboxes for a complete preliminary design.

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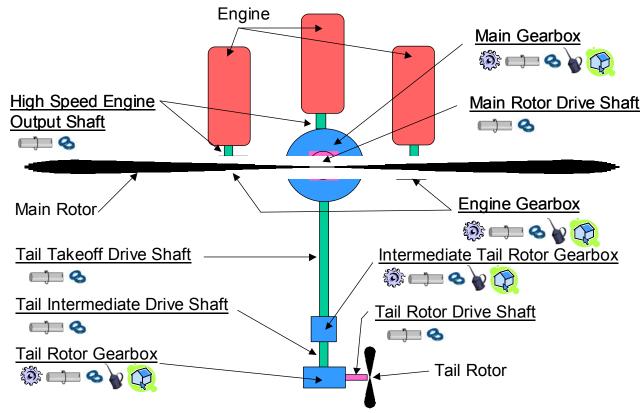


Figure 7: Furnished Baseline Drive System Configuration

WEIGHT ESTIMATION

Three different weight estimation methods are calculated for iterations in the planetary and split torque gearbox model. The solid rotor volume by Willis assumes gear weight is proportional to the solid rotor volume (Fd²) and uses the surface durability factor K for computation.¹² The next two methods are weight equations taken from a NASA sponsored comparative study of Soviet vs. Western helicopters (1983).¹³ The weight equations are from Boeing-Vertol and Research Technology Laboratories (RTL) as cited in the NASA study. The comparative study's Soviet set of equations were from Tishchenko's Soviet Weight Formulae; however, his estimation had to be excluded because the formulae as shown were for tandem helicopters only.

<u>Solid Rotor Volume Method</u>

R.J. Willis's solid rotor volume weight estimation method provides a simple way to estimate gearbox weight and select the optimal reduction ratio in multiple stage drives. The method assumes the weight of a gear drive is "proportional to the solid rotor volume

¹² R.J. Willis, "New Equations and Chart Pick Off Lightest-weight Gears," <u>Product Engineering</u> v. 34, n.s. 2 (January 21, 1963): 64.

¹³ W.Z. Stepniewski and R.A. Shinn, <u>NASA TR 82-A-10 A Comparative Study of Soviet vs. Western</u> <u>Helicopters: Part 2-Evaluation of Weight, Maintainability and Design Aspects of Major Components</u> (Ames Research Center: AVRADCOM Research and Technology Laboratories, 1983).

(Fd²) of the individual gears in the drive"¹⁴ and that a surface durability factor, K, of each mesh is constant. The method is explained in Willis's "Lightest-weight Gears" published in *Product Engineering* in 1963.¹⁵ Similar methods are outlined in Dudley's <u>Handbook</u> of Practical Gear Design (1994) and in AGMA 911-A94, *Information Sheet-Design Guidelines for Aerospace Gearing* (1994).

Simple Gear Mesh

The solid rotor volume maybe expressed as:

$$Fd_{P}^{2} = \frac{126,000HP}{Kn_{P}} \left(\frac{m_{G}+1}{m_{G}}\right) = \frac{2T}{K} \left(\frac{m_{G}+1}{m_{G}}\right)$$

Equation 5

where

F is pinion face d_P is pinion diameter HP is horsepower K is surface durability factor n_P is the pinion speed m_G is reduction ratio T is torque

The solid rotor volume of the gear is proportional to the gear ratio squared:

$$Fd_G^2 = Fd_P^2 m_G^2$$

Equation 6

¹⁴ R.J. Willis, "New Equations and Chart Pick Off Lightest-weight Gears," <u>Product Engineering</u> v. 34, n.s. 2 (January 21, 1963): 64.

¹⁵ R.J. Willis, "New Equations and Chart Pick Off Lightest-weight Gears," <u>Product Engineering</u> v. 34, n.s. 2 (January 21, 1963): 64-75.

The total weight of a gearbox is a function of the sum of solid rotor volume of each gear. For a simple pinion-gear mesh:

$$\sum Fd^2 = Fd_P^2 + Fd_G^2$$

Equation 7

Combining Equation 5, Equation 6, and Equation 7 yields:

$$\sum Fd^2 = \frac{2T}{K} \left(\frac{m_G + 1}{m_G} \right) + \frac{2T}{K} \left(\frac{m_G + 1}{m_G} \right) m_G^2$$

Equation 8

For ease of notation, let: C = 2T / K. Equation 8 can reduce to:

$$\sum \frac{Fd^2}{C} = 1 + \frac{1}{m_G} + m_G + m_G^2$$

Values for the surface durability factor, K, are listed in Table 4.

Application	Service		Hardness		PLV	Accuracy	K
	Characteristics						factor
	Driver	Driven	Pinion	Gear			lb/in ²
Aircraft*	Engine	Auxiliary	58 Rc	58 Rc	10,000	High	1,000
(single pair)		Drive				ground	
Aircraft*	Engine	Propeller	58 Rc	58 Rc	3,000 -	Ground	600
Planetary					10,000		
Carburized			58 Rc	58 Rc	10,000	Ground	500-
Aerospace					and		600
Gears [#]					beyond		

Table 4.	K Factors	for Gears
1 aute 4.	IN F ACTORS	IUI Geals

* for helical and spur gears listed by Willis

[#] listed by AGMA 911-A94 for aerospace carburized and case hardened gears

According to Willis, estimating actual weight is possible by multiplying the total gearbox solid rotor volume by an application factor, c, where c is 0.25 to 0.30 for aircraft application.¹⁶ This weight factor assumes magnesium or aluminum casings, limited life design, high stress levels, and rigid weight controls—all properties that are included in the drive system design of the JHL.

$$W_{gearbox} = c \sum F d^2$$

Equation 9

One of the advantages of the solid rotor method is that the estimated weight includes the entire gearbox with housing, bearings, oil, and gear shafting; thus, the weight of an entire drive system maybe quickly calculated by the sum of the gearbox weights

¹⁶ R.J. Willis, "New Equations and Chart Pick Off Lightest-weight Gears," <u>Product Engineering</u> v. 34, n.s. 2 (January 21, 1963): 70.

and the total shafting weight (as calculated in section SHAFTING on page 86). The accuracy of the weight estimation is a product of the accuracies of the K factor and the application weight factor, c.

Composite Gear Systems

In addition to the weight estimation, Willis details a means to determine the optimal stage reduction ratio that provides the lightest overall weight for the gear system. Most of the stage reduction ratio optimization employs a series of graphs that is not conducive to computer modeling.

Epicyclic or planetary systems may also be optimized with the solid rotor volume method. For a planetary gearset, let:

$$m_s = \frac{d_p}{d_s} = \frac{M_o}{2} - 1$$

Equation 10

where

 m_s is ratio between planet and sun gear M_o is overall ratio P is planet gear subscript S is sun gear subscript R is ring gear subscript

If b represents the number of planets (or branches) in the gearset, then the sun gear volume is:

$$Fd_{s}^{2} = \frac{2T}{bK} \left(\frac{m_{G1} + 1}{m_{G1}} \right)$$

Equation 11

From Equation 10, one can deduce:

$$bFd_P^2 = bFd_s^2 m_s^2$$

Equation 12

The relationship between the ring gear and the sun gear is:

$$Fd_r^2 = Fd_s^2 \left(\frac{d_r}{d_s}\right)^2 0.4$$

Equation 13

where

0.4 is an adjustment to account for weight of the cage structure and housing

The total weight of the planetary gear system is sum of the sun gear weight,

number of planets times the gear weight, and the ring gear weight.

$$\sum Fd^2 = Fd_s^2 + bFd_P^2 + Fd_r^2$$

Equation 14

Substituting Equation 12 and Equation 13 into Equation 14 yields:

$$\sum Fd^{2} = Fd_{s}^{2} + Fd_{s}^{2}m_{s}^{2} + Fd_{s}^{2}\left(\frac{d_{r}}{d_{s}}\right)^{2} 0.4$$

Equation 15

After simplification, the following expression is reached:

$$\sum \frac{Fd^2}{C} = \frac{1}{b} + \frac{1}{bm_s} + m_s + m_s^2 + \frac{0.4(M_o - 1)^2}{b} + \frac{0.4(M_o - 1)^2}{bm_s}$$

Equation 16

Boeing-Vertol Weight Formulae

The following weight equations (Equation 17 to Equation 19) are taken from NASA Technical Report 82-A-10, *A Comparative Study of Soviet vs. Western Helicopters Part 2.*¹⁷

The weight of the main rotor drive system is:

$$(W_{ds})_{mr} = 250a_{mr} \left[(HP_{mr} / rpm_{mr}) z_{mr}^{0.25} k_t \right]^{0.67}$$

Equation 17

where

 a_{mr} is an adjustment factor (assumed to be 1) HP_{mr} is drive system horsepower rpm_{mr} is main rotor rpm

¹⁷ W.Z. Stepniewski and R.A. Shinn, <u>NASA TR 82-A-10 A Comparative Study of Soviet vs. Western</u> <u>Helicopters: Part 2-Evaluation of Weight, Maintainability and Design Aspects of Major Components</u> (Ames Research Center: AVRADCOM Research and Technology Laboratories, 1983), 61-4.

 z_{mr} is number of stages in main rotor drive k_t is configuration factor ($k_t = 1$ for single main rotor)

The weight of the tail rotor drive train with shafting is:

$$(W_{ds})_{tr} = 300a_{tr} [1.1(HP_{tr}/rpm_{tr})]^{0.8}$$

Equation 18

where

 a_{tr} is an adjustment factor ($a_{tr} = 0.9$) HP_{tr} is tail rotor horsepower rpm_{tr} is tail rotor speed

The total drive system weight is the sum of the main rotor drive and tail rotor

drive weight:

$$W_{ds} = \left(W_{ds}\right)_{mr} + \left(W_{ds}\right)_{tr}$$

Equation 19

<u>**RTL Weight Formulae**</u>

The following weight equations (Equation 20 to Equation 22) are taken from

NASA Technical Report 82-A-10, A Comparative Study of Soviet vs. Western

Helicopters Part 2.18

¹⁸, W.Z. Stepniewski and R.A. Shinn, <u>NASA TR 82-A-10 A Comparative Study of Soviet vs. Western</u> <u>Helicopters: Part 2-Evaluation of Weight, Maintainability and Design Aspects of Major Components</u> (Ames Research Center: AVRADCOM Research and Technology Laboratories, 1983), 66.

The RTL formulae take the drive system components as a function of transmitted

power and speed. The total gearbox weight is predicted with:

$$W_{gb} = 172.7T_{mrgb}^{0.7693}T_{trgb}^{0.079}n_{gb}^{0.1406}$$

Equation 20

where

where

W_{gb} is total gearbox weight

 $T_{mrgb} = HP_{ttrmr} / rpm_{mr}$ ratio of transmission hp to main rotor rpm $T_{trgb} = 100(HP_{ttr} / rpm_{tr})$ ratio of tail rotor power to its rpm n_{gb} is the number of gearboxes

The total shafting weight is:

$$W_{dsh} = 1.152 T_{mrgb}^{0.4265} T_{trgb}^{0.0709} L_{dr}^{0.8829} n_{dsh}$$

Equation 21

 W_{dsh} is total drive shafting weight L_{dr} is the horizontal distance between rotor hubs (feet) n_{dsh} is the number of drive shafts excluding rotor shaft

Total drive system weight is the sum of gearbox and shafting weight:

$$W_{ds} = W_{gb} + W_{dsh}$$

Equation 22

GEARING

Gear Fundamentals

This section includes a brief outline of fundamental gear geometry and properties that will be referenced throughout the remainder of the thesis. For further information, consult any machine design textbook such as Shigley and Mischke's <u>Mechanical</u> <u>Engineering Design</u>. ¹⁹ Another excellent source is the <u>Handbook of Practical Gear</u> <u>Design</u> by Dudley.²⁰ Figure 8 from Shigley and Mischke shows basic gear layout for the simplest of gearing types, the spur gear.

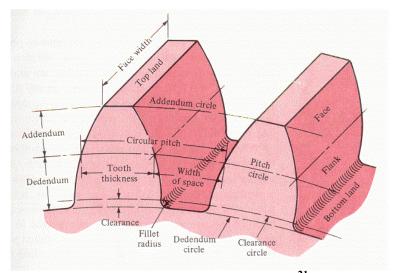


Figure 8: Gear Nomenclature²¹

¹⁹ Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989).

²⁰ Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994).

²¹ Figure extracted Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 529.

The *reduction ratio* or gear ratio, m_G, of a pair of gears is describes the ratio of pitch diameters of the larger gear to the smaller pinion.

$$m_G = \frac{N_2}{N_1} = \frac{d_2}{d_1} = \frac{n_1}{n_2}$$

Equation 23

where

 m_G is the reduction ratio N is the number of teeth d is the pitch diameter n is the rotational speed

The typical reduction ratio is from 1:1 to 10:1 for most gearing. Modern face gear designs can reach even higher ratios. As gears mesh, the larger gear has a slower rotational speed, n, than the smaller gear. This speed ratio is the same as the gear or reduction ratio and is also expressed in Equation 23.

Even though different size gears rotate at different speeds, the peripheral speed, or *pitch line velocity* (PLV), of a meshing gear pair is the same:

$$PLV = \frac{\pi dn}{12}$$

Equation 24

The *diametral pitch*, P, measures the size of the teeth with respect to the size of the gear and has units of teeth per inch.

$$P = \frac{N}{d}$$

Equation 25

The *circular pitch*, p, is the arc distance from one tooth width and space to the next tooth and has units of inches per tooth.

$$p = \frac{\pi d}{N} = \frac{\pi}{P}$$

Equation 26

The most common tooth shape is the *involute profile*. The involute profile creates a conjugate action similar to cams in order to maintain a steady torque over the meshing gear pair despite the fact that the load on an individual tooth constantly changes magnitude and direction. As involute teeth mesh, the point of contact between them changes. The line in which resultant forces act along is called the *line of action*, L_{cd}, and is shown as line cd in Figure 9. The *pressure angle*, φ , is the angle formed between the line of action and the tangency of the pitch circles. This tooth action is also showed in Figure 9.

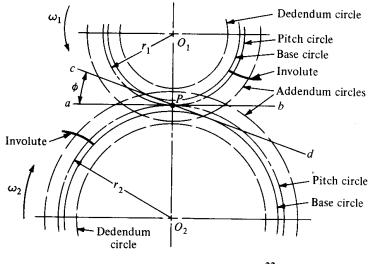


Figure 9: Tooth Action²²

The *contact ratio*, m_c, is the average number of tooth pair in contact during mesh. A higher contact ratio provides better load sharing, smoother meshing, and reduced noise.

Gear Failure Modes

Aerospace gearing is sized on three considerations: bending fatigue, surface compression (Hertz stress), and scuffing (scoring) resistance. A design must include adequate ability to resist all three types of failures. Of the three failures modes, tooth bending has "the most severe consequences . . . whereas pitting and scoring are durability type failures."²³

²² Figure extracted from Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 533. ²³ American Gear manufacturers Association, *AGMA 911-A94, Information Sheet-Design Guidelines*

for Aerospace Gearing (Alexandria: AGMA, 1994), 41.

Bending stress is concentrated tensile stress at the base of the tooth on the loaded side. A gear tooth can be thought of as a short cantilever beam with a force pushing at the end of the beam. The highest point of stress concentration will occur at the base of the beam, or for gears, at the root fillet (see Figure 10). The ability of a particular gear to resist this stress is called *allowable bending strength* and "is a function of the hardness and residual stress near the surface of the root fillet and at the core."²⁴ To determine failure, allowable bending stress is derated by factors such as dynamic loading, overloading, and reliability. This number is then compared to the bending stress.

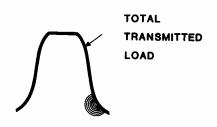


Figure 10: Bending Stress²⁵

Compressive or contact stress causes pitting that weakens the gear surface by increasing local stress concentrations. Gear teeth undergo compression and tension as the tooth rolls through the mesh with the mating tooth.²⁶ Over the life of a gear, this repetitive cycle progressively pits the surface until it eventually leads to a fatigue failure. Figure 11 shows the compressive stress point on a tooth. *Allowable Compressive*

²⁴ American Gear manufacturers Association, *AGMA 911-A94, Information Sheet-Design Guidelines* for Aerospace Gearing (Alexandria: AGMA, 1994), 6.

²⁵ Figure extracted from Peter Lynwander, <u>Gear Drive Systems Design and Application</u> (New York: Marcel Dekker, 1983), 96.

²⁶ American Gear manufacturers Association, *AGMA 911-A94, Information Sheet-Design Guidelines for Aerospace Gearing* (Alexandria: AGMA, 1994), 42.

strength measures the tooth surface's resistance to pitting. To increase compressive strength, aerospace gears are usually strengthened through carburized, case hardening. To determine failure, allowable compressive strength is derated by factors such as surface condition, hardness, and dynamic factors. This value is then compared to the contact stress.

COMPRESSIVE STRESS

Figure 11: Compressive Stress²⁷

As transmitted power increases, the bending stress increases linearly while compressive strength increase as the square root of transmitted power (Figure 12). For the same gear geometry and design, compressive stress will be the higher stress in regions of lower transmitted power while bending stress often dominates the higher power regions.

²⁷ Figure extracted from Peter Lynwander, <u>Gear Drive Systems Design and Application</u> (New York: Marcel Dekker, 1983), 103.

Gear Tooth Stress vs Power Stress (kpsi) Power (hp) -Bending Stress - Compressive Stress

Figure 12: Gear Tooth Stress vs. Power

The Preliminary Design Handbook recommends that a well balanced design occurs where the bending stress to allowable compress stress ratio and the compressive stress to allowable compressive stress are relatively equal and at the desired factor of safety.

Scuffing (scoring) failure occurs when the mating gear welds the metal surface of the mated gear. Although scoring is not a fatigue failure, excessive compressive stress over a period of time will create radial scratch lines in the surface that promotes the onset of scoring.²⁸ Scuffing is a durability failure that may occur instantaneously.²⁹ Heavily loaded, high-speed gears such as aircraft gears tend to fail by scoring.³⁰ The probability of a gear pair to resist scoring is called *scuffing resistance*.

 ²⁸ Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 2.24.
 ²⁹ American Gear Manufacturers Association, *AGMA Standard 2001-C95, Fundamental Rating Factors* and Calculation Method for Involute Spur and Helical Gear Teeth (Alexandria: AGMA, 1995), 9.

³⁰ Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 2.24.

Gear Types and Functions

In a drive system, gears serve to reduce or increase speed, change the direction of drive, and split or combine torque paths. A designer has several different types of gears to select from in order to best achieve the intended function. Spur, helical, and planetary gears transmit torque along a parallel axis. Bevel, worm, and face gears transmit torque along intersecting axis. Crossed helicals or hypoids are used for nonparallel axis gears. For helicopter transmissions, the nonparallel axis gears are not normally used because their efficiency "decreases rapidly as the helix angle increases."³¹ A low inefficiency also applies to worm gears. Table 5 outlines the gear types, functions, and typical uses in helicopter transmissions.

Gear Type	Axis Type	Function	Typical Use
Spur	Parallel	Speed reducer Combine/split paths	Planetary gearing Accessory gearing Tail rotor gearbox
Helical	Parallel	Speed reducer Combine/split paths	Low noise gearing High speed, high load
Bevel	Intersecting	Speed reducer Direction change Combine/split path	Change drive direction Intermediate gearboxes Crown/collector gear
Face	Intersecting	Speed reducer Combine/split paths 90° direction change	High gear ratio Crown/collector gear

Table 5: Gear Types and Applications

³¹ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-5.

Spur Gears in Helicopter Transmissions

The simplest and "most commonly used" gear in helicopter transmissions is the spur gear.³² The spur gear has straight teeth and transmits torque between parallel axis (Figure 8). Spur gears tend to have lower contact ratios than other types of gears and, therefore, generate more noise. The advantage of spur gears is that they do not develop axial loads or thrust like helical or bevel gears. This eliminates the need for thrust bearings and permits a lighter gearbox weight. Although not traditionally a high-speed gear, aerospace spur gears can operate at high pitch velocities up to 20,000 feet per minute³³ but do generate significant noise. The lack of an axial load makes spur gears well suited for planetary configurations and where permitted.

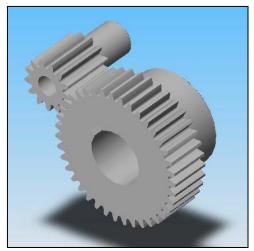


Figure 13: Example Spur Gear

 ³² Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-5.
 ³³ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-5.

Helical Gears in Helicopter Transmissions

Helical gears have teeth angled or twisted to the axis of rotation. This helix angle, ψ , varies from 15 to 30 degrees and generates radial and axial loads on associated bearings. Spur gears are a special form of helical gear with a 0° helix angle.

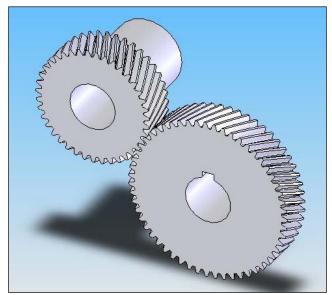


Figure 14: Example Helical Gear

Relative to a spur gear, the effective face width and line of contact in a helical mesh is longer due to the angled nature of the tooth face as shown in Figure 15. This generates higher contact ratios than spur gears and improves load sharing. This improved load sharing permits smoother meshing, reduces noise, and better handles higher speeds and horsepower.

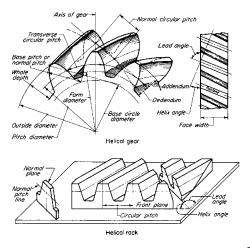


Figure 15: Helical Gear Terminology³⁴

Despite the higher load capacity, the Helicopter Engineering Preliminary Design

Handbook cautions against the employment of helical gears in helicopter transmissions:

"For the same face width, helical gears have more load carrying capacity than spur gears of equal size, are quieter, and have approximately the same efficiency. The overall design is not necessarily lighter; however, because the effect of thrust upon the mounting bearings must be considered. In general, helical gears designed for helicopters use do not offer a tremendous advantage over spur gears of the same size."³⁵

Bevel Gears in Helicopter Transmissions

Bevel gears are the primary means to change direction between intersecting axis.

The shaft angle created between the intersecting axis is typically between 0° and 115°,

with 90° being the most common angle. Straight bevel gears have radial teeth while

spiral bevel gears have curved teeth. Figure 16 shows the bevel gear terminology.

³⁵ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-7.

³⁴ Figure extracted from Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 1.31.

Straight bevel gears are limited to pitch line velocity less than 1,000 fpm while spiral bevel gears operate beyond 30,000 fpm.³⁶ Due to the restrictive pitch line velocity, straight bevel gears are generally not feasible for primary power paths in helicopter drive systems. Compared to straight bevel gears, spiral bevel gears form a contact area that permits smaller gear pitch for the same contact stress. This allows the use of coarser gears (lower diametral pitch) to increase bending strength.³⁷ The three dimensional curve of spiral bevel gears creates three-dimensional loads requiring multiple bearing restraints. Other forms of bevel gears such as Zerol are possible, but the high speed, high power applications found in helicopter drives best suit spiral bevel gears.

 ³⁶ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-8.
 ³⁷ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-8.

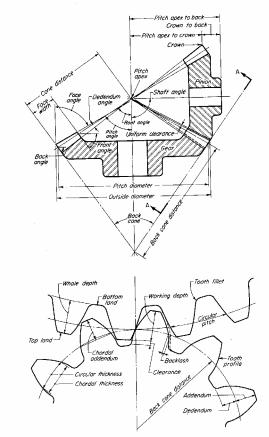


Figure 16: Bevel Gear Terminology³⁸

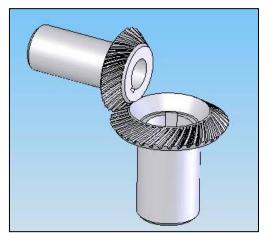


Figure 17: Example Spiral Bevel Gear

³⁸ Figure extracted from Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 134.

Face Gears in Helicopter Transmissions

Face gears have the teeth cut on top of gear instead of at the outer edge. The face gear tooth changes shape radially. The outside end of the gear limits the outer edge thickness and the pointed tooth limits the inner thickness. They are most similar to straight bevel gears but mate with a standard spur pinion. Gear and pinion have intersecting axis that normally creates a 90° shaft angle. Figure 18 shows the basic geometry for a face gear. Figure 19 shows a face gear-spur pinion mesh.

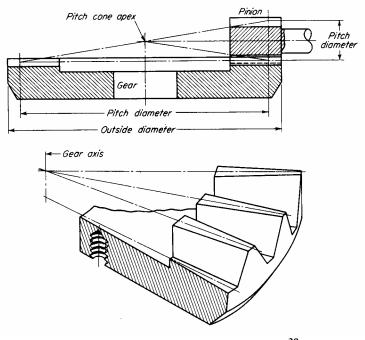


Figure 18: Face Gear Terminology³⁹

³⁹ Figure extracted from Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 1.40.



Figure 19: Example Face Gear⁴⁰

In 2004, the Rotorcraft Drive System for the 21st Century (RDS-21) demonstrated that a 5,100 hp face gear, split torque gearbox potentially increases horsepower to weight ratio by 35%, reduces noise by 12 dB, and reduces cost by 20%.⁴¹ Unfortunately, no standardized formulas to rate face gear stresses exist. All current face gear design accomplished with the partial use of Finite Element Analysis (FEA).⁴² The level of design involved with FEA is beyond the scope of this thesis and; therefore, face gears cannot be considered for this model's split-torque transmission. Dudley (1994) does suggest a possible means to preliminary estimate of face gear size. According to Dudley:⁴³

> "Face gears may be handled somewhat similarly to straight bevel gearsets. Generally it will be necessary to use less face width for the face gear than would be allowed as a maximum for the same ratio of bevel gears."

⁴⁰ NASA Research and Technology website, http://www.grc.nasa.gov/WWW/RT1995/2000/2730h.htm, November 1, 2005.

⁴¹ Yuriy Gmirya, et al, "Design and Analysis of 5100 HP RDS-21 Demonstrator Gearbox" 60th Annual Forum Proceedings, vol 2, (Alexandria: AHS International, 2004), 1221.

⁴² F.L. Litvin et al, NASA/CR-2000-209909 Handbook on Face Gear Drives With a Spur Involute Pinion (NASA, March 2000), 48. ⁴³ Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 2.51.

In the future, as face gear analyses improves and the gearing community accepts standards, face gears will likely be the gear of the choice for high reduction ratio transmissions found in heavy lift helicopters.

Rating Spur and Helical Gears

In preliminary design, prior to Finite Element Analysis, the rating methods used by most helicopter gear design engineers are from the American Gear Manufacturers Association (AGMA) standards.⁴⁴ Spur gears are a special case of helical gears where the helix angle, ψ , is zero. As such, the bending strength, compressive strength, and scuffing hazard calculations for both spur and helical gears are found using helical gear calculations. Equating ψ to zero reduces the helical equations to the set of spur equations.

The following force analysis, bending strength analysis, compressive strength analysis, and scuffing risk judgment are derived from Standard ANSI/AGMA 2001-C95 *Fundamental Rating Factor and Calculation Methods for Involute Spur and Helical Gear Teeth*. Additionally, much helicopter specific information and recommendations originates from AGMA 911-A94 *Information Sheet-Design Guidelines for Aerospace Gearing*.

⁴⁴ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-18.

Spur Gear Force Analysis

In general, the stresses on the gear tooth are a function of the transmitted tooth load. In its simplest form, the transmitted load is:

$$W_t = \frac{T}{r} = \frac{2T}{d}$$

Equation 27

where

 W_t is the transmitted load on a tooth T is torque r is the pitch radius or d is the pitch diameter

The total force or load between the driving tooth and the driven tooth is not tangent to the contact point, but is instead along the line of action as measured by the pressure angle. The resultant, total load is typically broken into tangential and radial components as shown in Figure 20.

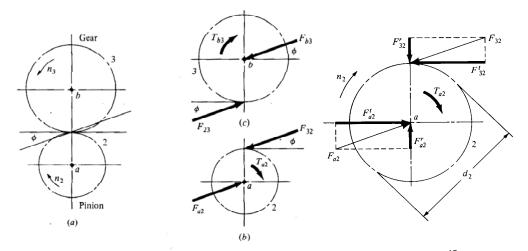


Figure 20: Free Body Diagram of a Simple Gear Train⁴⁵

⁴⁵ Figure extracted from Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 557.

In Figure 20, the total, resultant load on the larger gear is F_{23} while the load on the smaller pinion is F_{32} . The tangential component of F_{32} is F_{32}^{t} and the radial component is F_{32}^{r} . The reaction forces transferred to the gear are F_{a2}^{t} and F_{a2}^{r} . These reaction forces must be supported by bearings connected to the gearbox housing.

The tangential load serves to transfer the torque between the mating gears. The radial load "serves no useful purpose"⁴⁶ and must be reacted to by supporting bearings to hold the gear in mesh. The tangential and radial components of the total force, W, are:

 $W_r = W \sin \phi$ $W_t = W \cos \phi$

Equation 28

where

W is total force *W_r* is the radial force *W_t* is the tangential force or transmitted load φ is the pressure angle

The relationships between the transmitted load, torque, speed, and power are

shown in Equation 29 and Equation 30:

$$HP = \frac{W_t PLV}{(550)(60)} = \frac{2\pi nT}{(550)(60)(12)}$$

Equation 29

⁴⁶ Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 557.

$$T = \frac{dW_t}{2} = \frac{HP(550)(60)(12)}{2\pi n}$$

where

HP is power (hp) *PLV* is pitch line velocity (fpm) *T* is torque (in lb) W_t is transmitted load (lb) *n* is rotational speed (rpm) *d* is pitch diameter (in)

Helical Gear Force Analysis

The axial and component forces mentioned by the Preliminary Design Handbook

are drawn in Figure 21 and calculated with:

$$W_r = W \sin \phi_n$$
$$W_t = W \cos \phi_n \cos \psi$$
$$W_a = W \cos \phi_n \sin \psi$$

Equation 31

where

W is total force *W_r* is the radial force *W_t* is the tangential force or transmitted load *W_a* is the axial force or thrust load φ_n is the normal pressure angle ψ is the helix angle

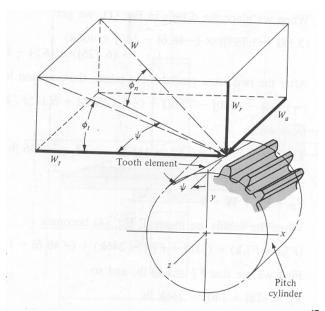


Figure 21: Helical Gear Free Body Diagram⁴⁷

Contact Stress

The fundamental formula for compressive or contact stress is:

$$s_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{dF}} \frac{C_f}{I}$$

Equation 32

where

 s_c is contact stress (lb/in²) C_p is elastic coefficient (lb/in²)^{0.5} W_t is transmitted tangential load (lb) K_o is overload factor K_v is dynamic factor K_s is size factor K_m is load distribution factor C_f is surface condition factor for pitting resistance F is net face width (in) I is geometry factor for pitting resistance

d is operating pitch diameter of pinion (in)

⁴⁷ Figure extracted from Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 562.

For a complete detailing of each factor, refer to ANSI/AGMA 2001-C95. A sample calculation of a planetary spur gear is shown in Table 30. A brief outline of the contact stress equation is listed below as it pertains to helicopter gears.

Elastic Coefficient, C_p

The elastic coefficient is a function of the material's Poisson's ratio and modulus of elasticity. Values for steel are approximately $2300 (lb/in^2)^{0.5}$. The elastic coefficient is:

$$C_{p} = \sqrt{\frac{1}{\pi \left[\left(\frac{1 - \mu_{p}^{2}}{E_{p}} \right) + \left(\frac{1 - \mu_{G}^{2}}{E_{G}} \right) \right]}}$$

Equation 33

where

 μ_P and μ_G is Poisson's ratio for pinion and gear, respectively E_P and E_G is Modulus of Elasticity for pinion and gear

Overload Factor, K_o

The overload factor accounts for externally applied loads beyond the tangential load. According to AGMA, "overload factors can only be established after considerable field experience."⁴⁸ In place of field experience, a preliminary value maybe located from Table 6. Considering turbines as uniform prime mover and the changing torque values of the rotors as light shock derives an overload factor of 1.25.

⁴⁸ AGMA Standard 2003-B97, *Rating the Pitting Resistance and Bending Strength of Generated Straight Bevel, Zerol Bevel and Spiral Bevel Gear Teeth.* (AGMA: Alexandria), 1997.

Character of	Character of load on driven machine				
prime mover	Uniform	Light shock	Medium shock	Heavy shock	
Uniform	1.00	1.25	1.50	1.75 or higher	
Light shock	1.10	1.35	1.60	1.85 or higher	
Medium shock	1.25	1.50	1.75	2.00 or higher	
Heavy shock	1.50	1.75	2.00	2.25 or higher	

 Table 6: Overload Factor Values⁴⁹

Dynamic Factor, K_v

The dynamic factor accounts for vibration of gear masses and resulting dynamic forces. K_v maybe approximated from Figure 22. Aerospace gears are usually precision ground gears with a high degree of accuracy. High accuracy gears are equivalent to AGMA Q_v 12 to 13.⁵⁰ A value of $Q_v = 12$ promotes successful operation at high pitch line velocities to and beyond 10,000 fpm. From Figure 22, this yields a value from 1.00 to 1.10.

⁴⁹ Extracted with permission from AGMA Standard 2003-B97, *Rating the Pitting Resistance and* Bending Strength of Generated Straight Bevel, Zerol Bevel and Spiral Bevel Gear Teeth. With the permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314., p. 35. ⁵⁰ Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 3.107.

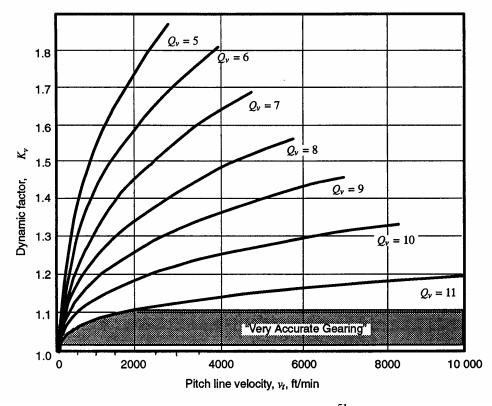


Figure 22: Dynamic Factor⁵¹

Size Factor, K_s

A size factor of unity is taken for properly heat-treated material.

Load Distribution Factor, K_m

The load distribution factor captures the impact of non-uniform load along the lines of contact. Several conditions influence the load distribution factor to include unmodified or properly modified leads, straddle mounted or overhung pinions, and adjusted gearing at assembly or improved compatibility by lapping. Values range from

⁵¹ Extracted with permission from AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*. With the permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314., p. 14.

1.10 to 1.20 for this design. Calculations for the load distribution factor may be found in Table 30.

Surface Condition Factor, C_f

The surface condition factor is unity for properly ground gears.

Geometry Factor, I

The geometry factor, I, measures the radii of curvature of contacting tooth profiles in order to evaluate the Hertzian contact stress in the tooth. The equation for the geometry factor is:

$$I = \begin{cases} \frac{\cos \phi_t \sin \phi_t}{2m_n} & \frac{m_G}{m_G + 1} \\ \frac{\cos \phi_t \sin \phi_t}{2m_n} & \frac{m_G}{m_G - 1} \end{cases}$$
 external gears

Equation 34

where

 φ_t is the transverse pressure angle m_N is load sharing ratio (1 for spur gears) m_G is the gear ratio

Allowable Contact Stress

The allowable contact stress adjusted for stress cycles, temperature effects, and

reliability must be greater than the calculated contact stress. The relationship between

contact stress and the allowable contact stress is:

$$s_c \le \frac{s_{ac}}{S_H} \frac{Z_N}{K_T} \frac{C_H}{K_R}$$

where

 s_{ac} is allowable contact stress (lb/in2) Z_N is stress cycle factor C_H is hardness ratio factor S_H is safety factor K_T is temperature factor K_R is reliability factor

Stress Cycle Factor, Z_N

Allowable contact stress for a given material is rated to 10^7 load cycles yet most aerospace gears often must last 10^9 cycles. The stress cycle factor adjusts the material contact stress for the required increased in fatigue life. The number of required cycles is:

$$N_{cvcles} = 60Lnq$$

Equation 35

where

 N_{cyc} is the number of stress cycles L is required life (hours) n is speed (rpm) q is number of contacts per revolution

A typical gear pair only experiences one contact per revolution; however, if a gear mates with q other gears it will experience q contacts per revolution. This often occurs in accessory drives and especially in planetary gearboxes. The stress cycle factor for pitting resistance can be found in Figure 23. For cycles beyond 10^7 , Z_N was assumed to be the mean of the shaded value.

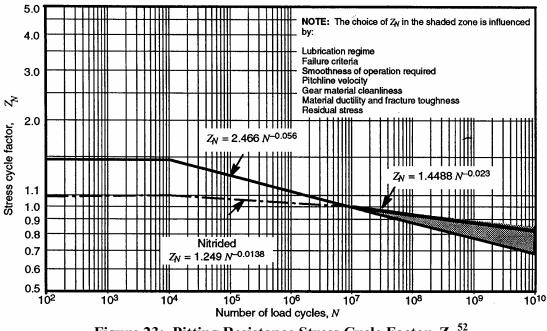


Figure 23: Pitting Resistance Stress Cycle Factor, Z_N⁵²

Hardness Ratio Factor, C_H

Gear capacity increases when the pinion is substantially harder than the gear.

This factor applies to the gear not the pinion.

$$C_H = 1.0 + A(m_G - 1.0)$$

Equation 36

where

$$4 = 0.00898 \left[\frac{H_{BP}}{H_{BG}} \right] - 0.00829$$

 H_{BP} is pinion Brinell hardness number H_{BG} is gear Brinell hardness number For $H_{BP}/H_{BG} < 1.2$, A = 0 For $H_{BP}/H_{BG} > 1.7$, A = 0.00698

⁵² Extracted with permission from AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*. With the permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314., p. 37.

Safety Factor S_H

The safety factor is added to include uncertainties in design analysis, material characteristics and manufacturing tolerances as well as human and economic risk. For the purposes of preliminary design, measuring these factors is beyond the scope of study and a value of unity is used.

Temperature Factor, K_T

For gear blank temperatures below 250°F, the temperature factor is unity. For gear blank temperatures above 250°F, values above 1.0 should be used. To maintain gear strength, sufficient lubrication is included to ensure temperatures do not exceed the 250°F and a value of unity maybe used.

Reliability Factor, K_R

The reliability factor relates the statistic variations in material to a normal probability distribution. For this normal distribution, the reliability constant is:

$$R_{el} = 1 - nv$$

Equation 37

where

 R_{el} is the reliability constant

n is the number of standard deviations

v is the coefficient of variation (standard deviation/mean), v = 0.1 for spur and helical gears

"For highly reliable aerospace design, a reliability of 3 standard deviations has been used in the past (or 3σ)."⁵³ Three standard deviations (n = 3) yields a reliability of 0.99875 and a constant of $R_{el} = 0.7$. AGMA reliabilities on material properties are listed at a failure of 1 in 100 for a standard deviation of n = 2.326, and a $R_{99} = 0.7674$. The reliability factor is:

$$K_R = \frac{R_{99}}{R_{desired}}$$

Equation 38

where

 R_{99} is the reliability constant for 99% (0.7674) $R_{desired}$ is the reliability constant

For a 3σ aerospace design, K_R is 1.096. The spur-helical model shown in Table 30 of APPENDIX B: SPUR-HELICAL GEAR RATING CALCULATIONS permits user input for other reliabilities from 1 in 10 failures up to 1 in 10,000 failures. This is calculated through a numerical approximation of:

$$R_{el} = \int_{-\infty}^{n} \frac{e^{-\frac{n^2}{2}}}{\sqrt{2\pi}} dn$$

Equation 39

⁵³ American Gear manufacturers Association, *AGMA 911-A94, Information Sheet-Design Guidelines* for Aerospace Gearing (Alexandria: AGMA, 1994), 44.

Bending Stress

The fundamental formula for bending stress is:

$$s_t = W_t K_o K_v K_s \frac{P_d}{F} \frac{K_m K_B}{J}$$

Equation 40

where

s_t is bending stress number (lb/in²) *K_B* is rim thickness factor *J* is geometry factor for bending strength *P_d* is transverse diametral pitch (in⁻¹) *P_d* is P_{nd} for spur gears $P_d = \frac{\pi}{p_x \tan \psi_s} = P_{nd} \cos \psi_s \text{ for helical gears}$ *P_{nd}* is normal diametral pitch (in⁻¹) *p_x* is axial pitch (in) *ψ_s* is helix angle at standard pitch diameter

For a complete detailing of each factor, refer to ANSI/AGMA 2001-C95. A complete detailed calculation of a planetary spur gear is in Table 30. A brief outline of the bending stress equation is listed below as it pertains to helicopter gears. Values for W_t , K_o , K_v , K_s , F, and K_m are the same as those for the contact stress formula Equation 32.

Rim Thickness Factor, K_B

The rim thickness factor is a stress concentration factor when the rim thickness cannot fully support the tooth root. All gears in this analysis are designed to ensure sufficient tooth support ($K_B = 1$).

Geometry Factor, J

The bending strength geometry factor accounts for stress concentration from tooth bending and compression from the radial load. For calculations of spur gear geometry factor consult AGMA 911-A94, *Information Sheet-Design Guidelines for Aerospace Gearing*. The guideline includes the geometry factor for both internal and external gears. This method finds the geometry factor, J, by calculating the value at discrete intervals along the edge, A, of the tooth. The point of lowest J value occurs where the bending stress is at its maximum. Associated calculations are in Table 34 and Table 35 of APPENDIX B: SPUR-HELICAL GEAR RATING CALCULATIONS. This is graphically shown in Figure 24.

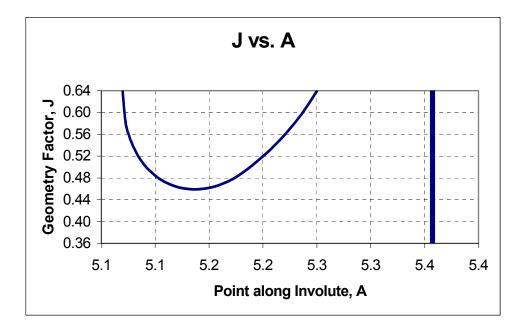


Figure 24: Minimum J Along the Involute Profile

For helical gears, the solution becomes lengthy and is beyond the level of detail required for a preliminary design. Typically, J for helical gears is found in a series of tables. To automate this table look up, the geometry factor for helical gears is estimated by means of least squares regression from the I and J Factor Tables found in AGMA Standard 908-B89, INFORMATION SHEET, *Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical, and Herringbone Gear Teeth*.⁵⁴ The estimated J factor lies within ± 5 percent of actual values. Figure 25 and Figure 26 are replicas of graphs shown in Shigley and Mischke (1989) and show graphically J for a standard pressure angle of 20°.

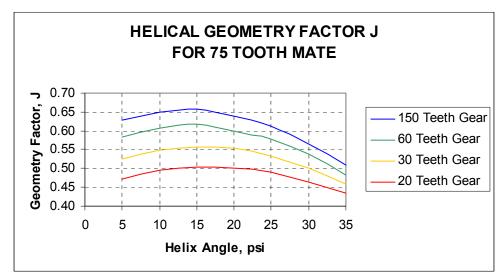


Figure 25: Geometry Factor for Helical Gears

⁵⁴ American Gear Manufacturers Association, *AGMA Standard 908-B89, Information Sheet, Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth* (Alexandria: AGMA, 1989).

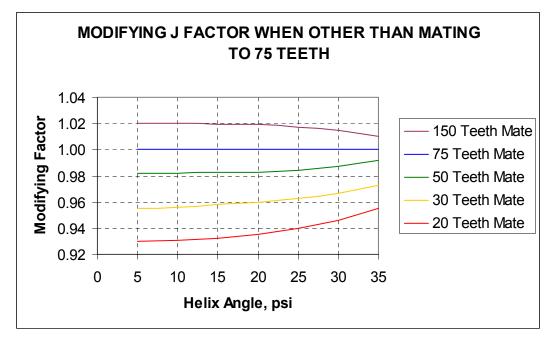


Figure 26: Modifying Factor for J for Different Mating Gears

Allowable Bending Stress

The allowable bending stress (adjusted for stress cycles, temperature effects, and reliability) must be greater than the calculated bending stress. The correlation between bending stress to allowable bending stress is:

$$S_t \le \frac{S_{at}Y_N}{S_F K_T K_R}$$

Equation 41

where

 s_{at} is allowable bending stress (lb/in²) Y_N is stress cycle factor S_F is safety factor

The values for K_T and K_R are the same as those used in section Allowable Contact Stress. Safety factor, S_F , is evaluated in the same method as S_H .

Stress Cycle Factor, Y_N

The stress cycle factor for bending strength is found similarly as the stress cycle factor for pitting resistance. Cycles are calculated from Equation 35 in section Allowable Contact Stress. Figure 27 applies to the bending stress cycle factor.

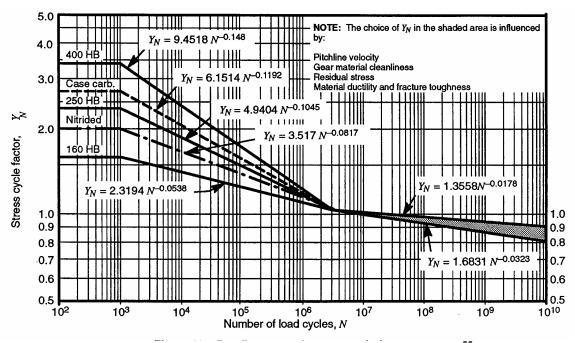


Figure 27: Bending Strength Stress Cycle Factor, Y_N⁵⁵

Spur and Helical Gear Materials

High performance aerospace gears require high tensile strength to endure high bending stresses and high surface hardness to resist pitting from high contact stresses.

⁵⁵ Extracted with permission from AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*. With the permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314., p. 37.

The majority of helicopter gearing are produced from "heat treated alloy steels and surface hardened."⁵⁶ Aerospace gears are carburized and case hardened to increase the surface hardness. For alloy steel processing, gas carbon is infused into the steel at high temperature, cooled slowly, and repeated at specified levels and temperatures. The carburizing and hardening cycles "produce a very hard, martensitic layer on the surface with a less hard, tough core."⁵⁷ Five different high strength steels are listed in Table 7. Typical applications and relevant material properties are included.

Description	Units	AISI	VASCO	PYRO-	CBS	AISI
-		9310	X2M	WEAR 53	600	4340
AMS Spec		6265/		6308	6255	6414
-		6260				
Heat Treatment		C-H	С-Н	C-H	C-H	TH-N
Main drive		Х	X	Х		
Application		Λ	Λ	Λ		
Accessory		Х				X
application		Λ				Λ
High Temp.			Х	Х	Х	
Application			Λ	A	<u></u>	
Case Hardness	HRC	61	62	62	60	50.5
Core Hardness	HRC	37	40	40	38	31
Brinell	BH	632	647	647	617	488
Hardness						
Allowable	psi	244,897	250,145	250,145	239,736	195,086
Contact Stress						
Allowable	psi	52,102	51,990	51,990	52,149	49,966
Bending Stress						
Poisson's Ratio		0.292	0.300	0.292	0.296	0.82
Modulus of		29.00E6	29.64E6	30.00E6	29.30E6	29.50E6
Elasticity						
Density (weight)	lb/in3	0.283	0.280	0.282	0.282	0.283

 Table 7: Spur-Helical Gear Steels

C-H is Case Hardened. TH-N is Through Harden and Nitride.

 ⁵⁶ American Gear manufacturers Association, AGMA 911-A94, Information Sheet-Design Guidelines for Aerospace Gearing (Alexandria: AGMA, 1994), 41.
 ⁵⁷ American Gear manufacturers Association, AGMA 911-A94, Information Sheet-Design Guidelines

³⁷ American Gear manufacturers Association, AGMA 911-A94, Information Sheet-Design Guidelines for Aerospace Gearing (Alexandria: AGMA, 1994), 41.

The bending fatigue strength and compressive strength of case hardened steels (AISI 9310, VASCO X2M, PYROWEAR 53, CBS600) are generally the same; however VASCO X2M and PYROWEAR 53 also have high temperature capability to resist scuffing. This makes these two carburized steels the best choice for the scuffing critical main gearbox. In addition to VASCO X2M's high strength, high hardness, high temperature capability, it is also the lightest of the steels. Due to these advantages, the power gears were evaluated using VASCO-X2M. The spur-helical model shown in Table 33 of APPENDIX B: SPUR-HELICAL GEAR RATING CALCULATIONS includes user options to select any of the six listed steels. In accordance with the Preliminary Design Handbook for Helicopter Engineering, "primary gear drives should be made from electrode vacuum melt (CEVM) processed steel.⁵⁸

Scuffing Hazard

Scuffing (scoring) failure occurs when the mating gear welds and tears the metal surface of the mated gear. Scuffing risk is a function of oil viscosity, operating bulk temperature of gear blanks, sliding velocity, surface roughness, gear materials, and surface pressure.⁵⁹ Following Blok's contact temperature theory as outlined by AGMA, scuffing will occur when the maximum contact temperature (t_{cmax}) exceeds a critical

⁵⁸ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u>

Handbook: Helicopter Engineering (Part One: Preliminary Design) (Alexandria: GPO, 1974), 7-2. ⁵⁹ American Gear Manufacturers Association, *AGMA Standard 2001-C95, Fundamental Rating Factors and Calculation Method for Involute Spur and Helical Gear Teeth* (Alexandria: AGMA, 1995), 9.

temperature known as the scuffing temperature (t_S) .⁶⁰ The contact temperature is the sum of the flash temperature (t_{fl}) and the bulk oil temperature (t_M) .

$$t_c = t_M + t_f$$

Equation 42

where

 t_M is bulk temperature t_{fl} is the flash temperature

Figure 28 shows the variation of local temperature (y-axis) along the line of

contact (x-axis) where:

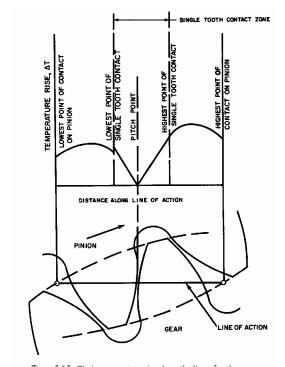


Figure 28: Contact Temperature Along the Line of Action⁶¹

⁶⁰ Blok, H. Les temperatures de Surface dans les Conditions de graissage Sans Pression Extreme, Second World Petroleum Congress, Paris, June, 1937 as outlined in AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.* (American Gear Manufacturers Association: Alexandria), 1995, Appendix A.

⁶¹ Figure extracted from Peter Lynwander, <u>Gear Drive Systems Design and Application</u> (New York: Marcel Dekker, 1983), 127.

The flash temperature is the local rise in temperature at a specified contact point along a tooth's line of action. Steps to find the flash temperature along the line of contact are detailed in Annex A of AGMA Standard 2001-C95, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth*. The full set of

calculations for the flash temperature is shown in Table 31: Spur-Helical Gear Scuffing. The fundamental formula for flash temperature is:

$$t_{fl} = K\mu_m \frac{X_{\Gamma} w_{Nr}}{B_M (b_H)^{0.5}} \left| (v_{r1})^{0.5} - (v_{r2})^{0.5} \right|$$

Equation 43

where

K is 0.80, numerical factor for frictional heat over the contact band μ_M is mean coefficient of fraction X_{Γ} is load sharing factor W_{Nr} is normal unit load v_{rl} is rolling velocity of pinion v_{r2} is rolling velocity of gear B_M is thermal contact coefficient b_H is semi-width of Hertzian contact band

Figure 29 graphs the results from the calculations in Equation 43. The maximum

flash temperature (t_{flmax}) used to measuring the scuffing risk is the highest temperature

along the line of contact. This compares favorably to Figure 28.

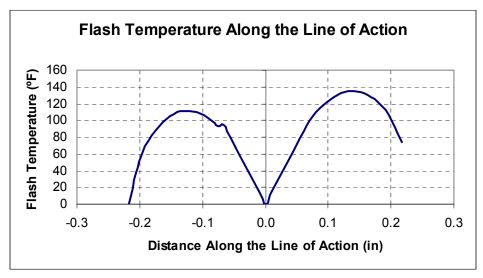


Figure 29: Calculated Flash Temperature Along the Line of Action

The oil temperature is taken to be the average of the cooler incoming oil and the outgoing hotter oil. The inlet temperature is assumed to be 125° F⁶² while the maximum rise (Δ T) is limited to 45° F.⁶³

$$t_{oil} = \frac{t_{oilin} + t_{oilout}}{2} = t_{oilin} + \frac{\Delta T}{2}$$

Equation 44

The bulk temperature maybe roughly approximated as:

$$t_M = -24 + 1.2t_{oil} + 0.56t_{fl \max}$$

Equation 45

where

t_{oil} is oil temperature (°F) t_{flmax} is maximum flash temperature

⁶² Peter Lynwander, <u>Gear Drive Systems Design and Application</u> (New York: Marcel Dekker, 1983), ⁶³ Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 3.129.

The maximum contact temperature is the sum bulk temperature and the maximum flash temperature.

$$t_{c \max} = t_M + t_{fl \max}$$

Equation 46

Helicopters mostly use synthetic oils to provide protection in a wider temperature range (approximately –50 to 400° F) than mineral oils. For MIL lubricants, the scuffing temperature is taken as a constant value with a normal distribution. Table 8 shows the MIL lubricants mean and standard deviation. The Boeing HLH aircraft had great success using MIL-L-23699 with VASCO X2M steel gears. For the lowest scuffing risk, the main gearbox shall use VASCO X2M with MIL-L-23699.

Table 8: MIL Lubricant Mean Scutting Temperatures						
Lubricant	Mean Scuffing	Standard Temperature				
	Temperature °F	Deviation °F				
MIL-7808 ⁶⁴	366	56.6				
MIL-L-6081 ⁶⁵	264	74.4				
(grade 1005)						
MIL-L-23699 ⁶⁶	391	58.65				
MIL-L-23699 with	459	31				
VASCO X2M ⁶⁷						

Table 8: MIL Lubricant Mean Scuffing Temperatures

For synthetic oils, the scuffing risk is the probability the maximum contact temperature exceeds the lubricant scuffing temperature. Probabilities for scuffing

⁶⁵ American Gear Manufacturers Association, *AGMA Standard 2001-C95, Fundamental Rating Factors and Calculation Method for Involute Spur and Helical Gear Teeth* (Alexandria: AGMA, 1995), 50.
 ⁶⁶ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u>

Handbook: Helicopter Engineering (Part One: Preliminary Design) (Alexandria: GPO, 1974).

⁶⁴ American Gear Manufacturers Association, *AGMA Standard 2001-C95, Fundamental Rating Factors and Calculation Method for Involute Spur and Helical Gear Teeth* (Alexandria: AGMA, 1995), 50.

⁶⁷ Mack, J.C., <u>USAAMRDL-TR-77-38</u>: <u>HLH Drive System</u> (Boeing Vertol: Philadelphia, 1977).

guidelines are mentioned in AGMA Standard 2001-C95.⁶⁸ Due to the critical nature aerospace gearing, only a low risk should be considered acceptable.

Table 9: Scuffing RiskProbability of ScuffingScuffing Risk<10%</td>Low10 to 30%Moderate>30%High

Rating Bevel Gears

Bevel gears stresses and allowables closely follow the procedure for spur and helical gears (section Rating Spur and Helical Gears, p. 49). The AGMA standard for bevel rating is ANSI/AGMA 2003-B97, *Rating the Pitting Resistance and Bending Strength of Generated Straight Bevel, Zerol Bevel, and Spiral Bevel Gear Teeth* and ANSI/AGMA 2005-C96, *Design Manual for Bevel Gears*. All formulas in the section are based upon AGMA 2003-B97. Bevel gear sample calculations are enclosed in Table 36 through Table 42 of APPENDIX C: BEVEL GEAR RATING CALCULATIONS.

Bevel Gear Force Analysis

A bevel gear creates a resultant force that has three components: tangential, radial, and axial loading as shown in Figure 30.

⁶⁸ American Gear Manufacturers Association, *AGMA Standard 2001-C95, Fundamental Rating Factors and Calculation Method for Involute Spur and Helical Gear Teeth* (Alexandria: AGMA, 1995), 50.

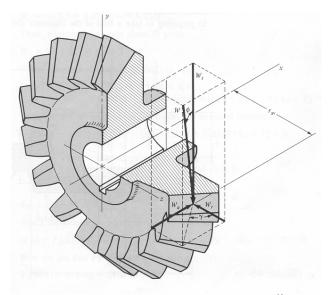


Figure 30: Bevel Gear Tooth Forces⁶⁹

The direction of the radial and axial forces depends upon the type of load face: concave or convex. Load face is a function of the hand of spiral, rotation of gear, and whether the gear drives or is driven. Table 10 lists the combinations for load face.

 Table 10: Bevel Gear Load Face⁷⁰

Pinion hand		Rotation Rotation		Load face	
of spiral	Gear hand of spiral	of driver	of driven	Driver	Driven
Right	Left	Clockwise	Counterclo	Convex	Concave
Right	Leit	Counterclo	Clockwise	Concave	Convex
Left	Right	Clockwise	Counterclo	Concave	Convex
Leit	Kight	Counterclo	Clockwise	Convex	Concave

 ⁶⁹ Figure extracted from Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 559.
 ⁷⁰ The majority of the information in this table is extracted from ANSI/AGMA 2005-C96, *Design*

⁷⁰ The majority of the information in this table is extracted from ANSI/AGMA 2005-C96, *Design Manual for Bevel Gears*, with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314.

The axial or thrust load is different for a concave and convex load face. A positive sign indicates a load away from the pitch apex. A negative sign indicates a load towards the pitch apex.

 $W_a = \frac{W_t}{\cos\psi} (\tan\phi\sin\gamma + \sin\psi\cos\gamma) \qquad \text{concave load face}$

 $W_a = \frac{W_t}{\cos\psi} (\tan\phi\sin\gamma - \sin\psi\cos\gamma) \qquad \text{convex load face}$

Equation 47

where

 W_a is the axial force or thrust load W_t is transmitted load ψ is helix angle φ is pressure angle γ is pitch angle for the gear of interest

Like the axial force, the radial force is different for each load face. A positive sign (separating force) indicates the direction of force is away from the mate. A negative sign (attracting force) indicates the direction of force is towards the mate.

$$W_r = \frac{W_t}{\cos\psi} (\tan\phi\cos\gamma - \sin\psi\sin\gamma) \qquad \text{concave load face}$$
$$W_r = \frac{W_t}{\cos\psi} (\tan\phi\cos\gamma + \sin\psi\sin\gamma) \qquad \text{convex load face}$$

Equation 48

where

 W_r is the axial force or thrust load W_t is transmitted load ψ is helix angle φ is pressure angle γ is the pitch angle for the gear of interest Force analysis for the bevel gear sizing tool is in Table 38 of APPENDIX C:

BEVEL GEAR RATING CALCULATIONS.

Contact Stress

The rating formula for bevel gear bending stress is:

$$s_c = C_p \sqrt{\frac{2T_p}{Fd^2 I} K_o K_v K_m C_s C_{xc}}$$

Equation 49

where

 s_c is calculated contact stress C_p is elastic coefficient C_s is size factor T_P is operating pinion torque (lb in) K_o is overload factor K_v is dynamic factor F is net face width d is pinion outer pitch diameter K_m is load distribution factor C_{xc} is crowning factor I is pitting resistance

Much of the rating calculations for bevel gears follow the rating procedure for spur or helical gears (see page 73). C_p is derived from Equation 33. K_o is from Table 6. K_v is from Figure 22. The factors C_s , K_m , C_{xc} , and I as they apply to aerospace gears are listed below.

Size Factor, Cs

The size factor for pitting resistance accounts for nonuniformity of material and is a function of face width:

$$C_s = 0.125F + 0.4375$$

where

 $0.5 \le F \le 4.5$ for F > 4.5, C_s = 1.0

Load Distribution Factor, K_m

The load distribution factor accounts for non-uniform loads across along the line

of contact. The factor is expressed in Equation 51. $K_{mb}\xspace$ is:

1.00 for both members straddle mounted1.10 for one member straddle mounted1.25 for neither member straddle mounted.

$$K_m = K_{mb} + 0.0036F^2$$

Equation 51

where

 K_{mb} is load distribution modifier F is net face width

Crowning Factor, C_{xc}

The crowning factor compensates for variation in the crowning during

manufacturing. Cxc is 1.50 for properly crowned teeth and 2.0 for larger non-crowned

teeth. A value of 1.50 has been used in the bevel gear calculations.

Geometry Factor, I

The geometry factor for pitting resistance is:

$$I = \frac{s\rho_o \cos\psi \cos\phi}{FdC_i m_{NI}} \frac{P_d}{P_m}$$

Geometry factor calculations for the bevel gear model are included in Table 42: Bevel Gear Pitting Resistance Geometry Factor of APPENDIX C: BEVEL GEAR RATING CALCULATIONS. The geometry factor, I, is found by assuming a critical loading point over the tooth line. The true critical point and I occur as the minimum from Figure 31.

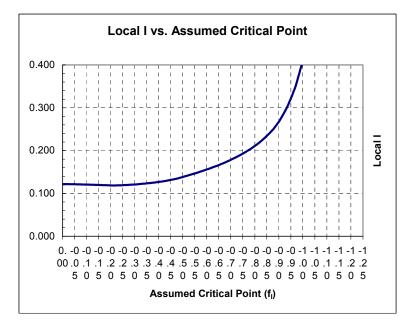


Figure 31: Finding the Bevel Gear Geometry Factor, I

Allowable Contact Stress

The relationship between contact stress and allowable contact stress is expressed in Equation 53. C_L is found from Figure 23. S_H has the same considerations as for spur and helical gears. Unity is used for S_H during preliminary analysis.

$$s_c \le \frac{s_{ac} C_L C_H}{S_H K_T C_R}$$

where

 s_{ac} is allowable contact stress C_L is stress cycle factor S_H is contact safety factor C_H is hardness ratio K_T is temperature factor C_R is reliability factor

Temperature Factor, K_T

For gear blank temperatures at or below 250° F, K_T is 1.0. For operations above

250° F, K_T is:

$$K_T = \frac{460 + T_T}{710}$$

Equation 54

where

T_T is the peak operating gear blank temperature

Reliability Factors, C_R

The reliability factor is approximated in the same way as spur and helical gears but v is 0.156.⁷¹ For bevel gears, Table 11 shows the factor for different requirements. An aerospace value of 1.09 has been applied to the bevel gear model. For this table, C_R is the square root of K_R.

⁷¹ American Gear manufacturers Association, *AGMA 911-A94, Information Sheet-Design Guidelines for Aerospace Gearing* (Alexandria: AGMA, 1994), 43.

Desired Rate of Failure	Probability	C _R	K _R
Fewer than one failure in 10,000	0.9999	1.22	1.50
Fewer than one failure in 1,000	0.999	1.12	1.25
Aerospace Design (3σ)	0.99875	1.09	1.20
Fewer than one failure in 100	0.99	1.00	1.00
Fewer than one failure in 10	0.9	0.92	0.85
Fewer than one failure in 2	0.5	0.84	0.70

Table 11: Bevel Gear Reliability Factors

Bending Stress

The formula for bending stress in a bevel gear is:

$$s_t = \frac{2T_P}{Fd} \frac{P_d K_o K_v}{1} \frac{K_s K_m}{K_v J}$$

Equation 55

where

 s_t is calculated bending stress K_o is overload factor K_v is dynamic factor P_d is outer transverse diametral pitch K_s is size factor K_m is load distribution factor K_x is tooth lengthwise curvature factor J is bending strength factor

 K_o is found as in spur and helical gears using Table 6. K_v is from Figure 22. K_m

is the same value as in pitting resistance (Equation 51).

Size Factor, Ks

Instead of being a function of face width like pitting resistance, the size factor for bending stress is a function of outer transverse pitch. For $16 \ge P_d \ge 0.5$, the range of aerospace gearing, the size factor is:

$$K_s = 0.4867 + \frac{0.2133}{P_d}$$

Equation 56

Lengthwise Curvature Factor, K_x

The lengthwise curvature factor is a function of spiral angle and tooth curvature. Table 38 and Table 39 of APPENDIX C: BEVEL GEAR RATING CALCULATIONS contain the relevant geometry and calculations for the factor.

Bending Strength Geometry Factor, J

Calculations for the bending strength geometry factor for bevel gears is complex and beyond the needs of a preliminary design. Often charts are used for faster reference of J in preliminary design. To permit automated estimations, the geometry factor charts are calculated from least squares regressions for the following values:

Straight bevel gears for	$\psi = 0^{\circ}, \phi = 20 - 25^{\circ}, \Sigma = 90^{\circ}$
Spiral bevel gears for	$\psi = 35^{\circ}, \phi = 20^{\circ}, \Sigma = 60 - 90^{\circ}$
	$\psi = 15 - 35^{\circ}, \phi = 20^{\circ}, \Sigma = 90^{\circ}$

Allowable Bending Stress

The relationship between bending stress and allowable bending stress is expressed in Equation 57. K_L is found with Figure 27. K_T estimated with Equation 54. The bevel gear calculations employ an aerospace reliability of 0.99875 (3 σ) which yields a K_R value of 1.20 (Table 11). The bending safety factor has the same considerations as spur and helical gears and is assumed to be unity for preliminary design.

$$s_t \le \frac{s_{ac} K_L}{S_F K_T K_R}$$

Equation 57

where

 s_{at} is allowable bending stress S_F is bending safety factor K_L is stress cycle factor K_T is temperature factor K_R is reliability factor

Bevel Gear Materials

Helicopter gearing demands the high strength, hardened steels that carburized, case hardened alloy steels provide (refer to Spur and Helical Gear Materials on page 66); however, for most helicopter drives, bevel gears are not used at the last few stages of the drive as the thrust loading becomes excessive. Operating earlier in the drive train, bevel gears tend to have higher speeds (rpm) and lower torque values. Steels listed in Table 12 are suggested steels for bevel gears with tested strengths and properties.⁷² All steels are recommended for accessory drives but only Grade 3, carburized and case hardened steels are strong enough for heavy lift main drive applications.

⁷² ANSI/AGMA 2003-B97, <u>Rating the Pitting Resistance and Bending Strength of Generated Straight</u> <u>Bevel, Zerol Bevel and Spiral Bevel Gear Teeth</u>. (AGMA: Alexandria), 1997, p. 24.

Description	Units	Steel TH	Steel	Steel	Steel	Nitralloy
Description	Units	Sterin	С-Н	С-Н	С-Н	135M
		(Grade 2)	(Grade 1)	(Grade 2)	(Grade 3)	(Grade 2)
AMS Spec		AGMA				
1		Class 5				
Heat Treatment		TH	С-Н	C-H	C-H	Nitrided
Main drive					Х	
Application					Λ	
Accessory		Х	Х	Х	Х	Х
application		Λ	Λ	Λ	Λ	Λ
Case Hardness	HRC	43	59.5	61	61	60
Core Hardness	HRC		21	25	30	
Brinell	BH	400	610	632	632	614
Hardness						
Allowable	psi	175,000	200,000	225,000	250,000	145,000
Contact Stress						
Allowable	psi	25,180	30,000	35,000	40,000	24,000
Bending Stress						
Poisson's Ratio		0.300	0.300	0.300	0.300	0.291
Modulus of		30E6	30E6	30E6	30E6	30E6
Elasticity						
Density (weight)	lb/in3	0.282	0.282	0.282	0.282	0.280

Table 12: Spur-Helical Gear Steels

C-H is Case Hardened. TH-N is Through Harden and Nitride.

Scuffing Hazard

Bevel gear scuffing hazard is not easily calculated and usually found through testing. The bevel gear rating model, due to these reasons, omits scuffing hazard assessment for bevel gears.

SHAFTING

Simplified Shafting Model

Transmission shafting experiences three-dimensional loads in the x, y, and zdirections; undergoing a combination of bending loads, torsional loads, and axial tension or compression loads. The Preliminary Design Handbook for Helicopter Engineering suggest that even though "a complete analysis normally would not be performed during the preliminary design phase, the principal static and dynamic loads should be analyzed sufficiently to insure structural integrity with the selected size, weight, and performance of the drive."⁷³ Capturing gear shaft loading, to include aerodynamic loads, can quickly become one of the "most complex loading conditions of the drive system."⁷⁴ To estimate the loads and moments on shafts in the preliminary design phase, a simplified loading model for the y and z-directions is created (the x-direction contains the axial loads and does not impact shear or bending moments). Figure 32 shows a loading diagram in the ydirection.

 ⁷³ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 4-70.
 ⁷⁴ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 4-74.

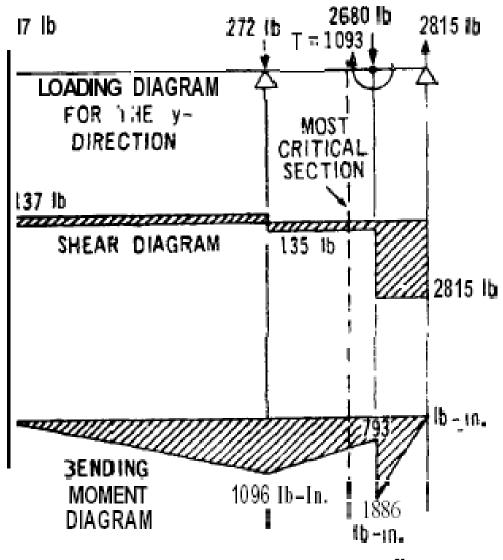


Figure 32: Loading Diagram for y-Direction⁷⁵

⁷⁵ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u>. (Alexandria: GPO, 1974), 4-75.

The sample of the shafting tool shown in APPENDIX C: SHAFT DESIGN

CALCULATIONS contains a simple bearing-gear (or loading)-bearing arrangement similar to Figure 33. The example shown in the figure has no moment about the z-axis and a significant moment (-275,000 in lb) moment about the y-axis. Other configurations are possible: the example shown in Figure 32 contains a bearing-gear-gear-bearing arrangement (BGGB). Within the shafting model, the user may select any of the following arrangements:

1. BGB

- 2. BGBG
- 3. GBGB
- 4. GBBG
- 5. BGGB

where

B is support bearing G is splined gear or any applied load or torsion

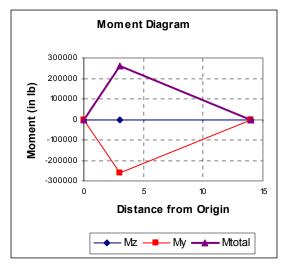


Figure 33: Sample Shaft Moment Diagram

Margin of Safety

Total moment on the shaft does not have a steady direction with respect to the shaft because the shaft is rotating. This loading is "of a vibratory nature" ⁷⁶ and requires an interaction equation to account for vibratory bending stress, axial tension stress, and torsion shear stress. The Preliminary Design Handbook for Helicopter Engineering suggest a margin of safety based upon the maximum shear theory of failure:⁷⁷

$$MS = \frac{1}{\sqrt{\left(\frac{f_b}{F_{en}} + \frac{f_a}{F_{ty}}\right)^2 + 4\left(\frac{f_s}{F_{sy}}\right)^2}}$$

Equation 58

where

MS is margin of safety f_a is axial tension stress, psi f_b is vibratory bending stress, psi f_s is torsional shear stress, psi F_{en} is endurance limit stress, psi F_{sy} is shear yield stress, psi F_{tv} is tensile yield stress, psi

The maximum shear theory accounts for the fact that shaft failures normally occur from fatigue loading.⁷⁸ To save weight, transmission shafting is usually hollow with as high of a diameter to thickness ratio as possible.⁷⁹ The shafting tool in APPENDIX C: SHAFT DESIGN CALCULATIONS iterates the Margin of Safety by varying the shafts

Handbook: Helicopter Engineering (Part One: Preliminary Design) (Alexandria: GPO, 1974), 7-3. ⁷⁷ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u>

⁷⁶ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u>

Handbook: Helicopter Engineering (Part One: Preliminary Design) (Alexandria: GPO, 1974), 7-3. ⁷⁸ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u>

Handbook: Helicopter Engineering (Part One: Preliminary Design) (Alexandria: GPO, 1974), 4-75. ⁷⁹ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> Handbook: Helicopter Engineering (Part One: Preliminary Design). (Alexandria: GPO, 1974), 7-2.

diameter and thickness. The output recommends the lightest shaft that meets the user specified minimum Margin of Safety.

Vibratory Bending Stress

From the maximum shear theory, the alternating bending stress is:

$$f_b = \frac{Mc}{I}$$

Equation 59⁸⁰

where

M is the total bending moment c is the radius where stress is calculated I is the moment of inertia for a hollow shaft $I = \frac{\pi (D^4 - d^4)}{64}$

The maximum bending stress will occur at the outer diameter where product of

the moment and a stress concentration factor, K_b, are greatest:

$$f_{b\max} = \frac{32K_b M_{\max} D}{\pi \left(D^4 - d^4\right)}$$

Equation 60

⁸⁰ Peter Lynwander, <u>Gear Drive Systems Design and Application</u> (New York: Marcel Dekker, 1983), 137.

According to Mischke, the endurance limit or fatigue limit of steels maybe approximated as:

$$F_{en} = \begin{cases} 0.504S_{ut} & S_{ut} < 200 \text{ kpsi} \\\\ 100 \text{ kpsi} & S_{ut} > 200 \text{ kpsi} \end{cases}$$

Equation 61⁸¹

where

 S_{ut} is the ultimate tensile strength

Axial Tension Stress

On helicopter shafting, axial tension or compression stress is the axial loading from helical or bevel gears and also from aerodynamic loads (thrust) on the main rotor or tail rotor drive shafts. Axial stress is the force to area ratio and is expressed as:

$$f_a = \frac{F}{A} = \frac{4F}{\pi (D^2 - d^2)}$$

Equation 62

where

 f_a is axial stress A is cross sectional area F is the axial load D is the outer shaft diameter d is the inner shaft diameter (d = 0 for a solid shaft; D – d = thickness)

⁸¹ Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 278.

The tensile yield stress, F_{ty} , is also referred to as the yield strength and is an easily referenced material property.

Torsional shear stress

Engine drive and tail rotor drive shafting primarily carry torsional loads.⁸² For the simple model, the torsional shear stress is assumed steady and equal to:

$$f_s = \frac{Tr}{J}$$

Equation 63⁸³

where

fs is torsional shear stress T is torque r is radial distance of the desired stress point J is the polar moment of inertia $J = \frac{\pi (D^4 - d^4)}{32}$

The maximum shear stress will occur along the outside diameter at the greatest

value of the product of the stress concentration factor K_s, and the local moment:

$$f_{s\max} = \frac{16TK_sD}{\pi \left(D^4 - d^4\right)}$$

Equation 64

 ⁸² Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-3.
 ⁸³ Peter Lynwander, <u>Gear Drive Systems Design and Application</u> (New York: Marcel Dekker, 1983), 138.

Most often, the critical area on a shaft occurs at a sharp narrowing of the shaft diameter. This increased stress is captured with a stress concentration factor, K_s.

For the maximum-shear stress theory, the endurance limit stress is one-half the yield strength:⁸⁴

$$F_{sy} = 0.5F_{ty}$$

Equation 65

Critical Speeds

One of the greatest challenges to shaft design is the divergence phenomenon nicked named "critical speeds." As shaft speed increase, residual unbalances create large centrifugal forces. The centrifugal forces bend the rotating shaft and are balanced by the elastic forces in the shaft. This balance is likened to a skipped rope.⁸⁵ As the shaft increases past this critical speed, the bent mass moves to the centerline until the shaft rotates about its axis.

Operation below the critical speed is called a subcritical condition while operation above the critical speed is termed supercritical. For safe operation, subcritical shafts must not operate within 30% of the critical speed. Supercritical shafts must not operate within 10% of the critical speed.⁸⁶ Drive shafts connecting gearboxes and engines

⁸⁴ Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 250.

 ⁸⁵ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-62.
 ⁸⁶ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-69.

usually operate below or between critical speeds.⁸⁷ If a shaft is designed to run above the critical speed, damping is required during run-up as the shaft passes through the avoid range.

Nonuniform Shafts

Nonuniform shafts are defined as having a nonuniform rigidity or concentrated masses such as splined gears. A variation on Rayleigh's method as outlined in the Preliminary Design Handbook for Helicopter Engineering derives the natural frequency by comparing kinetic energy with potential energy.⁸⁸ This is done by dividing the shaft into concentrated mass of m_i connected by massless, stiff shaft elements (Figure 34). The shaft deflection from a bending moment maybe calculated by numerically integrating:

$$y = \int \int \frac{M}{EI} dx dx$$

Equation 66

where

y is shaft deflection (in) M is bending moment E is modulus of elasticity I is moment of inertia

 ⁸⁷ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-3.
 ⁸⁸ Headquarters, U.S. Army Material Command, <u>AMC Pamphlet 706-201 Engineering Design</u> <u>Handbook: Helicopter Engineering (Part One: Preliminary Design)</u> (Alexandria: GPO, 1974), 7-63.

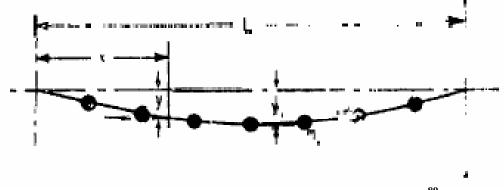


Figure 34: Mathematical Model of Nonuniform Shaft⁸⁹

As an example, Equation 66 is numerically integrated with Simpson's Rule in Table 47. Figure 35 shows the shaft bending and moment for the shaft example in Table 47.

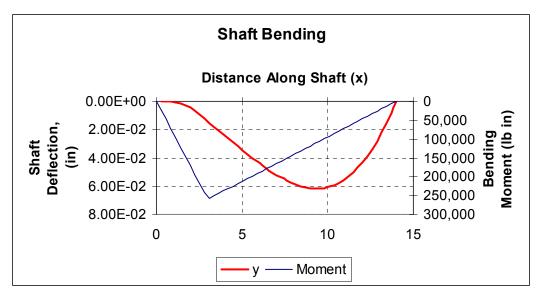


Figure 35: Nonuniform Shaft Bending

⁸⁹ Headquarters, U.S. Army Material Command. AMC Pamphlet 706-201 Engineering Design Handbook: Helicopter Engineering (Part One: Preliminary Design) (GPO: Alexandria, Virginia), August 1974, p. 7-64.

With the deflection known, the kinetic energy is:

$$T = \frac{\omega^2}{2} \sum_{i=1}^p \frac{w_i}{g} y_i^2$$

Equation 67

where

T is kinetic energy (in lb) ω is shaft frequency w_i are lumped weights g is 386.4 in/sec² y_i is element deflection

The potential energy is:

$$V = \frac{1}{2} \sum_{i=1}^{p} w_i y_i$$

Equation 68

where

V is potential energy (in lb) w_i are lumped weights y_i is element deflection

The natural frequency is found by equating potential energy to kinetic energy.

Expressing the natural frequency as a critical speed in rpm yields:

$$N_{c} = \frac{60}{2\pi} \sqrt{g} \sqrt{\frac{\sum_{i=1}^{p} w_{i} y_{i}}{\sum_{i=1}^{p} w_{i} y_{i}^{2}}}}$$

Equation 69

where

N_c is critical speed (rpm)

Uniform Shafts

For shafts with uniform mass distribution and with no moment restraint on the bearings, there will be no bending moment. This is common in the tail rotor drive shaft connecting the main transmission to the tail rotor gearbox. This type of shaft maybe modeled as a beam with length L where bearings support the ends. The critical speed for a uniform shaft is a "function of the mean radius of the tube and the length between supports."⁹⁰ The critical speed, N_c , is expressed in rpm as:

$$N_{c} = \frac{30\pi^{3}}{2\pi L^{2}} \sqrt{\frac{4gE\pi (D^{4} - d^{4})}{64\pi\rho (D^{2} - d^{2})}}$$

Equation 70

where

L is shaft length g is 386.4 in/sec^2 E is modulus of elasticity ρ is density (weight) D is outside diameter d is inside diameter

<u>Shaft Materials</u>

Lightweight metals such as aluminum alloys and titanium are preferred for shafts in helicopter drives systems. Great strength can be gained from high diameter to thickness ratios that permits the use of weight saving aluminum alloys. The Boeing HLH

⁹⁰ Headquarters, U.S. Army Material Command. AMC Pamphlet 706-201 Engineering Design Handbook: Helicopter Engineering (Part One: Preliminary Design) (GPO: Alexandria, Virginia), August 1974, p. 7-63.

employed aluminum alloy for most shafting except in the main rotor drive shaft. There, the stronger titanium was added to withstand high aerodynamic loads. The shafting model has several aluminum alloys, steal 4340 and the same titanium found in the HLH for user selection (Table 49). All designs presented in this thesis use Aluminum Alloy T7075 and Titanium Forging 6 A1-4V. Table 13 has representative samples from the material database.

		Aluminum	Steel AISI	Titanium Forging
Property	Units	Alloy T7075	4340	(6 A1-4V)
Ultimate tensile strength	psi	86,000	250,000	135,821
Yield tensile strength	psi	78,600	230,000	122,642
Shear yield stress	psi	39,300	115,000	61,321
Endurance strength	psi	20,000	100,000	20,000
Endurance limit	psi	14,280	71,400	14,280
Surface factor		1.00	1.00	1.00
Size factor		0.70	0.70	0.70
Load factor		1.00	1.00	1.00
Temperature factor		1.02	1.02	1.02
Miscellaneous effects		1.00	1.00	1.00
Total endurance factor		0.71	0.71	0.71
Density (weight)	lb/in ³	0.098	0.283	0.161
Modulus of Elasticity		10.3E6	30.0E6	15.5E6

Table 13: Shaft Material Properties

Endurance strength has been derated in accordance with Shigley and Mischke⁹¹ to

account for outside factors on the material strength (Equation 71). Values for each

⁹¹ Joseph Shigley and Charles Mischke, <u>Mechanical Engineering Design</u>, 5th ed (New York: McGraw-Hill, 1989), 283.

modifying factor are in Table 13. Total endurance factor k, is estimated as 0.71.

$$k = k_a k_b k_c k_d k_e$$

Equation 71

GEARBOX COOLING

Although gear efficiency is high, at the power levels of the JHL sizeable amounts of heat must be dissipated by the lubrications system. Mesh efficiencies maybe taken as:⁹²

HP:

- 1. 99.5% per spur mesh
- 2. 99.5% per helical mesh
- 3. 99.5% per bevel mesh
- 4. 99.25% per planetary set

Power loss from a gear mesh is a function of the efficiency, η , and applied power,

$HP_{loss} = (1 - \eta)HP$

Equation 72

Assuming any power loss is converted into heat by the meshing gears, the oil

temperature rise across the gearbox is:

$$\Delta T = \frac{Q}{MC_p}$$

Equation 73

where

 ΔT is temperature rise (°F)

⁹² Headquarters, U.S. Army Material Command. AMC Pamphlet 706-201 Engineering Design Handbook: Helicopter Engineering (Part One: Preliminary Design) (GPO: Alexandria, Virginia), August 1974, p. 7-6 to 7-12.

M is oil flow (lb/min) Q is heat generated (Btu/min) where Q = HP (42.4) Cp specific heat of oil ≈ 0.5 Btu/lb-°F

For example, in the sample spur gear calculations of Table 32 and summarized below, total oil flow required to maintain a ΔT of 30° F is 5.32 gallons.

Table 14: Sample Cooler Design				
Value	General Rule	Aerospace Limit		
HP	2,600 hp			
η	99.5%			
HP _{loss}	13.3 hp			
Q	563.9 Btu/min			
М	39.9 lb/min	25.1 lb/min		
	or	or		
	5.3 gal/min	3.3 gal/min		
T _{oilin}	125° F	125° F		
ΔT	30 ° F	45° F		
T _{oilout}	155° F	170° F		
T _{oilavg}	140° F	147.5° F		

Aerospace designs normally operate up to a maximum ΔT of 45° F.⁹³ The calculations used in the lubrication and cooling model utilize a ΔT of 45°F to determine oil flow required. A typical helicopter main gearbox lubrication system is shown in Figure 36.

⁹³ Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 3.129.

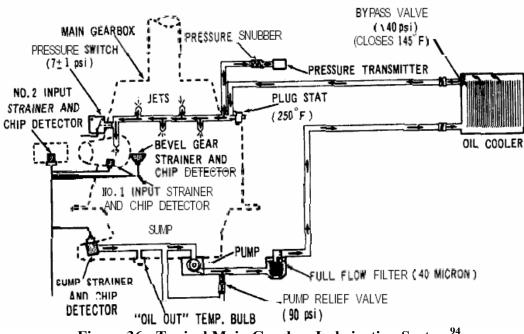


Figure 36: Typical Main Gearbox Lubrication System⁹⁴

⁹⁴ Headquarters, U.S. Army Material Command. AMC Pamphlet 706-201 Engineering Design Handbook: Helicopter Engineering (Part One: Preliminary Design) (GPO: Alexandria, Virginia), August 1974, p. 7-4.

TRADITIONAL PLANETARY MODEL

The traditional planetary drive system model consists of a 2-stage planetary main gear box where the sun gear of the first stage is the input and the carrier of the second stage is the output and transfers torque to the main rotor shaft. The main gearbox is similar to the Boeing prototype HLH's forward or aft transmissions. The tandem rotor HLH transmission had a 2-stage planetary gearset driving both the forward and aft rotors. A sketch of the HLH's transmission layout is shown in Figure 37 and the 2-stage planetary aft transmission is shown in Figure 38.

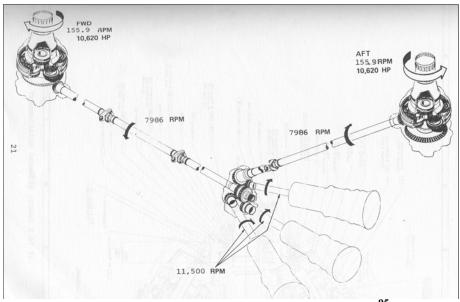


Figure 37: HLH Drive System Arrangement⁹⁵

⁹⁵ Mack, J.C., <u>USAAMRDL-TR-77-38: HLH Drive System</u> (Boeing Vertol: Philadelphia, 1977), p. 21.

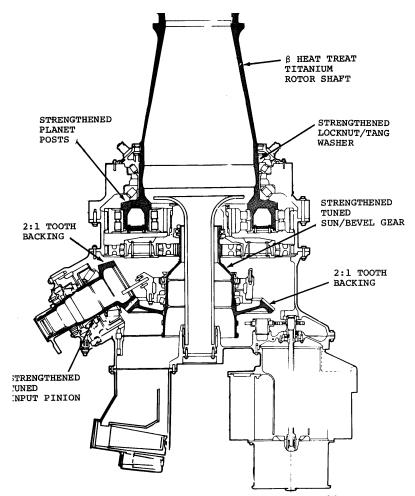


Figure 38: HLH Aft Transmission⁹⁶

⁹⁶ Mack, J.C., <u>USAAMRDL-TR-77-38: HLH Drive System</u> (Boeing Vertol: Philadelphia, 1977).219.

<u>Drive Arrangement</u>

The Supplemental Information defines the JHL baseline aircraft with 3 scaleable turbine engines, a single main rotor and traditional anti-torque tail rotor (Figure 4 and Figure 5). While the Boeing HLH was a tandem helicopter, the aft and front 2-stage planetary main gearbox was used as the model for the JHL planetary gearbox. Figure 39 shows the planetary drive system that fits within the requirements of the JHL Supplement.⁹⁷ Although there are many possible configurations that satisfy the JHL Supplement's requirements, this arrangement follows a traditional layout for helicopters.

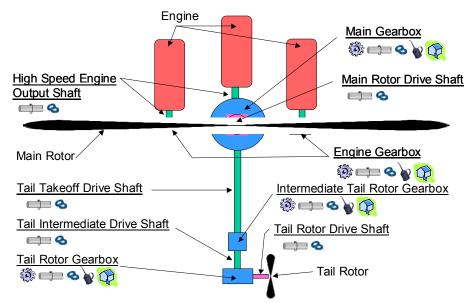


Figure 39: Drive System Components

⁹⁷ Aviation Applied Technology Directorate, "Joint Heavy Lift Supplemental Package" (Fort Eustis: AATD, November 4, 2004).

Engine Input Gearbox

Each turbine engine undergoes an initial reduction ratio at the Engine Input Gearbox. Inside the gearbox is a freewheeling unit to allow autorotation. The left and right Engine Input Gearbox have a pair of bevel gears to change the direction of the drive 90°. The bevel gear also mates with an accessory bevel gear to power the left or right accessory set. The center engine has a helical idler and helical gear to power the accessory set.

Bevel Crown

The bevel crown resides below the 2-stage planetary set and receives the power input from the Engine Gearbox Output Shaft. Three bevel pinions drive the Crown Bevel gear which drives the 1st stage sun gear.

Oil Cooler Gearbox

A tail takeoff bevel pinion mates with the crown gear and links an oil cooler located directly aft of the main gearbox. The oil cooler gearbox distributes power to the main oil cooler, an auxiliary accessory module, and the tail take off shaft.

Intermediate Tail Rotor Gearbox

The tail takeoff shaft has 12 equally spaced 45" segments linked down the tail boom of the aircraft. A bevel gear set conducts a 60° direction change to power the intermediate tail rotor shaft. The shaft connects to the tail rotor gearbox at the top of the vertical tail.

Tail Rotor Gearbox

The Tail Rotor Gearbox conducts a final speed reduction and a 90° direction change to drive the Tail Rotor Drive Shaft.

Planetary Main Gearbox

The planetary gearbox overall ratio is considered an independent variable; however, the individual reduction ratio of each planetary stage is minimized for weight according to Willis' "Lightest-weight gears" procedure for compound drives.⁹⁸ Figure 40 shows the weight minimization for a 2-stage planetary gearset.

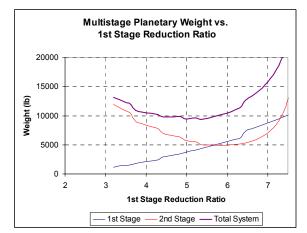


Figure 40: Example Optimized 2-Stage Planetary Gearbox

⁹⁸ R.J. Willis, "New Equations and Chart Pick Off Lightest-weight Gears," <u>Product Engineering</u> v. 34, n.s. 2 (January 21, 1963): 65.

For a given overall reduction ratio, the procedure assumes a first stage gear ratio. The corresponding second stage ratio is the quotient of the overall ratio to the 1st stage ratio.

$m_{G2} = M_G/m_{G1}$

Equation 74

The total rotor volume is computed for each assumed first stage reduction ratio. The procedure iterates until the minimum total rotor volume (and weight) is found. The iteration is complicated by the fact that the number of planets is a function of a planetary gear's reduction ratio. As the stage reduction ratio increase, the number of allowable whole planets decreases. This function is not continuous but a step since it is impossible to have a fraction of a planet installed. The discrete function is what gives rise to the step-like nature of the curves. These calculations are found in Table 54: Planetary Drive Minimum Weight Solution.

<u>Planetary Drive Modeling</u>

To capture the behavior of the planetary drive, the weight estimation and shafting elements were entered into Model Center 6.0. Model Center's planetary drive model consisted of a weight estimation spreadsheet (Table 50) and individual shaft sizing spreadsheets (refer to SHAFTING for shaft discussion and APPENDIX C: SHAFT DESIGN CALCULATIONS for sample calculations). Shafting spreadsheets were added to include the engine output shaft, main rotor drive shaft, tail takeoff shaft, intermediate

108

tail rotor shaft, and tail rotor drive shaft. Figure 41 shows the planetary drive model in Model Center:

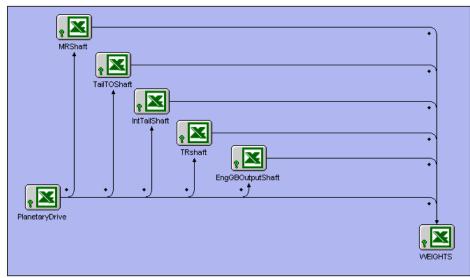


Figure 41: Planetary Drive in Model Center

The weight estimation spreadsheet calculated all speed, torque, power, and power losses for each gear and shaft of the drive system. This spreadsheet also provided a total gearbox weight based on the solid rotor volume method, and total drive system weight estimations based on the Boeing-Vertol and RTL weight equations. Since the solid rotor volume method only estimates gearbox weight, shafting linking gearboxes needed inclusion. To accomplish this, individual spreadsheets were added into Model Center for all external shafts connecting the drive's gearboxes. A final spreadsheet called "WEIGHTS" summed all gearbox, shaft, and drive system weights to allow easy comparison of estimates.

Weight Estimation Results

Weight equations results for the baseline JHL from the solid rotor volume method plus individual shafting calculations, Boeing-Vertol, and RTL yielded a minimum weight results shown in Table 15. The optimal baseline design was determined through a variable metric method optimization in Model Center. The drive system design methodology compares very favorably to the Boeing-Vertol and RTL predictions.

Table 15: Planetary Drive weight Estimate Results				
Component	Shafting Plus Solid Volume	Predicted from RSE	Boeing- Vertol	RTL
	Rotor			
Total Gearbox	13,729			15,835
Weight (lb)				
Total Shafting	1,474			785
Weight (lb)				
Main Rotor Drive			12,024	
Weight (lb)				
Tail Rotor Drive			1,338	
Weight (lb)				
Total Drive System	15,203	15,262	13,361	16,620
Weight (lb)				

 Table 15: Planetary Drive Weight Estimate Results

Table 16: Planetary Baseline Design SummaryParameterValueParameterValue			
rarameter	value	Farameter	value
Overall Reduction Ratio	130.43	MR Power (hp)	22,247
Engine Input Gearbox Reduction Ratio	2.68	Tail Rotor Power (hp)	1989
Crown Bevel Reduction Ratio	4.86	Accessory Power (hp)	120
1 st Stage Planetary Reduction Ratio	3.74	Main Rotor Speed (rpm)	115
2d Stage Planetary Reduction Ratio	2.67	Tail rotor Speed (rpm)	476.3
Overall Planetary Reduction Ratio	10	Main Gearbox Weight (lb)	13,729
Short Shaft Bevel Takeoff Reduction Ratio	0.51	Shaft Weight (lb)	1,474
Intermediate Tail Rotor GB Reduction	1.59	Drive System Weight (lb)	15,203
Tail Rotor Gearbox Reduction Ratio	2.98	Efficiency	97.18%

The key parameters for the weight optimized baseline aircraft is shown below.

SPLIT TORQUE MODEL

A split-torque drive offers great weight savings potential. Litvin (2000) sums up the benefit of the split torque model best by commenting:

> "Gear volume is proportional to the square of gear diameter, while torque-carrying capacity of gearing is proportional to lower order determinants of gear diameter (depending on whether bending or compressive stress evaluations are being used). Therefore, if torque is reduced by approximately one-half (based on the actual percentage of torque split between gears) for a load carrying gear, the weight of the gear can be reduced by more than one-half, due to the square relationship of weight to gear diameter."⁹⁹

The Mi-26 houses a highly successful, operational split-torque helicopter

transmission that is close to the JHL needs. The 105,000 lb Mi-26 is about 35,000 lb

lighter than the JHL Baseline. The Mi-26 main gearbox served as the initially layout for

the JHL torque-split drive. Below is a picture of the Mi-26 main gearbox.

⁹⁹ F.L. Litvin et al, <u>NASA/CR-2000-209909 Handbook on Face Gear Drives With a Spur Involute</u> <u>Pinion</u> (NASA, March 2000), 47.

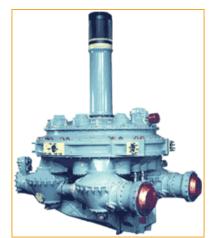


Figure 42: Mi-26 Main Gearbox¹⁰⁰

Drive Arrangement

With a dual engine configuration, the Mi-26 transmission splits the torque evenly three times for a total torque split of 8 (2^3) per engine. The engine input shafts extend through the main gearbox housing power two bevel gears for the main drive and one set of bevel gears to the tail rotor shaft. The total gearbox reduction ratio is 62.52.

¹⁰⁰ Figure extracted from Reductor website: http://www.reductor-pm.ru/eng/allpr.html, September 14, 2005.

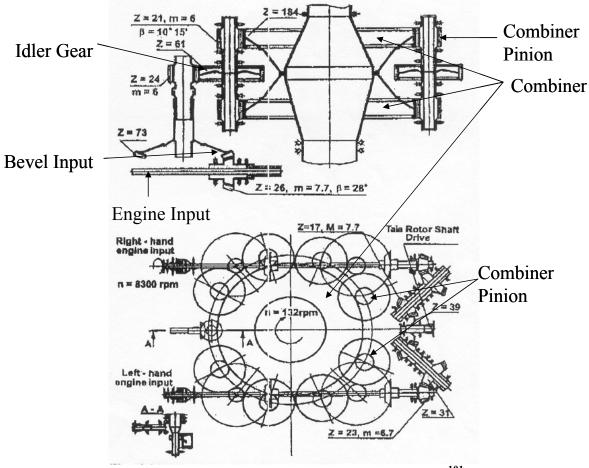


Figure 43: Mi-26 Main Gearbox Arrangement¹⁰¹

¹⁰¹ Marat N. Tishchenko, "Mil Design Bureau Heavy-Lift Helicopters" (as presented at local chapters of the AHS in June 1996), 142.

The split-torque drive study in this thesis follows the same layout and torque splits as the Mi-26; however, the baseline JHL's overall reduction ratio is 130. This figure is almost twice that of the Mi-26's ratio and presents a new challenge in design. Additionally, the JHL has a third engine which means the combiner receives input from 24 combiner pinions as opposed to the Mi-26's 16 pinions. Finally, the tail rotor must also have a bevel takeoff from each combiner to permit power from all three engines.

Torque Split Drive Modeling

The model employed in Model Center for the torque split drive is similar to the planetary drive model. Shafting spreadsheets capture shaft weight for the main rotor drive shaft, tail shaft, intermediate tail rotor shaft, and tail rotor drive shaft. Speed, torque, and power were calculated for the split torque drive much the same way the planetary drive spreadsheet functioned. All calculations are shown in APPENDIX E: SPLIT TORQUE DRIVE CALCULATIONS.

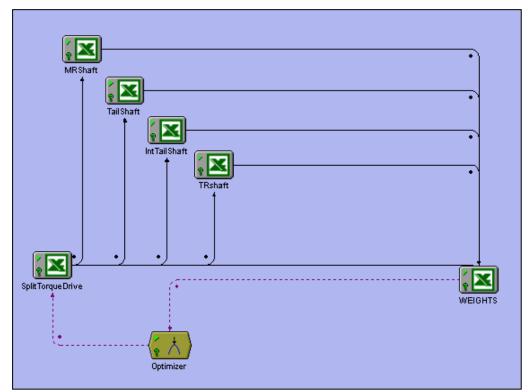


Figure 44: Split Torque in Model Center

Weight Estimation Results

To achieve the higher reduction ratio required by the baseline JHL, slightly higher gear ratios at each stage are required. Especially important is the final stage between the combiner pinion and combiner. In the last stage, the Mi-26 has a reduction ratio of 8.76:1. In the JHL, the last stage has a reduction ratio of 11-13:1. The last stage reduction ratio results in a larger combiner gear, increasing the total size of the main gear box. A reduction ratio close to 12:1 places the main gearbox at the space limits for the

current structure. A reduction ratio just over 13:1 causes the pitchline velocity of the spur combiner to exceed the general spur gear limit of 4,000 feet per minute.¹⁰²

As shown by Figure 45, the success of the split torque drive as configured depends greatly on the final stage reduction ratio.

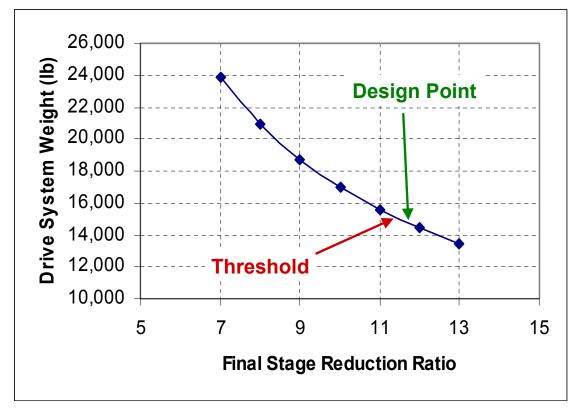


Figure 45: Drive System Weight vs. Final Reduction Ratio

From the graph, the threshold point where the split torque drive becomes lighter than the 15,200 lb planetary drive occurs at a reduction ratio of 11.2:1. The size of the combining gear imposes a limit of 11.7:1 on the reduction ratio. This limit becomes the design point for the final reduction ratio and the main gearbox.

¹⁰² Darle Dudley, <u>Handbook of Practical Gear Design</u> (Lancaster: Technomic, 1994), 1.27.

At 11.7:1, the total system weight is 14,749.6 pounds. Boeing-Vertol and RTL's weight equations are a little higher than the solid rotor volume and shafting estimate. In both the planetary and split torque drive systems, the model's shafting weight is almost double the RTL's estimate.

Component Shafting Predicted RTL **Boeing-Plus Solid** from RSE Vertol Rotor Volume Total Gearbox 13,325 14,274 Weight (lb) Total Shafting 1,425 835 Weight (lb) 15,619 Main Rotor Drive Weight (lb) Tail Rotor Drive 914 Weight (lb) Total Drive System 16,534 14,750 14,478 15,109 Weight (lb)

 Table 17: Split Torque Drive Weight Estimate Results

Table Parameter	A	8: Split Torque Baseline Design Summary Value Parameter	
i ai aniceci	value	i arameter	Value
Overall Reduction Ratio	130.43	MR Power (hp)	22,247
Input Bevel Reduction	3.39	Tail Rotor Power (hp)	1989
Ratio			
Idler Reduction Ratio	3.29	Accessory Power (hp)	120
Combiner Reduction	11.7	Main Rotor Speed (rpm)	115
Ratio			
Tail Takeoff Reduction	0.64	Tail rotor Speed (rpm)	476.3
Ratio			
Intermediate Tail Rotor	2.19	Main Gearbox Weight (lb)	13,325
Gearbox Reduction			
Ratio			
Tail Rotor Gearbox	2.00	Shaft Weight (lb)	1,425
Reduction Ratio			

Drive System Weight (lb)

Efficiency

14,750

97.90%

The key parameters for the weight optimized baseline aircraft is shown below.

RESPONSE SURFACE METHODOLOGY

Overview

Response Surface Methodology (RSM) is a technique to build and optimize empirical models. Through multivariate least squares regression, RSM approximates output response to input parameters with a polynomial empirical equation. This multivariate equation is known as the Response Surface Equation (RSE). To intelligently obtain the regression data, a Design of Experiments (DoE) is created. A properly designed DoE with RSM can capture the underlying factors which influence a response. Using a polynomial equation as an RSE permits rapid, efficient prediction of a much more complex, time consuming calculation.

Initially, a Taylor series, second order approximation form is used for the RSE:¹⁰³

$$R = b_0 + \sum_{i=1}^{k} b_i x_i + \sum_{i=1}^{k} b_i x_i^2 + \sum_{i=1}^{k-1} \sum_{j=i+1}^{k} b_{i,j} x_i x_j + \varepsilon$$

Equation 75

where

R is the dependent parameter (response of interest) b_i are regression coefficient for the first order terms b_{ii} are coefficients for the pure quadratic terms b_{ij} are the coefficients for the cross-product terms x_{ij} are the independent variables ϵ is the associated error for neglecting higher order effects

¹⁰³ Michelle R. Kirby, "An Overview of Response Surface Methodology" as presented in AE6373 lecture (Atlanta: Georgia Institute of Technology, August 25, 2004), 7.

Design of Experiments are "a series of tests in which purposeful changes are made to input variables so that one may observe and identify the reasons for change in an output response."¹⁰⁴ The DoE varies input parameters in an intelligent pattern to capture the response of the system over the entire design space.

To model the drive system responses, an initial DoE casts a wide net around many independent variables in order to obtain data that identifies the most influential factors. This is called a screening test. A Pareto Chart is a convenient method to graphically communicate the percent of response variability, in the given ranges, attributed to a single input parameter. Generally, 20% of the input variables are responsible for 80% of the system variability. The purpose of the screening test is to identify the most important factors influencing the response.

Once the most important independent variables are identified, a second DoE is established that contains only those selected important input parameters. This DoE typically has more independent variable levels and iterations to yield higher fidelity to the final model. The regression data is again fit to the second order linear model and the final RSE is calculated.

<u>Planetary Drive RSM</u>

The planetary drive has 12 independent variables for weight estimation. These variables are:

1. Main rotor power

¹⁰⁴ Michelle R. Kirby, "An Overview of Response Surface Methodology" as presented in AE6373 lecture (Atlanta: Georgia Institute of Technology, August 25, 2004), 8.

- 2. Main rotor rpm
- 3. Planetary reduction ratio
- 4. Tail rotor rpm
- 5. Engine input gearbox reduction ratio
- 6. Tail rotor gearbox reduction ratio
- 7. Engine rpm
- 8. Intermediate tail rotor reduction ratio
- 9. Accessory rpm
- 10. Accessory power
- 11. Oil cooler power
- 12. Tail rotor power required

A DoE with a fractional (half) factorial (2,048 runs) on the twelve variables

permitted screening of six variables (for cumulative 95%) as shown by the Pareto chart;

however, tail rotor power required was kept to allow for future regression of shaft weight.

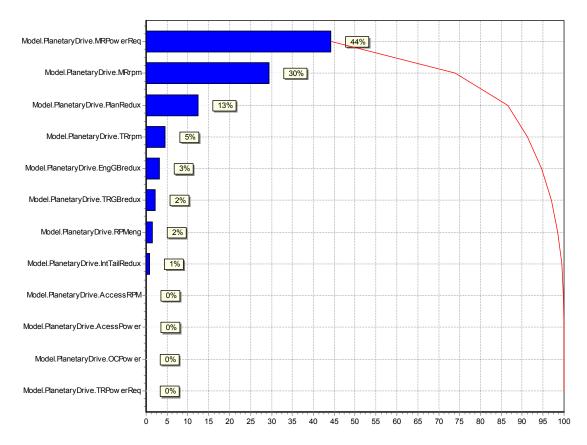


Figure 46: Planetary Drive Screening Test Pareto Chart

For the remaining seven variables, a 3-level full factorial DoE (2,187 runs) captured the data for the RSE model. The RSE formulation from JMP is shown in APPENDIX F: MODEL FIT FOR PLANETARY DRIVE. An optimization within Model Center of the baseline JHL showed the following input variables for the RSE:

Table 19. Dasenne i fanetary Drive KSE input variables								
RSE Input Variables	Units	Туре	Value					
Intermediate Tail Rotor		Optimized	1.593					
GB Reduction								
MR Power Required	hp	Baseline	22,247					
Main Rotor rpm	rpm	Baseline	115					
Overall Planetary		Optimized	10					
Reduction Ratio								
Tail Rotor Gearbox		Optimized	2.984					
Reduction Ratio								
Tail rotor rpm	rpm	Optimized	476.3					
Tail rotor power	hp	Baseline	1,989					
Baseline Drive System	lbs	15,203						
Weight								
RSE Prediction	lbs		15,262					

 Table 19: Baseline Planetary Drive RSE Input Variables

The fitted RSE for the planetary drive demonstrates excellent promise as a weight estimation model. The optimized baseline drive system weight within Model Center was 15,203 pounds while the RSE predicted 15,262 pounds. This residual of 59 pounds is a -0.4% model percent error, which is only a slight deviation from the weight prediction. The RSE has an R² of 99.9% and an R² Adjusted of 99.9%--both well above the recommend 90%. The Actual by Predicted plot (Figure 47) is excellent with vary little deviation from the perfect fit line. The Residual by Predicted plot (Figure 47) shows a good normal distribution about the mean and no discernable pattern—all good indications. The total span of error (about 600 lbs) over the minimum predicted (about 10,000 lbs) is close to 6%--an acceptable, but not ideal, level for this model fit.

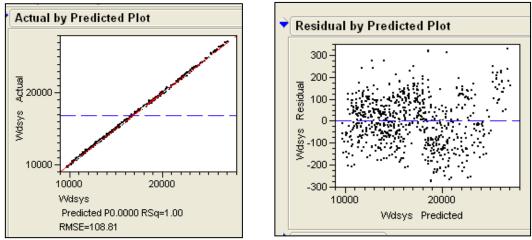


Figure 47: Planetary Drive RSE Model Fit

Split Torque Drive RSM

The split torque drive has lower variety of shafts and gears than the planetary

drive. For the split torque drive nine independent variables were present in the

calculation of weight:

- 1. Combiner reduction ratio
- 2. Main rotor HP
- 3. Main rotor RPM
- 4. Tail rotor HP
- 5. Idler reduction ratio
- 6. Engine RPM
- 7. Tail rotor gearbox reduction ratio
- 8. Tail rotor RPM
- 9. Intermediate tail rotor reduction ratio

The screening test consisted of a 2-level full factorial with 512 runs (2^9) . As shown by the Pareto Chart, the intermediate tail rotor reduction ratio and tail rotor reduction ratio accounted for little influence on the weight response. Of the nine variables, these two were eliminated from consideration.

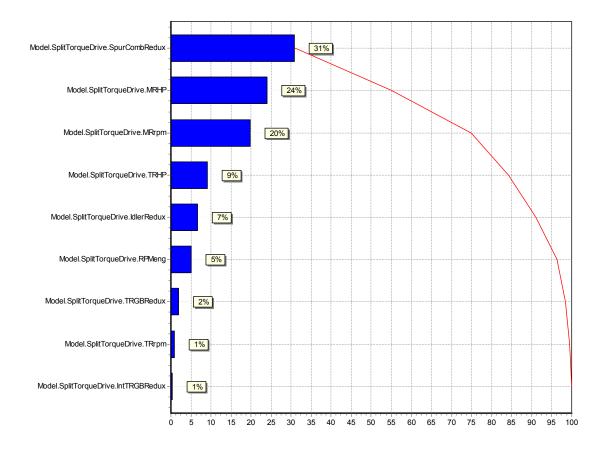


Figure 48: Split Torque Screening Test Pareto Chart

The second DoE consisted of a 3-level, full factorial with 2,187 (3⁷) runs. From this data, a response surface was generated using JMP (see APPENDIX G: MODEL FIT FOR SPLIT TORQUE DRIVE for regression analysis data). Of the 2,187 data sets, 49 or

2.2% did not converge properly for weight estimation and were removed as outliers from the regression data.

Table 20. Split Torque KSE Dasenne inputs								
RSE Input Variables	Units	Туре	Value					
Main rotor speed	rpm	Baseline	115					
Main rotor power	hp	Baseline	22,274					
Tail rotor power	hp	Baseline	1,989					
Engine speed	rpm	Baseline	15,000					
Combiner reduction		Optimized	11.7					
ratio								
Idler reduction ratio		Optimized	3.29					
Tail rotor gearbox		Optimized	2.19					
reduction ratio								
Baseline Drive System	lbs	14,750						
Weight								
RSE Prediction	lbs		14,478					

 Table 20:
 Split Torque RSE Baseline Inputs

The initial model fit for the split torque drive was not as successful as the fit for the planetary drive. Although the initial RSE had an R^2 of 99.6% and an R^2 Adjusted of 99.6%, the Actual by Predicted and Residual by Predicted plots (Figure 49) proved unacceptable. The Actual by Predicted plot showed decent adherence to the perfect fit until about 30,000 lbs; however, the model over predicts at weights above 30,000 lbs. This is confirmed by the Residual by Predicted plot's undesirable S-curve trend. The RSE fit had to be improved for the model to acceptable and serve as a useful tool.

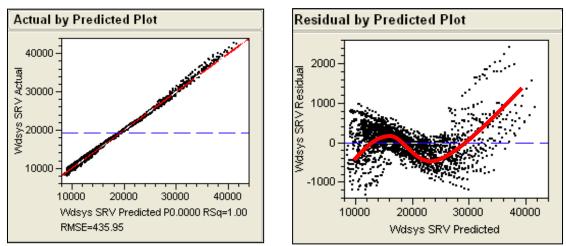


Figure 49: Initial Split Torque RSE Model Fit

To improve the model fit, several techniques were attempted. Results are shown in Table 21. The final model fit output from JMP is listed in APPENDIX G: MODEL FIT FOR SPLIT TORQUE DRIVE.

Table 21. Split Torque Would Fit Comparison									
Type of Model Fit	Weight (lbs)	Model Percent							
		Error							
Optimized Baseline from Model Center	14,750								
Initial RSE	14,122	4.3%							
Bias (x3) RSE	14,133	4.2%							
3/2 Power Transformation RSE	13,710	7.1%							
Square Root Transformation RSE	14,384	2.5%							
Cube Root Transformation RSE	14,478	1.8%							

Table 21: Split Torque Model Fit Comparison

The first technique applied was to bias the fit towards the baseline by adding three data points of the baseline input and response to the DoE data table. Although the biasing did improve the baseline residual, the impact was so minor as to not warrant further application. The second technique consisted of checking different transformations on the response. Drive system weight was transformed by 3/2 power (R

= $W_{dsys}^{3/2}$), square root (R = $W_{dsys}^{1/2}$), and cube root (R = $W_{dsys}^{1/3}$) and then fitted against the 2d order polynomial RSE. The 3/2 power transformation exacerbated the model fit's flaws while transforming in the $\frac{1}{2}$ power direction improved the model in the correct direction (Table 21). Extending to a cube root transformation showed even better results and served as the final model fit. Higher root transformation did reduce the baseline residual; however, the reduction is minimal and higher root transformations experience a condition of diminishing improvement.

The final RSE fit for the split torque drive exhibited good potential as a weight estimation model. The optimized baseline drive system weight within Model Center was 14,750 pounds and the RSE predicted 14, 478 pounds. This 272 pound residual is a 1.8% model percent error. The RSE posses an R2 of 99.9% and an R2 Adjusted of 99.9%---both well above the recommended 90%. The Actual by Predicted plot (Figure 50) is acceptable with improved adherence to the perfect fit line at extremes. The Residual by Predicted plot (Figure 50), showed adequate normal distribution and little discernable pattern. The total span of error (about 0.7 lbs^{1/3}) over the minimal predicted (20 lbs^{1/3}) is approximately 3.5%--a good value for this model.

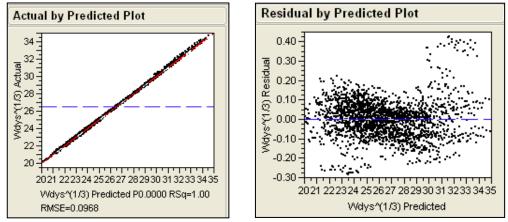


Figure 50: Split Torque Final RSE Model Fit

CONCLUSIONS

A drive system design methodology was presented that permitted integration of detailed component design and allowed higher fidelity and better weight estimates. The methodology is an open architecture allowing the designer to insert and remove component tools as necessary. Component tools for gearing, shafting, and gearbox are included as examples.

A spreadsheet component tool to size spur, helical, or bevel gears for bending and compressive stress as well as scuffing resistance revealed several important conclusions about heavy lift drive gearing. Due to high torque values in the main gearbox, the scuffing hazard proved the most difficult design criteria to meet. VASCO X2M steel with MIL-L-23699 provides the best resistance to scuffing for these high torque gears. Bending strength sized planetary gears due to the reduction in strength from reverse loading. Aerospace gears throughout the drive train operate at very high pitchline velocities requiring high precision, ground gears to reduce dynamic loading and create smooth meshing. Due to the critical nature of helicopter gearing, high reliabilities of a minimum 3σ at a long 2,500 to 5,000 hour life are required. Despite high efficiency gearing, the massive amounts of transmitted power produce a sizeable amount of lost power transformed into heat. In addition to increasing the gear blank temperature and higher scuffing hazards, these high heat quantities require large amounts of force-fed oil flow to properly cool gearing.

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A shafting model is also included as a component tool and integrated into the solid rotor volume weight estimate model. Due to the vibratory nature of the bending moments, an interaction equation is required to calculate a total margin of safety for the combined bending, shear, and torsional stresses experienced by a rotating shaft. From this interaction equation, it was shown that helicopter drive shafts tend to have high diameter to thickness ratios in order to withstand the high torsional stress while maintaining light weight. Throughout the drive system, hollow aluminum alloy shafts were preferred because of great weight savings; however, for shafts with high applied loads such as the main rotor and tail rotor drive shafts, heavier titanium had better resistance to bending and shear stresses. For uniform shafting, the critical speed is simply a function of shaft radius and length. For nonuniform shafting with mass concentrations or bending moments, the critical speed is calculated using a variation of Rayleigh's method. From the variation, the natural frequency is found by comparing the shaft's kinetic energy to its potential energy. For the heavy lift helicopter, shafts linking gearboxes are uniform and typically operate at subcritical speeds while gear shafts and drive shafts are nonuniform and may operate at supercritical or subcritical speeds.

The methodology demonstrated good potential to serve as a system weight predictor. Methodology weight estimates (combined solid rotor volume method and shaft weight estimations) for both a planetary and split torque drive system were within approximately 10% of the Boeing-Vertol and RTL weight equation estimates (Table 22).

rable 22. Total Drive System weight Wethou Summary								
Planetary Drive	Split Torque							
Weight (lb)	Weight (lb)							
15,203	14,750							
15,262	14,478							
13,361	16,534							
16,620	15,109							
	Planetary Drive Weight (lb) 15,203 15,262 13,361							

 Table 22: Total Drive System Weight Method Summary

This weight estimation method was successfully applied to a traditional multistage planetary drive and to a split torque drive similar to the Mi-26's main gearbox. For the planetary main gearbox, low planetary reduction ratios are preferred because of increased load sharing gained from more planets. The ideal main gearbox reduction ratio is a function of each stages reduction ratio. The shown split torque drive included three torque splits for a final stage total of eights paths per engine. Splitting the torque showed great weight savings potential over the traditional, 2-stage planetary drive system if high reduction ratios (above 12:1) can be achieved in the final stage. This requirement for a high final reduction ratio face gears.

Approximating the more complicated model created in Model Center by Response Surface Methodology produced good results. The planetary model's initial RSE was an excellent fit with a model percent error less than 1%; however, the split torque regression required a cube root transformation to yield a workable model with a model percent error less than 2%. With correct model fitting, Response Surface Methodology demonstrated the ability to serve as a simplified response predictor for a more complicated, high component integrated drive system model.

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FUTURE WORK

The addition of detailed, user friendly tools for bearings, freewheeling units, rotorbrakes, splines, housing, structural support, and aerodynamic loads on drives shafts is needed to fully capture the drive system behavior and find a feasible, optimized solution.

In addition to the RSM, the drive modeling must be examined from a probabilistic point of view. Placing distributions on assumptions and key factors will capture the uncertainty associated with a design. The impact and overall probability of success may then be judged through the application of Monte Carlo simulations.

Lastly, the face gear holds potential to save weight in split-torque designs and are "optimal for large reduction ratio applications" that occur in helicopter drive trains.¹⁰⁵ Testing of prototype face gears for a split-torque helicopter transmission by Handschuh, Lewicki, and Bossler (1992),¹⁰⁶ and the recent success of the 5,100 HP RDS-21 Demonstrator Gearbox (2004),¹⁰⁷ have confirmed the weight saving benefits. Handschuh, Lewicki, and Bossler indicated that face gears can have "an improved weight advantage compared to spiral bevel gears at [reduction] ratios higher than approximately 3.5:1." Face gear technology is a major facet of the RDS-21 and future drive systems.

¹⁰⁵ Yuriy Gmirya, et al, "Design and Analysis of 5100 HP RDS-21 Demonstrator Gearbox" <u>60th Annual</u> <u>Forum Proceedings</u>, vol 2, (Alexandria: AHS International, 2004), 1224.

 ¹⁰⁶ Handschuh, R., D. Lewicki, and R. Bossler, <u>NASA TR 92-C-008 Experimental Testing of Prototype Face Gears for Helicopter Transmissions</u> prepared for "Gearbox Configurations of the 90's" sponsored by the Institute of Mechanical Engineers Solihull, West Midlands, United Kingdom, October 28, 1992.
 ¹⁰⁷ Yuriy Gmirya, et al, "Design and Analysis of 5100 HP RDS-21 Demonstrator Gearbox" <u>60th Annual Forum Proceedings</u>, vol 2, (Alexandria: AHS International, 2004),

Development of a standardized, analytical estimate of gear stresses becomes the key to integrating face gear technology into the future drive system design methodology.

APPENDIX A: JHL SUPPLEMENTAL PACKAGE

EXTRACTS

Design	Baseline	E1	E2	E2A	E4	E5	E6	E7
Design Payload, ton	20	16	26	26	20	20	20	20
Design Radius, nm	250	250	250	250	400	500	250	250
High / Hot, 1000 ft / deg F	4k/95	4k/95	4k/95	4k/95	4k/95	4k/95	6k/95	4k/95
Shipboard Operations	Capable	Capable	Capable	Capable	Capable	Capable	Capable	Compatible
Summary								
Design Cruise Speed, kt	171.7	170.3	173.4	XXX.X	173.1	174.1	175	XXX.X
Design Gross Weight, Ib	138,868	114,035	177,392	XXX,XXX	172,197	206,501	148,606	XXX,XXX
Disk Loading, psf	12.3	12.3	12.3	XX.X	12.3	12.3	12.3	XX.X
Download %GW	4.06%	4.66%	3.44%	x.xx%	3.51%	3.11%	3.87%	x.xx%
Max Alternate Gross Wt, Ib	173,556	142,529	221,759	XXX,XXX	215,184	258,036	185,754	XXX,XXX
Number Engines	3	3	3	3	3	3	3	3
Engine Size, shp	10,985	9,114	13,912	XX,XXX	13,495	16,066	13,113	XX,XXX
Drive Rating (TO rpm), shp	25,964	21,420	33,011	XX,XXX	30,642	35,469	27,567	XX,XXX
Fuel Tank Capacity (JP-8), lb	50,015	40,899	63,823	XX,XXX	65,557	81,131	51,552	XX,XXX
Unit Flyaway Cost, FY05 \$M	120.38	98.80	154.94	XXX.XX	149.32	179.76	139.92	XXX.XX
Primary Thruster								
Number Primary Thrusters	1	1	1	1	1	1	1	1
Number Blades per Rotor	6	6	6	6	6	6	6	6
Hover Tip Speed, fps	725	725	725	725	725	725	725	725
Diameter, ft	120	108.6	135.5	XXX.X	133.5	146.2	124	XXX.X
Rotor Hub Separation, ft	-	-	-	-	-	-	-	-
Effective Disk Area, sq ft	11,310	9,263	14,420	XX,XXX	13,998	16,787	12,076	XX,XXX
Dimensions								
Operating Footprint, sq ft	17,904	14,683	22,832	XX,XXX	22,161	26,565	19,121	XX,XXX
Operating Length, ft	149.2	135.2	168.5	XXX.X	166	181.7	154.2	XXX.X
Operating Width, ft	120	108.6	135.5	XXX.X	133.5	146.2	124	XXX.X
Stowed Footprint, sq ft	3,744	X,XXX						
Stowed Length, ft	90.88	XXX.X						
Stowed Width, ft	41.2	XXX.X						
Stowed Height, ft	25.19	XX.X						
Cargo Box Volume, cu ft	5,080	5,080	5,080	X,XXX	5,080	5,080	5,080	5,080
Cargo Box Length, ft	50	50	50	XX.X	50	50	50	50
Cargo Box Width, ft	11.1	11.1	11.1	XX.X	11.1	11.1	11.1	11.1
Cargo Box Height, ft	9.2	9.2	9.2	XX.X	9.2	9.2	9.2	9.2
Weight Summary								
Weight Empty, Ib	76,739	63,074	98,454	XX,XXX	94,500	113,207	84,961	XX,XXX
Scar Weight, Ib	X,XXX	X,XXX	X,XXX	X,XXX	X,XXX	X,XXX	X,XXX	X,XXX
Operating Weight, Ib	78,829	65,124	100,584	XX,XXX	96,590	115,297	87,051	XX,XXX
Struc Design GW (SDGW), lb	138,845	114,024	177,407	XXX,XXX	172,148	206,429	148,603	XX,XXX
Maximum Payload Wt, lb	65,000	52,000	84,500	XX,XXX	65,000	65,000	65,000	65,000
Max VTOL Gross Wt, lb	XXX,XXX	XXX,XXX	XXX,XXX	XXX,XXX	XXX,XXX	XXX,XXX	XXX,XXX	XXX,XXX
Program Cost (FY05 \$)								
JHL Fleet Size	400	400	400	400	400	400	400	400
Development (RDT&E), \$B	X,XXX	X,XXX	X,XXX	X,XXX	X,XXX	X,XXX	X,XXX	X,XXX
Procurement, \$B	XX,XXX	XX,XXX	XX,XXX	XX,XXX	XX,XXX	XX,XXX	XX,XXX	XX,XXX
O&S (30 years), \$B	XX,XXX	XX,XXX	XX,XXX	XX,XXX	XX,XXX	XX,XXX	XX,XXX	XX,XXX
Unit Cost (FY05 \$)								
Flyaway, \$M	120.38	98.80	154.94	XXX.XX	149.32	179.76	139.92	XXX.XX
Flyaway / (Wt Empty), \$/lb	1,569	1,566	1,574	X,XXX	1,580	1,588	1,647	X,XXX
O&S (150 FH/yr), \$/FH	X,XXX	x,xxx	X,XXX	x,xxx	X,XXX	X,XXX	X,XXX	x,xxx
	^,^^^	,,,,,,	,,,,,,	^,^^^	,,,,,,,	^,^^^	,,,,,,	7,7000

Design	Baseline	E1	E2	E2A	E3	E4	E5	E6	E7
Design Payload, ton	20	16	26	26	20	20	20	20	20
Design Radius, nm	250	250	250	250	210	400	500	250	250
High / Hot, 1000 ft / deg F	4k/95	4k/95	4k/95	4k/95	4k/95	4k/95	4k/95	6k/95	4k/95
Shipboard Operations	Capable	Capable	Capable	Capable	Capable	Capable	Capable	Capable	Compatible
Areas									
Wetted (total), sq ft	5,723	5,428	6,151	x,xxx	5,640	6,093	6,460	5,864	x,xxx
Drag Area (cruise mode), sq ft	108.7	95.2	129.3	XXX.X	104.8	126.5	144.7	114.1	XXX.X
Drag Area (hover mode), sq ft	108.7	95.2	129.3	XXX.X	104.8	126.5	144.7	114.1	XXX.X
Group Weights									
Wing Group, Ib	0	0	0	0	0	0	0	0	0
Rotor Group, Ib	19,250	14,802	26,640	XX,XXX	17,961	25,594	32,591	22,321	XX,XXX
Empennage Group, Ib	2,250	1,750	3,073	x,xxx	2,122	2,879	3,563	2,441	x,xxx
Fuselage Group, lb	13,836	11,886	16,728	XX,XXX	13,297	16,339	18,816	14,580	XX,XXX
Alighting Gear Group, lb	3,627	2,979	4,635	X,XXX	3,446	4,497	5,393	3,882	x,xxx
Nacelle Group, lb	1,211	1,000	1,544	x,xxx	1,153	1,497	1,791	1,453	x,xxx
Air Induction Group, lb	247	199	323	x,xxx	234	312	381	302	x,xxx
Total Structure, lb	40,421	32,615	52,943	XX,XXX	38,213	51,118	62,535	44,979	XX,XXX
Propulsion Group, Ib	20,092	16,468	25,755	XX,XXX	19,137	24,692	29,426	21,999	XX,XXX
Flight Controls Group, Ib	3,995	3,280	5,134	x,xxx	3,799	4,979	5,995	4,925	x,xxx
Auxiliary Power Group, Ib	300	275	350	x,xxx	300	300	300	300	x,xxx
Instruments Group, Ib	135	135	135	x,xxx	135	135	135	135	x,xxx
Hydraulic Group, lb	963	839	1,133	X,XXX	937	1,089	1,210	1,091	X,XXX
Pneumatic Group, Ib	0	0	0	X,XXX	0	1	2	3	X,XXX
Electrical Group, lb	820	754	886	X,XXX	820	820	820	820	x,xxx
Avionics Group, Ib	1,150	1,150	1,150	x,xxx	1,150	1,150	1,150	1,150	X,XXX
Armament Group, Ib	145	145	145	x,xxx	145	145	145	145	x,xxx
Furnishings & Equip., lb	950	910	990	x,xxx	950	950	950	950	x,xxx
Environmental Control, lb	750	750	750	X,XXX	750	750	750	750	x,xxx
Anti-Icing Group, Ib	1,631	1,338	2,087	X,XXX	1,549	2,025	2,430	1,885	x,xxx
Load & Handling Group, lb	1,275	1,040	1,710	X,XXX	1,275	1,275	1,275	1,275	x,xxx
Contingency, Ib	4,112	3,375	5,286	X,XXX	3,914	5,073	6,087	4,558	x,xxx
Total Weight Empty, lb	76,739	63,074	98,454	XX,XXX	73,074	94,501	113,210	84,965	XX,XXX

Table 24: Example JHL Substantiation

Table 25: JHL Baseline Tabulated Data

Heavy Lift Helicopter (JHL-JH-20T) Tabulated Data

Main Rotor		Wetted Areas	
Diameter	120.00 ft	Fuselage	3,409.20 ft ²
Chord	4.24 ft	Sponson	1,455.60 ft ²
Twist	-12.00 deg	Nacelles	310.50 ft ²
Blades	6	Pylon	491.10
	-	-)	
Tail Rotor		Landing Gear	
Diameter	29.07 ft	Main Gear Track	20.05 ft
Chord	2.85 ft	Main Gear Tread	16.67 ft
Blades	5	Wheelbase	38.33 ft
Horizontal Tail		Cargo Compartment	
Planform Area	212.21 ft ²		50.00 ft
Span	32.57 ft	Length Width	11.08 ft
Chord	6.51 ft	Height	9.17 ft
Aspect Ratio	5.00 ND	Floor Area	554.00 ft ²
Taper Ratio	1.00 ND	Ramp Opening Width	11.08 ft
Sweep Angle	0.00 deg	Ramp Opening Height	9.17 ft
Dihedral Angle	0.00 deg		•••••
Incidence Angle	1.00 deg		
Thickness to Chord Ratio	0.12 ND		
Vertical Tail			
Planform Area	168.72 ft ²		
Span	18.69 ft		
Chord	7.60 ft		
Aspect Ratio	2.07 ND		
Taper Ratio	0.75 ND		
Sweep Angle	35.00 deg		
Dihedral Angle	0.00 deg		
Incidence Angle	1.50 deg		
Thickness to Chord Ratio	0.12 ND		

Sample: A004 Aircraft Drawings and Dimensions 6.b.

Gross Weight, Ib	138.867						
Cross Weight, is	100,007						
Airspeed, kt	0	20	40	60	80	100	110
Power Required, shp	22,247	20,402	16,369	13,050	11,291	10,746	10,840
Primary Thruster, shp	19,346	17,871	14,516	11,677	10,128	9,643	9,734
Induced, shp	17,309	15,770	12,295	9,287	7,351	6,152	5,786
Profile, shp	2,037	2,054	2,101	2,201	2,346	2,525	2,632
Parasite, shp	-	48	120	190	431	966	1,316
Control + Prop, shp	1,989	1,631	1,018	625	474	433	434
Drive Sys Loss, shp	792	779	716	628	569	549	552
Accessory Loss, shp	120	120	120	120	120	120	120
Power Available							
AEO Engine (MRP), shp	23,413	23,426	23,465	23,535	23,631	23,753	23,824
OEI Engine (MRP), shp	15,609	15,618	15,643	15,690	15,754	15,836	15,883
AEO Engine (MCP), shp	17,381	17,393	17,428	17,487	17,569	17,675	17,737
OEI Engine (MCP), shp	11,587	11,595	11,619	11,658	11,713	11,783	11,82
Drive System Rating, shp	25,964	25,964	25,964	25,964	25,964	25,964	25,964
Fuel Consumption							
SFC, lbf/hp-hr	0.349	0.356	0.374	0.398	0.416	0.422	0.420
Spec Range, nm/lb	0	0.0028	0.0065	0.0116	0.0171	0.0221	0.0242
Main Rotor							
Tip Speed, ft/sec	725	725	725	725	725	725	725
Thrust, Ib	144,500	144,146	142,596	142,516	142,915	142,226	141,937
TPP AoA (+ backward), deg	-3.58	-0.31	-0.39	-0.41	-0.7	-1.27	-1.57
CT/Sigma (TW)	0.10057	0.10032	0.09924	0.09919	0.09946	0.09899	0.09878
Lift							
Primary Thruster, Ib	144,492	144,144	142,593	142,512	142,905	142,192	141,883
Wing, Ib	0	0	0	0	0	0	0
Propulsion							
Primary Thruster, Ib	-1,482	785	978	1,030	1,754	3,147	3,898
Aux Thruster, Ib	0	0	0	0	0	0	0
Conversion States - N/A for Single Main Helicop	oter						
Data for Points on Pwr Required Curves	V-be (kts)	V-br (kts)	90% MCP				
Airspeed, kts	121.5	167.0	171.7				
Power Req, shp	11,212	15,637	16,435				

Table 26: JHL Baseline Power vs. Airspeed Data 0 to 110 knots

Table 27: JHL Baseline Power vs. Airspeed Data 120 knots or more

Gross Weight, Ib							
Gross weight, ib							V-mcp
Airspeed, kt	120	130	150	160	170	180	182
Anapeed, Kt	120	150	150	100	170	100	102
Power Required, shp	11,150	11,663	13,297	14,546	16,132	17,979	18,380
Primary Thruster, shp	10,022	10,499	12,022	13,189	14,676	16,412	16,789
Induced, shp	5,542	5,406	5,434	5,596	5,815	6,016	6,054
Profile, shp	2,750	2,880	3,180	3,464	3,921	4,547	4,690
Parasite, shp	1,730	2,212	3,407	4,129	4,940	5,849	6,045
Control + Prop, shp	444	462	520	566	626	701	718
Drive Sys Loss, shp	564	582	636	671	710	747	753
Accessory Loss, shp	120	120	120	120	120	120	120
Power Available							
AEO Engine (MRP), shp	23,901	23,988	24,181	24,288	24,405	24,525	24,549
OEI Engine (MRP), shp	15,934	15,992	16,121	16,192	16,270	16,350	16,360
AEO Engine (MCP), shp	17,806	17,880	18,048	18,141	18,241	18,347	18,369
OEI Engine (MCP), shp	11,871	11,920	12,032	12,094	12,161	12,231	12,240
Drive System Rating, shp	25,964	25,964	25,964	25,964	25,964	25,964	25,964
Fuel Consumption							
SFC, lbf/hp-hr	0.415	0.409	0.391	0.381	0.369	0.359	0.357
Spec Range, nm/lb	0.0259	0.0273	0.0288	0.0289	0.0285	0.0279	0.0277
Main Rotor							
Tip Speed, ft/sec	725	725	725	725	725	725	725
Thrust, Ib	141,704	141,524	141,338	141,337	141,399	141,528	141,563
TPP AoA (+ backward), deg	-1.9	-2.25	-3	-3.41	-3.84	-4.29	-4.39
CT/Sigma (TW)	0.09862	0.0985	0.09837	0.09837	0.09841	0.0985	0.09852
Lift							
Primary Thruster, Ib	141,627	141,416	141,144	141,086	141,081	141,131	141,148
Wing, Ib	0	0	0	0	0	0	0
Propulsion							
Primary Thruster, Ib	4,697	5,545	7,402	8,409	9,470	10,588	10,824
Aux Thruster, Ib	0	0	0	0	0	0	0
Conversion States - N/A for Single Main Helicopte							
Data for Points on Pwr Reguired Curves							
Airspeed, kts							
Power Reg, shp							

APPENDIX B: SPUR-HELICAL GEAR RATING

CALCULATIONS

<u>Spur-Helical Gear Summary</u>

SPUR-HELICAL RATING SUMMARY

AGMA Stress AGMA Stress	AGMA Stress AGMA Stress	AGMA Stress AGMA Stress	AGMA Stress AGMA Stress	211.2 AGMA Stress	AGMA Stress	AGMA Stress AGMA Stress AGMA Stress AGMA Stress AGMA Stress AGMA Stress AGMA Stress
AC	1.163 AC	1.130 AC	1.029	211.2 A0	AC	X X X X X
GEAR 0.0920 0.4796	38,568.7 33,168.4	171,281.2 151,609.4	7.6389	204.2	339,205.0	84,801.3
MESH	1.095	1.111	0.986 3.5000			7,907.2 11,101.3 4,040.5 0.0
PINION 0.0920 0.4591	38,242.4 34,909.3	169,054.5 152,170.9	5.4167 3.5000	102.7	240,527.2	60,131.8
UNITS						
	psi psi	psi psi	<u>c</u>	മ	d ri	ri 16 16 17 16 19 19 19 19 19 19 19 19 19 19 19 19 19
DESCRIPTION Pitting Resistance Geometry Factor Bending Strength Geometry Factor	Working bending stress number Bending tensile stress	Working contact stress number Contact stress	DESIGN BALANCE Standard (reference) pitch radius Effective face width	Estimated Weight	Tmember Torque (per member)	Torque (per mesh) Pitch Line Velocity Tangential load (transmitted) Radial load Axial load (thrust) Gear ratio
SYMBOL	swt V st E	swc sc	ж д	W	Tmember 1	Т, ТО, О v, v, V, PLV F Wr Wa Ma
MDLCTR	SH_swt_P, SH_swt_G SH_st_P, SH_st_G	SH_swc_P, SH_swc_G SH_sc_P, SH_sc_G	SH_R_P SH_R_G SH_Fe	SH_W_P, SH_W_G	SH_Tmember_P, SH_Tmember_G	SH_T_P, SH_T_G SH_PLV SH_Wt SH_Wr SH_Wa SH_wa SH_wa

Table 28: Spur-Helical Gear Summary

141

Lube Lube	Lube Scoring	Scoring	Scoring	Scoring	Material
39.90 5.32	144.1 142.53	371.3 0.23%	LOW	1.39	VASCO X2M
					VASCO X2M
lb/min gal/min	,÷ ÷	Ч°			
Oil flow	Oil temperature (average) Maximum flash temperature	Maximum contact temperature	Provability or scoring nazaru Scuffing Risk	Safety Factor (Scoring)	Selected Material
	toil tflmax		Risk		Material
SH_M	SH_toil SH_tflmax	SH_tcmax	SH_ScoreRisk	SH_ScoreSF	

Table 28: Spur-Helical (continued)

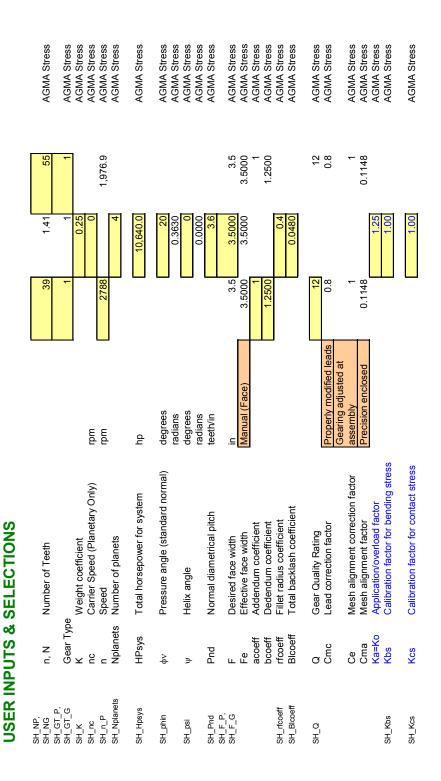


Table 29: Spur-Helical User Inputs and Selections

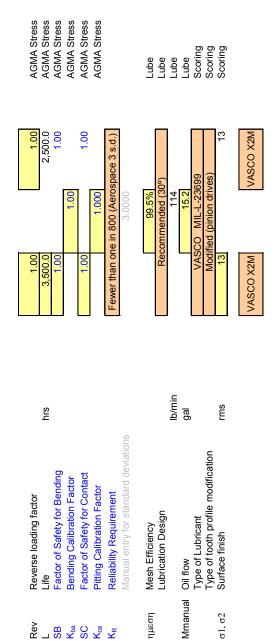


Table 29: Spur-Helical User Inputs and Selections (continued)

SH_eta

SH_SurfFin

SH_Mman

K K S S B C Kev

SH_FSc SH_Kca

SH_Rev_P, SH_Rev_G SH_Life SH_FSb SH_FSb

AGMA Stress Equations

Table 30: Spur-Helical AGMA Stress Equations

AGMA STRESS SUMMARY

SYMBOL I J	DESCRIPTION Pitting Resistance Geometry Factor Bending Strength Geometry Factor	<u>UNIT</u>	S <u>PINION</u> 0.0920 0.4591	<u>MESH</u>	<u>GEAR</u> 0.0920 0.4796	
S _{wt}	Working bending stress number	psi	38,242.4	1.095	38,568.7	1.163
st	Bending tensile stress	psi	34,909.3	1.000	33,168.4	1.105
S _{WC}	Working contact stress number	psi	169,054.5	1,111	171,281.2	1,130
Sc	Contact stress	psi	152,170.9	1.111	151,609.4	1.130
R	Standard (reference) pitch radius	in	5.4167		7.6389	
t _{cmax}	Maximum contact temperature	°F		371.306		
	Scuffing Risk			LOW		
S _B	Safety Factor (Scoring)			1.39		
	Estimated Weight		102.7		204.2	306.9
			Input		Output	
				Input-Output		

1

PINION

2788

Mesh

MESH

1.41

0.25

4

2

<u>GEAR</u>

55

1,976.9

INPUT PARAMETERS

n, N Gear Type	Number of Teeth	
K	Weight coefficient	
	Carrier Speed (Planetary Only)	rom
n _c	1 (, , ,,	rpm
n	Speed	rpm
N _{planets}	Number of planets	

FORCE ANALYSIS

			PINION MESH	GEAR
HP, P, H	Horsepower (per mesh)	hp	2,660.0	2,660.0
T, TQ, Q	Torque	ft Ib	5,011.0	7,066.8
		in lb	60,131.8	84,801.3
v _t , V, PLV	Pitch Line Velocity	ft/min	7,907.2	
		in/min	94,886.6	
Wt	Tangential load (transmitted)	lb	11,101.3	
Wr	Radial load	lb	4,040.5	
Wa	Axial load (thrust)	lb	0.0	
W	Total force	lb	11,813.7	

GEAR	GEOMETRY		PINION	MESH	GEAR
P, P _d	Diametrical pitch (transverse)	teeth/in		3.6000	
P _{design}	Recommended diametrical pitch for He	rtz stress		5.4152	
φ _n	Pressure angle (standard normal)	degrees		20	
		radians		0.3491	
ϕ, ϕ_t	Standard traverse pressure angle	degrees		20.0000	
		radians		0.3491	
Ψ	Helix angle	degrees		0	
_		radians		0.0000	
P _{nd}	Normal diametrical pitch	teeth/in		3.6	1.103377919
F _{design}	Recommeded spur face width for Hertz			7.2020	
F	Desired face width	in	3.5		3.5
F _e	Effective face width	Manual (Face)	3.5000		3.5000
m _G	Gear ratio			1.4103	
R	Standard (reference) pitch radius	in	5.4167		7.6389
d, d _p	Operating pitch diameter of pinion	in		10.8333	
R _b	Base radius	in	5.0900		7.1782
Cr	Operating center distance	in		13.0556	
φ _r	Operating transverse pressure angle	degrees		20.0000	
		radians		0.3491	
pb	Transverse base pitch	in/tooth		0.8200	
p _N	Normal base pitch	in/tooth		0.8200	
ψ_b	Base helix angle	degrees		0.0000	
		radians		0.0000	
x1, x2	Addendum modification		0		0
a _{coeff}	Addendum coefficient		1]	1
a	Addendum	in	0.2778	1	0.2778
b _{coeff}	Dedendum coefficient		1.2500	J	1.2500
b c	Dedendum Clearance	in in	0.3472 0.0694		0.3472 0.0694
rfcoeff	Fillet radius coefficient		0.0004	0.4	0.0004
r _f	Fillet radius	in		0.1431	
Blcoeff	Total backlash coefficient			0.0480	
BL total	Total backlash	in		0.0133	
Ro	Outside radius	in	5.6944		7.9167
C ₆		in		4.465262982	
C ₁		in		1.126558456	
C ₃		in		1.85260911	
C ₄	HPSTC	in		1.946594966	
C ₅		in		2.553151047	
C ₂	LPSTC	in		1.733114537	
Z	Length of line of contact	in		1.42659259	
m _p	Transverse contact ratio			1.7397	
p _x	Axial pitch			999,999.9999	
m _F	Axial contact ratio			0	
	fractional part of mp			0.7397	
n _r					
n _a	fractional part of mF			0	
L _{min}	Minimum length of lines of contact	in		3.5000	
m _N	Load sharing ratio			1.0000	
ψ_r	Operating helix angle	degrees		0	
		radians		0 0000	

radians

degrees

radians

 ϕ_{nr}

Operating normal pressure angle

0.0000

0.3491

20

BEND	NG STRESS				
			PINION	MESH	GEAR
st	Bending tensile stress	psi	34,909.3		33,168.4
t _{r_min}	Minimum rim thickness below tooth root	in	0.7500		0.7500
ht	Whole depth	in	0.6250		0.6250
m _B	Backup ratio		1.2000		1.2000
K _B	Rim thickness factor		1		1
Q	Gear Quality Rating		12		12
Q _v	Transmission Quality Rating			12	
В				0	
A		f		106	
v _t	Pitch line velocity	fps		7,907.2	
V _{tmax}	Maximum pitch line velocity	fps		13,225.0	
K _v	Dynamic factor			1.0000	
Ks	Size factor		1		1
C _{mc}	Lead correction factor	Properly modified leads	0.8		0.8
F/(10d)	Dinion propertion factor		0.0500 0.0386		0.0500 0.0292
C _{pf}	Pinion proportion factor				
C _{pm}	Pinion proportion modifier	Cooring adjusted at	1.1		1.1
Ce	Mesh alignment correction factor	Gearing adjusted at assembly	1		1
A	Mesh alignment empirical constant	accombry	0.0675		0.0675
В	Mesh alignment empirical constant		0.0128		0.0128
С	Mesh alignment empirical constant		-0.0000926		-0.0000926
C _{ma}	Mesh alignment factor	Precision enclosed	0.1112		0.1112
C _{mf}	Face load distribution factor		1.1229		1.1146
C _{mt}	Transverse load distribution factor		1		1
K _m	Load distribution factor		1.1229		1.1146
K _a =K _o	Application/overload factor			1.25	
K _{bs}	Calibration factor for bending stress			1.00	

CONTACT STRESS					
			PINION	MESH	GEAR
Sc	Contact stress	psi	152,170.9		151,609.4
Cp	Elastic coefficient	psi ^{1/2}		2,276.7	
C _a =C _o	Application/overload factor			1.25	
Cv	Dynamic factor			1.0000	
Cs	Size factor		1		1
d, d _p	Operating pitch diameter of pinion	in		10.8333	
F _e	Effective face width of narrowest membe	r in		3.5000	
C _m	Load distribution factor		1.1229		1.1146
C _f	Surface condition factor		1		1
I	Pitting Resistance Geometry Factor		_	0.0920	
K _{cs}	Calibration factor for contact stress			1.00	

PITTING RESISTANCE GEOMETRY FACTOR, I

			PIN	ION	MESH	GEAR
d, d _p	Operating pitch diameter of pinion	in			10.8333	
R _{m1}	Mean radius of pinion	in			5.4167	
ρ_1	Radius of curvature for pinion				1.7331	
ρ_2	Radius of curvature for gear				2.7321	
ρ_{m1}	Radius of curv. at mean radius of pinion	n			1.8526	
ρ_{m2}	Radius of curv. at mean radius of gear				2.6127	
C_{ψ}	Helical overlap factor				1.0000	
T,	Pitting Resistance Geometry Factor				0.0920	

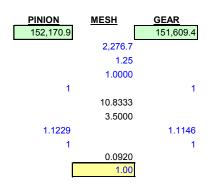
BEND	ING STRENGTH GEOMET	PINION	MESH	GEAR	
Ψ	Helix angle	degrees	0		0
N _G	Gear tooth count		39		55
N _{mate}	Mate tooth count		55		39
J _{helixregext}	J for helical, external gears		0.5351		0.5071
	J for spur (external or internal)		0.4591		0.4796
	J for helical, internal gears (est)		0.5951		0.5671
J	Bending Strength Geometry Factor		0.4591		0.4796

BENDING STRESS

DEND			PINION	MESH	GEAR
St	Bending tensile stress	psi	34,909.3		33,168.4
	Minimum rim thickness below tooth root	psi in	0.7500	L	0.7500
t _{r_min}					
ht	Whole depth	in	0.6250		0.6250
m _B	Backup ratio		1.2000		1.2000
K _B	Rim thickness factor		1	_	1
Q	Gear Quality Rating		12		12
Qv	Transmission Quality Rating			12	
В				0	
A				106	
Vt	Pitch line velocity	fps		7,907.2	
V _{tmax}	Maximum pitch line velocity	fps		13,225.0	
K _v	Dynamic factor			1.0000	
Ks	Size factor		1		1
C _{mc}	Lead correction factor	Properly modified leads	0.8		0.8
F/(10d)			0.0500		0.0500
C _{pf}	Pinion proportion factor		0.0386		0.0292
Cpm	Pinion proportion modifier		1.1		1.1
		Gearing adjusted at			
Ce	Mesh alignment correction factor	assembly	1		1
A	Mesh alignment empirical constant		0.0675		0.0675
В	Mesh alignment empirical constant		0.0128		0.0128
C	Mesh alignment empirical constant	Dessision englaced	-0.0000926		-0.0000926
C _{ma}	Mesh alignment factor	Precision enclosed	0.1112		0.1112
C _{mf}	Face load distribution factor		1.1229		1.1146
C _{mt}	Transverse load distribution factor		1		1
K _m	Load distribution factor		1.1229		1.1146
K _a =K _o	Application/overload factor			1.25	
K _{bs}	Calibration factor for bending stress			1.00	

CONTACT STRESS

Sc	Contact stress	psi
C _p	Elastic coefficient	psi ^{1/2}
C _a =C _o	Application/overload factor	
Cv	Dynamic factor	
Cs	Size factor	
d, d _p	Operating pitch diameter of pinion	in
Fe	Effective face width of narrowest member	in
C _m	Load distribution factor	
C _f	Surface condition factor	
I	Pitting Resistance Geometry Factor	
K _{cs}	Calibration factor for contact stress	



ALLOWABLE BENDING STRESS

S _{wt}	Working bending stress number	psi
S _{at}	Allowable bending stress	psi
Rev	Reverse loading factor	
K _T	Temperature factor for bending strength	
L	Life	hrs
n	Speed	rpm
q	Number of contacts per revolution	
N	Number of stress cycles	
Y _N	Stress cycle factors	
$s_{at}Y_{N}$	Localized yielding limit	psi
H _B	Brinell hardness number	
Say	Allowable yield strength number	
	Reliability Requirement	
K _R	Reliability factor	
	Manual entry for standard deviations	
	Number of standard deviations	
	Desired reliability	
	Coefficient of variation (bending)	
FS _B	Factor of Safety for Bending	
K _{ba}	Bending Calibration Factor	

PINION	MESH	GEAR
38,242.4		38,568.7
51,990.1		51,990.1
1.00		1.00
1.00		1.00
3,500.0		3,500.0
2,788.0		1,976.9
4		4
2.342E+09		1.661E+09
0.8809		0.8885
45,800.6		46,191.3
647		647
279,202.5		279,202.5
Fewer than	one in 800 (Aero	space 3 s.d.)
	1.20	
	3.0000	
	3.0000	
	0.9987	
	0.1560	
1.00		1.00
	1.00	

ALLOWABLE CONTACT STRESS

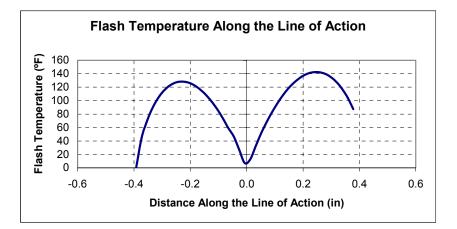
			PINION	MESH	GEAR
S _{WC}	Working contact stress number	psi	169,054.5		171,281.2
S _{ac}	Allowable contact stress number	psi	250,144.9		250,144.9
Z _N	Stress cycle factor for pitting resistance	e	0.8094		0.8201
H _B	Brinell hardness number		647		647
H_{BP}/H_{BG}	Pinion to gear hardness ratio			1	
А				0.000000	
С _н	Hardness ratio factor for pitting resistar	nce	1.0000	l.	1.0000
CT	Temperature factor		1.00	l.	1.00
C _R	Reliability factor		1.20	l.	1.20
FS _C	Factor of Safety for Contact		1.00	l	1.00
K _{ca}	Pitting Calibration Factor			1.000	

Scuffing (Scoring) Summary

Table 31: Spur-Helical Gear Scuffing

A. SCORING SUMMARY

<u>SYMBOL</u>	DESCRIPTION	UNITS	PINION	MESH	GEAR
t _{flmax}	Maximum flash temperature	°F		142.53	
t _{oil}	Oil temperature	°F		144.1333333	
t _M	Bulk temperature	°F		228.7766041	
t _{cmax}	Maximum contact temperature	°F		371.31	
	Type of Lubricant		V	ASCO MIL-L-23	699
μ_{tS}	Mean scuffing temperature	°F		459	
σ_{tS}	Standard temperature deviation	°F		31	
	Probability of scoring hazard Scuffing Risk			0.23%	
	Safety Factor			1.3860	



FLASH TEMPERATURE INDEX (DUDLEY/AMCP)

n, N	Number of teeth		39		55
ρ1, ρ2	Transverse radii of curvature at gener contact point	al in	2.3035		2.1618
		degrees		20.0000	
ϕ, ϕ_t	Standard traverse pressure angle	radians		0.3491	
Pd	Diametrical pitch (transverse)	teeth/in		3.6	
Zt	Geometry constant			0.004597605	
W _{te}	Effective tangential load	lb		12465.19157	

F_{e}	Effective face width	in	3.5
S	Mean surface finish	rms	18
n _p	Speed of pinion	rpm	2788
t _b	Gear body temperature	°F	158.2666667
t _{flash}	Flash temperature	°F	93.94686554
t _f	Flash temperature index	°F	252.2135
	Low Risk of Scoring (Dudley)		300
	High Risk of Scoring (Dudley)		350
	Risk of Scoring (Dudley)		LOW

Table 31: Spur-Helical Gear Scuffing (continued)

A.3.1 BASIC GEAR GEOMETRY

			PINION	MESH	GEAR
A _P , A _G	Type of gear (internal=-1)		1	1	1
n, N	Number of Teeth		39		55
m _G	Gear ratio			1.4103	
R	Standard (reference) pitch radius	in	5.4167		7.6389
Cr	Operating center distance	in		13.0556	
Rr	Operating pitch radius		5.4167		7.6389
		degrees		20.0000	
ϕ, ϕ_t	Standard traverse pressure angle	radians		0.3491	
R_{b1}, R_{b2}	Base radii	in	5.0900		7.1782
		degrees		20.0000	
φ _r	Operating transverse pressure angle	radians		0.3491	
pb	Transverse base pitch	in/tooth		0.8200	
p _N	Normal base pitch	in/tooth		0.8200	
p _x	Axial pitch			999999.9999	
		degrees		0.0000	
ψ_{b}	Base helix angle	radians		0.0000	
	• • • • •	degrees		0.0000	
ψ_r	Operating helix angle	radians		0.0000	
φ _{nr}	Operating normal pressure angle	degrees radians		20.0000 0.3491	
R_{o1}, R_{o2}	Outside radius	in	5.6944	0.0491	7.9167
••017••02		degrees	26.6384		24.9439
φ ₀₁ , φ ₀₂	Tip pressure angles	radians	0.4649		0.4354
	• • •				

A.3.2 DISTANCE ALONG THE LINE OF ACTION

C ₆		in	4.4653
C ₁	SAP	in	1.1266
C ₃	Operating pitch point	in	1.8526
C ₄	HPSTC	in	1.9466
C ₅	EAP	in	2.5532
C ₂	LPSTC	in	1.7331
Z	Length of line of contact	in	1.4266

A.3.3 PARAMETER ALONG THE LINE OF ACTION

n, N	Number of Teeth Linear coordinate in the transverse	39
Γ _A	plane on the line of action (SAP)	-0.3919
Г _в	Linear coordinate in the transverse plane on the line of action (LPSTC)	-0.0645
Γ _D	Linear coordinate in the transverse plane on the line of action (HPSTC)	0.0507
Γ_{E}	Linear coordinate in the transverse plane on the line of action (EAP)	0.3781

55

A.3.4 CONTACT RATIOS

m _p	Transverse contact ratio		1.7397
m _F	Axial contact ratio		0.0000
n _r	fractional part of mp		0.7397
n _a	fractional part of mF		0.0000
L_{min}	Minimum length of lines of contact	in	3.5000

A.3.5 ROLL ANGLES

		degrees	12.6811
ε ₁	Roll angle at C ₁	radians	0.2213
		degrees	19.5089
ε ₂	Roll angle at C ₂	radians	0.3405
		degrees	20.8540
8 ₃	Roll angle at C ₃	radians	0.3640
		degrees	21.9119
ε4	Roll angle at C ₄	radians	0.3824
		degrees	28.7396
£5	Roll angle at C_5	radians	0.5016

A.3.6 PROFILE OF RADII OF CURVATURE

ε Γ _i	Roll angle Parameter on line of action Transverse radii of curvature at general	degrees radians		25.9294 0.4526 0.2434 for max	flash
ρ ₁ , ρ ₂	contact point	in	2.3035		2.1618
ρ_r	Transverse relative radius of curvature	in		1.1152	
ρ_{rc}	Normal relative radius of curvature Equivalent radius of a cylinder that represents the gear pair curvatures in	in		1.0840	
ρ_n	contact along the line of action	in		1.1152	

A.4 GEAR TOOTH VELOCITIES AND LOADS

n _P , n _G	Speed of member	rpm	2,788.0		1,976.9
ω_1, ω_2	Rotational (angular) velocity	rad/s	292.0		207.0
V _{tr}	Operating pitchline velocity	fpm		7,907.2	
v _{r1} , v _{r2}	Rolling velocities	in/s	672.5260		447.5401
Vs	Sliding velocity	in/s		225.0	
Ve	Entraining velocity	in/s		1,120.1	
Р	Power	hp		2,660.0	
(W _{tr}) _{nom}	Nominal tangential load	lb		11,101.3	
K _a =K _o	Application/overload factor			1.25	
K _v	Dynamic factor			1.00	
K _m	Load distribution factor			1.12	
CD	Combined derating factor			1.40	
W _{tr}	Actual tangential load	lb		15,581.5	
W _{Nr}	Normal operating load	lb		16,581.5	
W _{tr}	Transverse unit load	lb		4,451.9	
W _{Nr}	Normal unit load	lb		4,737.6	

A.5 LOAD SHARING FACTOR

		degrees	25.9294
3	Roll angle	radians	0.4526
	Load sharing factor (unmodified tooth		
X_{Γ}	profiles)		0.4705
-	Load sharing factor (modified tooth		
X_{Γ}	profiles pinion driving)		0.4957
-	Load sharing factor (modified tooth		
X_{Γ}	profiles gear driving)		0.3528
	Load sharing factor (designed for		
X_{Γ}	smooth meshing)		0.6995
-	Type of tooth profile modification		Modified (pinion drives)
X_{Γ}	Load sharing factor		0.4957

A.6 HERTZIAN CONTACT BAND

X_{Γ}	Load sharing factor			0.4957	
W _{Nr}	Normal unit load	lb		4,737.6	
	Equivalent radius of a cylinder that represents the gear pair curvatures in contact along the line of action			4.4450	
ρ _n	contact along the line of action	in		1.1152	
ν ₁ , ν ₂	Poisson's ratio		0.300		0.300
E ₁ , E ₂	Modulus of elasticity	psi	29.64E+6		29.64E+6
Er	Reduced modulus of elasticity	psi		32.57E+6	
b _H	Semi-width of rectangular band	in		0.0143	

A.7.3.1 MEAN COEFFICIENT OF FRICTION

σ ₁ , σ ₂	Surface finish	rms	13		13
S	Average surface roughness	rms		13	
	Check on surface roughness			1.3514	
μ_{m}	Mean coeff of friction (approx)			0.0811	

A.7.4 THERMAL ELASTIC FACTOR

	Thermal elastic factor (martensitic		
X _M	steels)	°F lbs ^{-0.75} s ^{0.5} in ^{0.5}	1.7500

A.7.5 GEOMETRY FACTOR

X _G	Scoring Geometry Factor	0.1482
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A.7.2 FLASH TEMPERATURE EQUATION

t _{flmax} Maximum flash tempe	rature °F	142.5297
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Lubrication Analysis

Table 32: Spur-Helical Lubrication Analysis LUBRICATION ANALYSIS

<u>SYMBOL</u>	DESCRIPTION	UNITS	PINION	<u>MESH</u>	<u>GEAR</u>
HP	Power	hp	2,660.0		2,646.7
η_{mesh}	efficiency			<mark>99.5%</mark>	
Ploss	Power dissipated	hp	_	13.3	
Q	heat generated	Btu/min		563.92	
Cp	Specific heat of oil	Btu/lb-°F		0.5	
	Oil flow design		Reco	mmended (30°)
		lb/min		39.9	
М	Oil flow	gal		5.32	
		lb/min	-	114	
M _{manual}	Oil flow	gal		15.2	
		lb/min		25.1	
M _{min}	Minimum oil flow (ΔT =+45°F)	gal		3.3	
		lb/min		37.6	
M _{rec}	Recommended oil flow	gal		5.0	
		gpm/hp		0.002	
	Rule of thumb	gpm		5.3200	
ΔT	Temperature rise	°F		28.3	
t _{in}	Incoming oil temperature	°F		130	
t _{out}	Outgoing oil temperature	°F		158.3	
t _{oil}	Oil temperature (average)	°F		144.1	

Material Properties

DESCRIPTION	UNITS	VASCO X2M PINION	VASCO X2M GEAR	AISI 9310 V	Aero (Table 'ASCO X2MPY	Aero (Table 15) & Aero Carb (Table 1) VASCO X2MPYROWEAR 5: CBS600	-b (Table 1) CBS600	AISI 4340
AMS Spec		N/A	N/A	6265/6260	N/A	6308	6255	6414
Heat treatment		C-H	CH	H-O	C-H	C-H	C-H	N-HT
Main drive application		×	×	×	×	×		
Accessory application				×				×
High temperature application		×	×		×	×	×	
Case hardness (HRC)	HRC	62	62	61	62	62	60	50.5
Core hardness (HRC)	HRC	40	40	37	40	40	38	31
Surface hardness (HR15N)	HR15N							
Minimum Brinell hardness	뛰	647	647	632	647	647	617	488
Allowable contact stress number	psi	250,145	250,145	244,897	250,145	250,145	239,736	195,086
Allowable bending stres number	psi	51,990	51,990	52,102	51,990	51,990	52,149	49,966
Poisson's ratio		0.300	0.300	0.292	0.300	0.292	0.296	0.282
Young's moduli of elasticity	psi	29.64E+6	29.64E+6	29.00E+6	29.64E+6	30.00E+6	29.30E+6	29.50E+6
Density	lb/in ³	0.280	0.280	0.283	0.280	0.282	0.282	0.283
		2	2	~	7	с	4	

Table 33: Spur-Helical Gear Properties

GEAR MATERIALS

Table 33: Spur-Helical Gear Properties (continue)

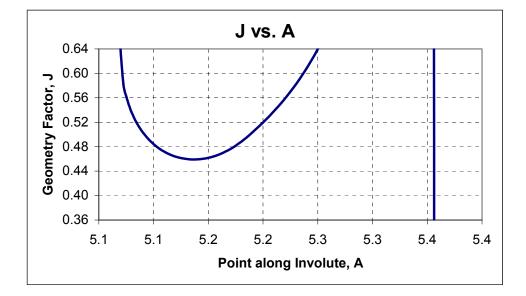


Bending Stress Geometry Factor

Table 34: Spur-Gear Bending Strength Geometry Factor for PinionJ0.4591

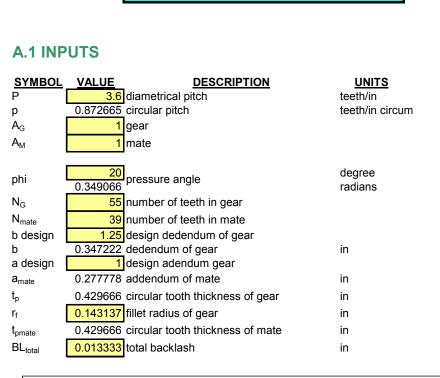
A.1 INPUTS

SYMBOL	VALUE	DESCRIPTION	<u>UNITS</u>
Р	3.6	diametrical pitch	teeth/in
р	0.872665	circular pitch	teeth/in circum
A _G	1	gear	
A _M	1	mate	
	-		
phi	20	pressure angle	degree
pin	0.349066		radians
N_{G}	39	number of teeth in gear	
N _{mate}	55	number of teeth in mate	
b design	1.25	design dedendum of gear	
b	0.347222	dedendum of gear	in
a design	1	design adendum gear	
a _{mate}	0.277778	addendum of mate	in
t _p	0.429666	circular tooth thickness of gear	in
r _f	0.143137	fillet radius of gear	in
t _{pmate}	0.429666	circular tooth thickness of mate	in
BL _{total}	0.013333	total backlash	in



A.2 GE	OMETRY FACTOR CALCULATION		
R	5.416667 pitch radius of gear	in	
R _r	5.069444 root radius of gear	in	
R _b	5.090002 base circle radius of gear	in	
R _{mate}	7.638889 pitch radius of mate	in	
R _{omate}	7.916667 outside radius of mate	in	
R _{bmate}	7.178208 base circle radius of mate	in	
Pb	0.820037 base pitch	in	
С	13.05556 center distance	in	
φ _n	20.92861 pressure angle at highest point of single	degrees	A.1
Ψn	0.365273 tooth contact on involute	radians	A.1
t _b '	0.55548 tooth thickness on base circule	in	A.2
φ _n	18.78552 angle which the normal force makes with a line perpendicular to the tooth centerline		A.3
	0.327869 at highest point of single tooth contact radius on tooth centerline to point of	radians	A.3
R _x	5.376399 application of worst load radius to tangency point of fillet and gear	in	A.4
A ₁	5.183607 tooth profile	in	A.6 & A.7
T ₁	10.90505 pressure angle at the intersection of the 0.190329 fillet and involute	degrees radians	A.6 & A.7
α'	0.02832		A.8
x'	0.146779 x coordinate of fillet-involute inflection	in	
y' delte	5.181529 y coordinate of fillet-involute inflection 0.218649	in	
delta aa	0.00705 x coordinate of fillet radius center	in	A.9
bb	5.212576 y coordinate of fillet radius center	in	A.8
А	5.137657		
α	0.024921		A.12 or A.13
h t	0.246691 0.571353		A.14
·	0.37 1333	LOW	A.15 5.06944444
J	0.127526	LOW R _r	
			5.06944444
J	0.127526	R _r	5.06944444 5.06944444
J X	0.127526 0.330823	R _r A ₁	5.06944444 5.06944444 5.18360708
J X K _t	0.127526 0.330823 1.976036	R _r A ₁ R _x	5.06944444 5.06944444 5.18360708 5.37639889

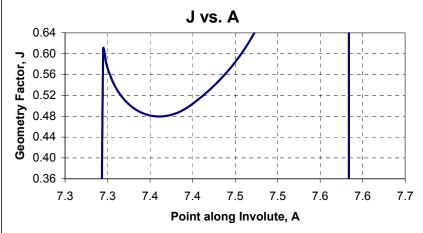
Table 34: Spur-Gear Bending Strength Geometry Factor for Pi (continued)



J

Table 35: Spur Gear Bending Strength Geometry Factor for Gear

0.4796



A.2 GEC	DMETRY FACTOR CALCULA	FION	
R	7.638889 pitch radius of gear	in	
R _r	7.291667 root radius of gear	in	
R _b	7.178208 base circle radius of gear	in	
R _{mate}	5.416667 pitch radius of mate	in	
R _{omate}	5.694444 outside radius of mate	in	
R _{bmate}	5.090002 base circle radius of mate	in	
Pb	0.820037 base pitch	in	
С	13.05556 center distance	in	
φ _h	20.83768 pressure angle at highest point of	of single degrees	A.1
1.11	0.363686 tooth contact on involute	radians	A.1
t _b '	0.617727 tooth thickness on base circule	in	A.2
φn	19.34243 angle which the normal force ma a line perpendicular to the tooth	centerline	A.3
	0.337589 at highest point of single tooth co radius on tooth centerline to point		A.3
R _x	7.607612 application of worst load radius to tangency point of fillet a	in and gear	A.4
A ₁	7.398814 tooth profile	in	A.6 & A.7
T ₁	14.02655 pressure angle at the intersectio	-	
α'	0.24481 fillet and involute 0.019103	radians	A.6 & A.7 A.8
x'	0.141329 x coordinate of fillet-involute infle	ection in	7.0
у'	7.397464 y coordinate of fillet-involute infle	ection in	
delta	0.263912		
aa bb	0.003148 x coordinate of fillet radius cente 7.434803 y coordinate of fillet radius cente		A.9 A.8
UU	7.454605 y coordinate of fillet radius cente	1 11	A.0
А	#NUM!		
α	#NUM!		A.12 or A.13
h t	#NUM! #NUM!		A.14 A.15
ı		LOW	#NUM!
J	#NUM!	R _r	7.29166667
Х	#NUM!	A ₁	7.39881417
K _t	#NUM!	R _x	7.60761221
Н	0.180044	HI	#NUM!
L	0.149972		
Μ	0.450028		

A.2 GEOMETRY FACTOR CALCULATION

APPENDIX C: BEVEL GEAR RATING CALCULATIONS

Bevel Gear Summary

Table 36: Bevel Gear User Inputs and Selections

BEVEL RATING SUMMARY

MDLCTR	<u>SYMBOL</u>	DESCRIPTION	<u>UNITS</u>	PINION	<u>MESH</u>	<u>GEAR</u>	WORKSHEET
		Selected Material		VASCO X2M		VASCO X2M	AGMA Stress
	l J	Geometry factor for pitting resistance Geometry factor for bending strength		0.2697	0.1187	0.2631	AGMA Stress AGMA Stress
BV_swc_P, BV_swc_G	S _{wc}	Permissible contact stress number	psi	212,984.3		227,546.6	AGMA Stress
BV_sc	s _c	Contact stress number	psi		207,070.9		AGMA Stress
BV_Bend_P, BV_Bend_G				1.03		1.10	AGMA Stress
BV_swt_P, BV_swt_G	s _{wt}	Permissible bending stress number	psi	37,936.1		27,286.1	AGMA Stress
BV_st_P, BV_st_G	s _t	Bending stress number	psi	26,157.9		26,811.0	
BV_Contact_P,	οι	Ŭ	por	,			
BV_Contact_G				1.45		1.02	AGMA Stress
BV_d_P, BV_D_0	Эd, D	Standard reference pitch diameter	in	6.291		18.874	AGMA Stress
BV_W_P, BV_W_G,							
BV_W_Mesh	W	Estimated weight	lb	39.6	395.8	356.2	AGMA Stress
		DESIGN BALANCE		0.7092		1.0797	AGMA Stress
		Load Face		Concave		Convex	Forces
BV_Hpmesh	HP_{mesh}	Power per tooth mesh			8,659.0		
BV_T_P, BV_T_0	Э Т _Р , Т _G	Torque	in lb	36,382.4		109,147.3	Forces
BV_PLV	Vt	Pitch line velocity	fpm		19,738.5		Forces
BV_Wt	W_{tP}, W_{tG}	Tangential force	lb		14,476.6		Forces
BV_Wx_P, BV_Wx_G BV_Wr_P,	W _x	Axial force	lb	11,650.5		2,896.8	Forces
BV_WI_F, BV_Wr_G	Wr	Radial force	lb	2,896.8		11,650.5	Forces
BV_mG BV_gamma,	m _G	Gear ratio			3.0000		
BV_ganna, BV_tau	γ, Γ	Pitch angle	degrees radians	18.4349 0.3218			Bevel Geometry Bevel Geometry

User Inputs and Selections

Table 36: Bevel Gear User Inputs and Selections (continued)

USER INPUTS & SELECTIONS

MDLCTR	SYMBOL	DESCRIPTION	UNITS	PINION	MESH	GEAR	WORKSHEET
BV_Kw	K _w	Weight factor for bevel gears	<u>onno</u>	<u></u> [0.25	02/41	TOTALONEET
BV Life	L	Life	hrs	3,600	0.20	3.600	AGMA Stress
BV_numberconta						_,	
cts	q	Number of contacts per revolution	contacts		1		AGMA Stress
BV_rms	f _P	Pinion surface roughness	μin	23.0			AGMA Stress
BV_Tt_P, BV_Tt_G	Τ _T	Peak operating gear blank temperature	°F	250.0000		250 0000	AGMA Stress
BV_FSc	S _H	Contact factor of safety	I	1.00			AGMA Stress
BV_Kac	Сн K _{ac}	Calibration factor to allow contact		1.00	1.0000	1.00	AGMA Stress
51_100	IN _{ac}	Reliability Requirement		Fewer than or		aspace 3 c d)	AGMA Stress
		Reliability Requirement		rewei than of		DSpace 5 S.U.)	AGIMA Stress
BV_Rev_P,							ſ
BV_Rev_G	Rev	Reverse loading factor		1.00			AGMA Stress
BV_FSb	S _F	Bending factor of safety		1.00		1.00	AGMA Stress
BV_Kac	K _{ab}	Calibration factor to allow bending			1.0000		AGMA Stress
BV_Ko	Ko	Overload factor			1.25		AGMA Stress
BV_Q	Q	Gear Quality Rating		12		12	AGMA Stress
		Amount of straddle mounting		One mem	ber straddle	mounted	AGMA Stress
		Properly crowned teeth		l	Yes		AGMA Stress
		Hand of Spiral		Right		Left	Forces
		Rotation		Counterclockw	ise	Clockwise	1 01065
		Power flow		Driver		Driven	Forces
		Load Face		Concave		Convex	Forces
BV_n_P, BV_n_G		Speed	rpm	15,000.0		5,000.0	Forces
BV_HPsys	P _P , P _G	Power	hp	8,659.0		8,659.0	
,.	· p, · G	i owei	ΠP	0,009.0		0,000.0	101003
BV_Pd	Pd	Diametrical pitch	teeth/in]	2.8611		Bevel Geometry
BV_NP, BV_NG	n, N	Number of teeth		18		54	Bevel Geometry
BV_ShaftAngle			degrees		90		Bevel Geometry
	Σ	Shaft angle	radians		1.5707963		Bevel Geometry
BV_F	F	Face	in		4		Bevel Geometry
BV_psi			degrees	l	35		Bevel Geometry
BV_phi	ψ	Spiral angle	radians	1	0.6108652	1	Bevel Geometry
DV_piii			degrees	l	20		Bevel Geometry
BV DesiredmC	φ, φ _n	Standard pressure angle (normal)	radians	r	0.3490659		Bevel Geometry
BV_DesiredmG		Desired m _G			3	2.864864865	Bevel Geometry
		Depth type for tooth taper			Duplex		Bevel Geometry
BV_rcmanual		Cutter radius (Manual)	in		ecommende 4.5	a	Bevel Geometry Bevel Geometry
DV_ICITATION		Type of cutting process			Face milling		Bevel Geometry
							_ 5.0. Coomony
				VASCO X2M		VASCO X2M	Material
		Gear Material		VASCO AZIVI		VASCO AZIVI	Material

AGMA Stress Equations

Table 37: Bevel Gear AGMA Stress Equations

AGMA STRESS ANALYSIS FOR BEVEL GEARS

SUMMARY

		UNITS	PINION	MESH	GEAR
	Selected Material		VASCO X2M		VASCO X2M
I	Geometry factor for pitting resistance			0.1187	,
J	Geometry factor for bending strength		0.2697		0.2631
S _{wc}	Permissible contact stress number	psi	212,984.3		227,546.6
Sc	Contact stress number	psi		207,070.9)
			1.03		1.10
s _{wt}	Permissible bending stress number	psi	37,936.1		27,286.1
st	Bending stress number	psi	26,157.9		26,811.0
			1.45		1.02
d, D	Standard reference pitch diameter	in	6.2913		18.8739
	Estimated weight	lb	39.5803		356.2226
	DESIGN BALANCE		0.7092		1.0797

FORCE ANALYSIS

n _P , n _G	Speed	rpm	15,000.0	5,000.0
P_P, P_G	Power	hp	8,659.0	8,659.0
		ft lb	3,031.9	9,095.6
T_P, T_G	Torque	in lb	36,382.4	109,147.3
d _m , D _m	Mean pitch diameter	in	5.026	15.079
Vt	Pitch line velocity	fpm		19,738.5
W_{tP}, W_{tG}	Tangential force	lb		14,476.6
W _x	Axial force	lb	11,650.5	2,896.8
Wr	Radial force	lb	2,896.8	11,650.5
W	Total force	lb	16,729.0	15,190.8

GEAR GEOMETRY

n, N	Number of teeth		18	54
d, D	Pitch diameter	in	6.2913	18.8739
P _d	Diametrical pitch (outer transverse)	teeth/in	2.8611	
r _c	Cutter radius	in	6.4808	
A _m	Mean cone distance	in	7.9474	
		degrees	35	
Ψ	Spiral angle	radians	0.61086524	
F	Face	in	4	

PERMISSIBLE CONTACT STRESS

S _{wc}	Permissible contact stress number	psi	212,984.3	227,546.6
S ac	Allowable contact stress number	psi	250,144.9	250,144.9
CL	Stress cycle factor		0.9318	0.9955
L	Life	hrs	3,600	3,600
n	Speed	rpm	15,000.0	5,000.0
q	Number of contacts per revolution	contacts		1
N_{L}	Number of stress cycles	cycles	3.240E+09	1.080E+09
C _H	Hardness ratio factor		1.0000	1.0000
H_{BP}, H_{BG}	Minimum Brinell hardness		647	647
B ₁	Intermediate variable		(0.0007
f _P	Pinion surface roughness	μin	23.0	
B2	Intermediate variable			0.0006
Κ _T	Temperature factor		1.0000	1.0000
	Peak operating gear blank			
Τ _Τ	temperature	°F	250.0000	250.0000
	Required reliability		Fewer than one in 80	0 (Aerospace 3 s.d.)
C _R	Reliability factor			1.09
S _H	Contact safety of factor		1.00	1.00
C _f	Pitting resistance derating factor		0.8514	0.9097
K_{ac}	Calibration factor to allow contact			1.0000

PERMISSIBLE BENDING STRESS

s _{wt}	Permissible bending stress number	psi	37,936.1	27,286.1
S at	Allowable bending stress number	psi	51,990.1	51,990.1
KL	Stress cycle factor		0.8739	0.8979
L	Life	hrs	3,600	3,600
n	Speed	rpm	15,000.0	5,000.0
q	Number of contacts per revolution	contacts		1
N_L	Number of stress cycles	cycles	3.240E+09	1.080E+09
KT	Temperature factor Peak operating gear blank		1.0000	1.0000
Τ _T	temperature	°F	250.0000	250.0000

Table 37: Bevel Gear AGMA Stress Equations (continued)

К _R	Reliability Requirement Reliability factor	Fewer than one in 800 (Aerospace 3 s.d.) 1.20
n_manual	Manual entry for standard deviations	0.0000
n R_el	Number of standard deviations Desired reliability	3.0000 0.9987
V	Coefficient of variation (bending)	0.1560
Rev S _F	Reverse loading factor Bending safety of factor	1.00 1.00 1.00
K _f	Bending strength derating factor	0.7297 0.5248
K_{ab}	Calibration factor to allow bending	1.0000

CONTACT STRESS FORMULA

Sc	Calculated contact stress number	psi		207,070.9	
Cp	Elastic coefficient	(lb/in ²) ^{0.5}	_	2,276.7	
μ _P , μ _G	Poisson's ratio		0.300		0.300
E _P , E _G	Young's moduli of elasticity	psi	29.6E+06		29.6E+06
Cs	Size factor for pitting resistance			0.9375	
F	Face width	in		4.0000	
Τ _Ρ	Operating pinion torque	in Ib	36,382.4		109,147.3
Ko	Overload factor		Γ	1.25	
K _v	Dynamic factor		-	1.0500	
Q	Gear Quality Rating		12		12
Q _v	Transmission Quality Rating			12	
В	Intermediate coefficient			0	
A	Intermediate coefficient			106	
d, D	Pitch diameter	in	6.2913		18.8739
n	Speed	rpm	15,000.0		5,000.0
v _t	Pitch line velocity (outside edge)	fps		24,724.8	
V _{tmax}	Maximum pitch line velocity	fps		13,225.0	
F	Net face width	in		4.0000	
	Amount of straddle mounting		One mem	ber straddle m	ounted
K _m	Load distribution factor			1.1576	
K _{mb}	Load distribution modifier		_	1.1000	
	Properly crowned teeth			Yes	
C _{xc}	Crowning factor			1.5000	
I	Pitting resistance geometry factor			0.1187	

Table 37: Bevel Gear AGMA Stress Equations (continued)

BENDING STRESS FORMULA

st	Calculated bending stress number	psi	26,157.9	26,811.0
Ko	Overload factor		1.25	
K _v	Dynamic factor		1.05	
P _d	Diametrical pitch (outer transverse)	teeth/in	2.8611	
Ks	Size factor		0.56	
	Amount of straddle mounting		One member straddle r	nounted
K _m	Load distribution factor		1.16	
K _x	Tooth lengthwise curvature factor		1.0000	
r _c	Cutter radius	in	6.4808	
A _m	Mean cone distance	in	7.9474	
		degrees	35	
ψ	Spiral angle	radians	0.6109	
q	Intermediate coefficient		-1.1557	
J	Bending strength geometry factor		0.2697	0.2631
Τ _Ρ	Operating pinion torque	in lb	36,382.4	109,147.3
F	Net face width	in	4.0000	
d, D	Pitch diameter	in	6.2913	18.8739

Force Analysis

Table 38: Bevel Gear Force Analysis BEVEL GEAR FORCE ANALYSIS

Distant have d		Rotation	Rotation	Load	face
Pinion hand of spiral	Gear hand of spiral	of driver	of driven	Driver	Driven
Right	Left	Clockwise	Counterclo	Convex	Concave
Right	Len	Counterclo	Clockwise	Concave	Convex
Left	Right	Clockwise	Counterclo	Concave	Convex
Leit	Kigiit	Counterclo	Clockwise	Convex	Concave

			PINION	GEAR
	Hand of Spiral Rotation Power flow Load Face		Right Counterclocky Driver Concave	Left <mark>wise</mark> Clockwise Driven Convex
n, N	Number of teeth	teeth	18	54
n _P , n _G	Speed	rpm	15,000.0	5,000.0
P _{Psys} , Pgsys	Power for gear system	hp	8,659.0	8,659.0
q	Number of contacts per re-	vo contacts		1
P_P, P_G	Power per mesh	hp	8,659.0	8,659.0
		ft lb	3,031.9	9,095.6
T _P , T _G	Torque	in lb	36,382.4	109,147.3
d _m , D _m	Mean pitch diameter	in	5.026376	15.07913
Vt	Pitch line velocity	fpm		19,738.5
W_{tP}, W_{tG}	Tangential force	lb		14,476.6
	Axial force (concave)	lb	11,650.5	9,307.7
	Axial force (convex)	lb	-7,582.4	2,896.8
W _x	Axial force	lb	11,650.5	2,896.8
	Radial force (concave)	lb	2,896.8	-7,582.4
	Radial force (convex)	lb	9,307.7	11,650.5
W _r	Radial force	lb	2,896.8	11,650.5
W	Total force	lb	16,729.0	15,190.8

Bevel Gear Geometry

Table 39: Bevel Gear Geometry

BEVEL DESIGN INPUTS

			PINION		GEAR	
P _d	Diametrical pitch	teeth/in		2.8611		
n, N	Number of teeth		18			54
		degrees		90		
Σ	Shaft angle	radians		1.570796	_	
F	Face	in		4		
		degrees		35		
Ψ	Spiral angle	radians		0.610865	_	
		degrees		20		
φ, φ _n	Standard pressure angle (normal)	radians		0.349066	-	
	Desired m _G			3		

BEVEL GEAR DESIGN FORMULAS

m _G	Gear ratio			3	
d, D	Pitch diameter	in	6.291287		18.87386
		degrees	18.43495		71.56505
γ, Γ	Pitch angle	radians	0.321751		1.249046
A _o	Outer cone distance	in		9.947397	
A _m	Mean cone distance	in		7.947397	
k ₁	Depth factor			2	
h	Mean working depth	in		0.457485	
k ₂	Clearance factor			0.125	
С	Clearance	in		0.057186	
h _m	Mean whole depth	in		0.514671	
m ₉₀	Equivalent 90° ratio			3	
C ₁	Mean addendum factor			0.242222	
p _m	Mean circular pitch	in/teeth		0.877268	
a _P . a _G	Mean addendum	in	0.346672		0.110813
b _P , b _G	Mean dedendum	in	0.167999		0.403858

	Depth type for tooth taper			Duplex	
	Sum of dedendum angles	degrees		4.120045	
$\Sigma \delta_{\mathbf{S}}$	(Standard)	radians		0.071908	
20	Sum of dedendum angles	degrees		0	
Σδυ	(Uniform)	radians		0 3.14617	
$\Sigma \delta_{D}$	Sum of dedendum angles (Duplex)	degrees radians		0.054911	
_00	Cam of acachadin angles (Duplex)	degrees		3.14617	
$\Sigma \delta_{T}$	Sum of dedendum angles (TRL)	radians		0.054911	
		degrees		3.14617	
Σδ	Sum of dedendum angles	radians		0.054911	
			Re	ecommende	d
	Cutter radius (Manual)	in in	l	4.5 5 60905	
	Cutter radius (Uniform) Cutter radius (Duplex)	in in		5.69805 6.480841	
r _c	Cutter radius	in		6.480841	
10		degrees	1.210985	0.400041	2.90906
d _{PS} , δ _{GS}	Dedendum angles (Standard)	radians	0.021136		0.050773
	<u> </u>	degrees	0		0
$\delta_{\text{PU}},\delta_{\text{GU}}$	Dedendum angles (Uniform)	radians	0		0
		degrees	0.762072		2.384098
$\delta_{\text{PD}}, \delta_{\text{GD}}$	Dedendum angles (Duplex)	radians	0.013301		0.04161
8 8	Dedendum angles (TDL)	degrees	0.762072		2.384098
$\delta_{\text{PT},} \delta_{\text{GT}}$	Dedendum angles (TRL)	radians	0.013301		0.04161
δ _P , δ _G	Dedendum angles	degrees radians	0.762072 0.013301		2.384098 0.04161
0p, 0G	Dedendum angles	degrees	20.81905		72.32712
γ ₀ , Γ ₀	Face angle	radians	0.363361		1.262346
	5	degrees	17.67288		69.18095
γ_{R}, Γ_{R}	Root angle	radians	0.30845		1.207435
a_{oP}, a_{oG}	Outer addendum	in	0.429941		0.137416
b _{oP} , b _{oG}	Outer dedendum	in	0.194602		0.487126
h _k	Outer working depth	in		0.567357	
h _t	Outer whole depth	in		0.624542	
d _o , D _o	Outside diameter	in	7.107042		18.96077
x _o , X _o	Pitch cone apex to crown	in	9.300971		3.015279
P _{dm}	Mean diametrical pitch	teeth/in		3.581109	
d _m , D _m	Mean pitch diameter	in	5.026376		15.07913
k ₃	Thickness factor			0.1136	
- •3	Mean normal circular thickness			0.1100	
t _n , T _n	theoretical without backlash		0.471139		0.247477
B	Outer normal backlash allowance			0.01	
	Type of cutting process			ace milling	
		degrees		47.37766	
ψ_{omilling}	Outer spiral angle (face milling)	radians		0.826896	

Table 39: Bevel Gear Geometry (continued)

Nc	Number of crown gear teeth	teeth		56.921	
Ns				5	
		degrees		5.062248	
V	Lead angle of cutter	radians		0.088353	
		degrees		60.06225	
λ	First auxiliary angle	radians		1.048284	
S1	Center distance: crown gear to cu	itt in		7.331682	
		degrees		14.9963	
η1	Second auxiliary angle	radians		0.261735	
	Lengthwise tooth mean radius of				
ρ	curvature	in		6.164631	
Q	Intermediate variable			6.739663	
		degrees		40.62781	
ηο	Intermediate angle	radians		0.709089	
		degrees		47.75562	
Ψ ohobbing	Outer spiral angle (face hobbing)	radians		0.833493	
Ū		degrees		47.37766	
ψ_{o}	Outer spiral angle	radians		0.826896	
t _{nc} , T _{nc}	Mean normal chordal thickness	in	0.466935		0.243952
a _{cP} , a _{cG}	Mean chordal addendum	in	0.357146		0.111134
	Transverse contact ratio	-	1.190935		
m _p			1.190955		

Table 39: Bevel Gear Geometry (continued)

BEVEL UNDERCUT CHECK

A _{iG}	Inner cone distance		5.947397
	Inner gear spiral angle (straight	degrees	0
Ψ iGspiral	bevel)	radians	0
	Inner gear spiral angle (face	degrees	23.95187
ΨiGmilling	milling)	radians	0.418039
		degrees	57.29215
η _i	Gear offset angle at inside	radians	0.999937
		degrees	22.12453
Ψ iGhobbing	Inner gear spiral angle (hobbing)	radians	0.386146
		degrees	23.95187
ψ_{iG}	Inner gear spiral angle	radians	0.418039
		degrees	22.81897
φті	Inner transverse pressure angle	radians	0.398266
b _{ilP}	Limit inner dedendum	in	0.298173
b _{i₽}	Inner dedendum	in	0.141396

Material Properties

		i TH	Steel F/IH	F/IH		Steel C-H		
		Steel TH	Steel F/IH	Steel F/IH	Steel C-H	Steel C-H	Steel C-H	Nitralloy 135M
		(Grade 2)	(Grade 1)	(Grade 2)	(Grade 1)	(Grade 2)	(Grade 3)	(Grade 2)
	AMS Spec	AGMA Class 5						
	Heat treatment	ΗT	F/IH	F/IH	Ч-О	Ч- С	Ч-О О	Nitrided
	Main drive application						×	
	Accessory application	×		×	×	×	×	×
	High temperature applicati	on						
	Case hardness (HRC)	43	50	50	59.5	61	61	60
	Core hardness (HRC)			28	21	25	30	
	Surface hardness (HR15N)							06
H _{BP} , H _{BG}	Minimum Brinell hardness	400	482	482	610	632	632	614
	Allowable contact stress							
Sac	number	175,000	175,000	190,000	200,000	225,000	250,000	145,000
sat	Allowable bending stres number	umber 25,180	12,500	13,500	30,000	35,000	40,000	24,000
μь, μс	Poisson's ratio	0.300	0.300	0.300	0.300	0.300	0.300	0.291
E _P , E _G	Young's moduli of elasticity	Y 30.00E+6	30.00E+6	30.00E+6	30.00E+6	30.00E+6	30.00E+6	30.00E+6
٩	Density	0.282	0.282	0.282	0.282	0.282	0.282	0.280

Table 40: Bevel Gear Material Selection

MATERIAL SELECTION

Bending Strength Geometry Factor

Table 41: Bevel Gear Bending Strength Geometry Factor

GEOMETRY FACTOR FOR BENDING STRENGTH (J)

		J	0.2697	0.2631
			0.1344	0.0931
INPU	VARIABLES		PINION	GEAR
P _d n, N	Diametrical pitch Number of teeth	teeth/in	2.86	
11, I N		degrees		54 90
Σ	Shaft angle	radians	1.570)8
F	Face	in degrees		4
Ψ	Spiral angle	radians degrees	0.610)9 20
φ, φ _n	Standard pressure angle (normal)	radians	0.349	91

C.1.3.1 INITIAL DATA

			PINION	GEAR
Ao	Outer cone distance	in		9.9474
a _{oP} , a _{oG}	Outer addendum	in	0.4299	0.1374
d, D	Pitch diameter	in	6.2913	18.8739
n, N	Number of teeth		18	54
		degrees	18.4349	71.5651
γ, Γ	Pitch angle	radians	0.3218	1.2490
		degrees	20.8190	72.3271
γ ₀ , Γ ₀	Face angle	radians	0.3634	1.2623

C.2.4.1 INITIAL DATA

b _{oP} , b _{oG}	Outer dedendum	in	0.1946016	0.4871264 Table 9
k _r	Tool edge radii factor		0.3	0.3
r_{TP}, r_{TG}	Tool edge radii	in	0.1048548	0.1048548
	Mean backlash per tooth Mean normal circular thickness	in	0.0039947	0.0039947
t _n , T _n	theoretical without backlash		0.4711387	0.2474771
	Mean normal circular thickness with			
t _P , t _G	backlash	in	0.467144	0.2434824
		degrees	0.7620723	2.3840978
$δ_P, δ_G$	Dedendum angles	radians	0.0133007	0.0416104
b_P, b_G	Mean dedendum	in	0.1679987	0.4038577

 Table 41: Bevel Gear Bending Strength Geometry Factor (continued)

C.2.4.1 DETERMINATION OF POINT OF LOAD APPLICAION FOR MAX BENDING STRESS

m _o	Modified contact ratio			3.459168866	
p _N	Mean normal base pitch Length of action within the contact	in		0.6753	
η	ellipse Assumed locations of critical point on	in		2.3011	
f _J	tooth for bending stress Length of action within the contact	in		0	
η_{J}	ellipse for bending Length of action in mean normal			2.301052827	
Z _N	section	in degrees		1.1335 32.6146	
Ψb	Mean base spiral angle	radians		0.5692	
k'	Location constant Load Face		Concave	0.1471	Convex
	Location of points of load application on path of action (straight bevel)		Concave	0.5668	CONVEX
	Location of points of load application on path of action (concave)			0.7229	
	Location of points of load application on path of action (convex) Location of points of load application			0.4106	
р _{3Р} , р _{3G}	on path of action	in	0.7229		0.4106
	Factors used for calcs (straight) Factors used for calcs (concave) Factors used for calcs (convex)			0.289765129 0.205586789 0.205586789	
x ₀" _P , x0 " _G	Factors used for pinion/gear calcs		0.2055868		0.2055868
r_N, R_N	Mean normal pitch radius	in	3.9480		35.5318
ΣR_N	Sum of mean normal pitch radii	in		39.4798	
r_{bN}, R_{bN}	Mean normal base radius	in	3.7099		33.3890
r _{oN} , R _{oN}	Mean normal outside radius	in	4.2946		35.6426
ΦLP, ΦLG	Normal pressure angles at point of load application	degrees radians	27.0725 0.4725		20.1630 0.3519
YLF, YLG	Rotation angles used in bending	degrees	2.0312		0.1745
θ_{hP}, θ_{hG}	strength calcs	radians	0.0355		0.0030
		degrees	25.0413		19.9885
ϕ_{hP}, ϕ_{hG}	Distance from pitch circle to point of	radians	0.4371		0.3489
Δr_N , ΔR_N	load application on tooth centerline	in	0.1468		-0.0026
r, R	Mean transverse pitch radius	in	2.6491		23.8422
A _m	Mean cone distance Mean transverse radius to point of	in		7.9474	
r _t , R _t	load application	in	2.8645		24.4564

C.2.4.5 TOOTH FILLET RADIUS AT ROOT CIRCLE

r _{fP} , r _{fG}	Fillet radius at root of tooth	in	0.1058	0.1073
110				

C.2.4.6 TOOTH FORM FACTOR

Y _P , Y _G	Tooth form factors excluding stress concentration factors	0.587038	0.3530
t _{np} , t _{ng}	One-half tooth thickness at critical section	0.2661	0.2345
h _{NP} , h _{NG}	Load heights from critical section	0.2716	0.3254

C.2.4.7 STRESS CONCENTRATION AND STRESS CORRECTION F#

Н	Empirical exponent	0.18	000
L	Empirical exponent	0.15	500
М	Empirical exponent Stress concentration and stress	0.45	600
K_{fP},K_{fG}	correction factors	1.9047	1.6504

C.2.4.8 TOOTH FORM FACTOR

$\mathbf{Y}_{\mathrm{KP}},\mathbf{Y}_{\mathrm{KG}}$	Tooth form factor	0.3082	0.2139

C.2.4.9 LOAD SHARING RATIO

f _J	Assumed locations of critical point on tooth for bending stress Length of action within the contact	in	0.000000
$\eta_{\rm J}$	ellipse for bending		1.133500
p _N	Mean normal base pitch	in	0.675278
	Term A (k=1)		0
	Term A (k=2)		0
	Term B (k=1)		0
	Term B (k=2)		0
η _J ' ³			1.4563
m _{NJ}	Load sharing ratio		1.0000

C.2.4.10 INERTIA FACTOR

Ci	Inertia factor for pitting resistance	1
Ki	Inertia factor for bending strength	1

Table 41: Bevel Gear Bending Strength Geometry Factor (continued)

C.2.4.11 EFFECTIVE FACE WIDTH

0	Projected length of instantaneous line of contact in the lengthwise direction of the tooth	1.39	7990168
$\Delta F'_{TP}, \Delta F'_{TG}$ T	oe increment	1.8392089	1.8392089
$\Delta F'_{HP}, \Delta F'_{HG}$ H	leel increment	1.3372586	1.3372586
$\Delta F_{TP}, \Delta F_{TG}$		1.8392089	1.8392089
$\Delta F_{HP}, \Delta F_{HG}$		1.3372586	1.3372586
F _{eP} , F _{eG} E	Effective face width	2.0197304	2.125104

C.2.4 FORMULA FOR GEOMETRY FACTOR J

Y_{KP}, Y_{KG}	Tooth form factor		0.3082		0.2139
m _{NJ}	Load sharing ratio			1.0000	
K _i	Inertia factor for bending strength Mean transverse radius to point of			1	
r _t , R _t	load application	in	2.8645		24.4564
r, R	Mean transverse pitch radius	in	2.6491		23.8422
F_{eP}, Fe_{G}	Effective face width		2.0197304		2.125104
F	Face width			4	
Pd	Diametrical pitch	teeth/in		2.8611	
Pm	Mean transverse diametrical pitch	in ⁻¹	-	3.5811	

REGRESSION WORK

ΣShaft angleψSpiral angle ϕ, ϕ_n Standard pressure angle (normal)Straight Bevel (ψ=0, φ=20, Σ=90) Nmate<=50Straight Bevel (ψ=0, φ=20, Σ=90) 50 <nmate<60< td="">Straight Bevel (ψ=0, φ=20, Σ=90) Nmate>60Straight Bevel (ψ=0, φ=20, Σ=90) Nmate>60</nmate<60<>	degrees radians degrees radians degrees radians Nmate Ngear 54	0. 0.	90 .5708 35 .6109 20 .3491
$ \phi, \phi_n $ Standard pressure angle (normal) Straight Bevel (ψ=0, φ=20, Σ=90) Nmate<=50 Straight Bevel (ψ=0, φ=20, Σ=90) 50 <nmate<60 (ψ="0," bevel="" nmate="" straight="" σ="90)" φ="20,">60</nmate<60>	radians degrees radians Nmate Ngear	0.	.6109 20
$ φ, φ_n $ Standard pressure angle (normal) Straight Bevel (ψ=0, φ=20, Σ=90) Nmate<=50 Straight Bevel (ψ=0, φ=20, Σ=90) 50 <nmate<60 Straight Bevel (ψ=0, φ=20, Σ=90) Nmate>60</nmate<60 	degrees radians Nmate Ngear	0.	20
Straight Bevel (ψ=0, ϕ =20, Σ=90) Nmate<=50 Straight Bevel (ψ=0, ϕ =20, Σ=90) 50 <nmate<60 Straight Bevel (ψ=0, ϕ=20, Σ=90) Nmate>60</nmate<60 	radians Nmate Ngear		.3491
Straight Bevel (ψ =0, ϕ =20, Σ =90) 50 <nmate<60 Straight Bevel (ψ=0, ϕ=20, Σ=90) Nmate>60</nmate<60 	J		
Straight Bevel (ψ =0, ϕ =20, Σ =90) 50 <nmate<60 Straight Bevel (ψ=0, ϕ=20, Σ=90) Nmate>60</nmate<60 	54		Ngear
Straight Bevel (ψ =0, ϕ =20, Σ =90) 50 <nmate<60 Straight Bevel (ψ=0, ϕ=20, Σ=90) Nmate>60</nmate<60 		18	18 54
Straight Bevel (ψ =0, ϕ =20, Σ =90) Nmate>60	0.2632		0.1939
			0.2110
	0.2382		0.2366
Straight Bevel (ψ =0, ϕ =20, Σ =90) Selected	0.2532		0.1939
Straight Bevel (ψ =0, ϕ =25, Σ =90) Nmate<=50	0.2546		0.2174
Straight Bevel (ψ =0, ϕ =25, Σ =90) 50 <nmate<60< td=""><td></td><td></td><td>0.1280</td></nmate<60<>			0.1280
Straight Bevel (ψ =0, ϕ =25, Σ =90) Nmate>60	0.2397		0.2454
Straight Bevel (ψ =0, ϕ =25, Σ =90) Selected	0.2486		0.2174
Straight Bevel (ψ =0, ϕ =20 to 25, Σ =90) Selected	0.2532		0.1939
		Ψ	
	0.0007	35	0.0004
Spiral Bevel (ψ=15,25,35, φ=20, Σ=90)	0.2697		0.2631
Straight/Spiral for Σ =90	0.2697		0.2631
Spiral Poyel ($w=25$, $b=20$, $\Sigma=60$)	0.2824		0 0777
Spiral Bevel (ψ =35, ϕ =20, Σ =60)	0.2824 0.2697		0.2777 0.2631
Spiral Bevel (ψ =35, ϕ =20, Σ =60-90) Spiral Bevel (ψ =35, ϕ =20, Σ =90)			0.2631
Spirai Devel (ψ-35, ψ-20, 2-90)	0 2607		
Spiral Bevel (ψ =35, ϕ =20, Σ =60-90) Selected	0.2697		0.2001
Selected J	0.2697 0.2697		0.2631

<u>Pitting Resistance Geometry Factor</u>

Table 42: Bevel Gear Pitting Resistance Geometry Factor

GEOMETRY FACTOR FOR PITTING RESISTANCE (I)

				0.1187	
P _d	Diametrical pitch	teeth/in		2.8611	
n, N	Number of teeth		18		54
		degrees		90	
Σ	Shaft angle	radians		1.5708	
F	Face	in		4	
		degrees		35	
Ψ	Spiral angle	radians		0.6109	
		degrees		20	
φ, φ _n	Standard pressure angle (normal)	radians		0.3491	
	fmanual			-0.208532912	

INITIAL DATA

			PINION	GEAR
Ao	Outer cone distance	in		9.947397484
a _{oP} , a _{oG}	Outer addendum	in	0.4299	0.1374
d, D	Pitch diameter	in	6.2913	18.8739
n, N	Number of teeth		18	54
		degrees	18.4349	71.5651
γ, Γ	Pitch angle	radians	0.3218	1.2490
		degrees	20.8190	72.3271
γ _ο , Γ _ο	Face angle	radians	0.3634	1.2623

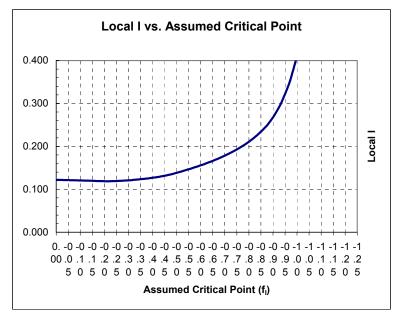
INITIAL FORMULAS

A _m	Mean cone distance	in		7.9474	
		degrees	2.3841		0.7621
α_{P}, α_{G}	Addendum angle	radians	0.0416		0.0133
а _Р . а _G	Mean addendum	in	0.3467		0.1108
k'	Location constant			0.1471	
Pm	Mean transverse diametrical pitch	in ⁻¹		3.5811	
р	Outer transverse circular pitch	in		1.0980	
p _N	Mean normal base pitch	in		0.6753	
p _n	Mean normal circular pitch	in		0.7186	
p ₂	Auxiliary pitch	in		0.9518	

r, R	Mean transverse pitch radius	in	2.6491		23.8422
r _N , R _N	Mean normal pitch radius	in	3.9480		35.5318
r_{bN}, R_{bN}	Mean normal base radius	in	3.7099		33.3890
r _{oN} , R _{oN}	Mean normal outside radius	in	4.2946		35.6426
Z'_{P}, Z'_{G}	Length of mean normal addendum Length of action in mean normal	in	0.8132		0.3203
Z _N	section	in		1.1335	
m _p	Transverse contact ratio			1.1909	
Kz	Intermediate variable			0.5373	
m _F	Face contact ratio			3.2477	
m _o	Modified contact ratio			3.4592	
		degrees		32.6146	
ψ_{b}	Mean base spiral angle Length of action within the contact	radians		0.5692	
η	ellipse	in		2.3011	
ρ_P, ρ_G	curvature at pitch circle	in	1.2770		11.4934

Table 42: Bevel Gear Pitting Resistance Geometry Factor (continued)

DETERMINATION OF PROFILE RADIUS OF CURVATURE AT CRITICAL POINT



Ci	Inertial factor Assumed location of critical point			1.0000
f _l	on tooth (straight bevel) Local length of action within the	in	-0.	208532912
η _ι	contact ellipse (straight bevel) Distance along path of action in mean normal section from pitch line t point of maximum stress	in		2.2629
Zo	contact (straight bevel)	in		0.3642
	Profile radius of curvature at point			
ρ_1,ρ_2	f _I (straight bevel)	in	1.6412	
ρ _o	Relative radius of profile curvature Length of line of contact (straight	in		1.4303
S	bevel)	in		1.6322
Term 1k1	Intermediate value (straight bevel)			9.3034
Term 1k2	Intermediate value (straight bevel)			0.0218
	Intermediate value (straight bevel)			3.1974
η' <mark>'</mark> 3	Cube of local η_i (straight bevel)			24.1108
m _{NI}	Load sharing ratio (straight bevel)			0.4806
I	Geometry Factor (straight bevel)			0.1187
I	Geometry Factor (spiral bevel)			0.1187

Table 42: Bevel Gear Pitting Resistance Geometry Factor (continued)

APPENDIX C: SHAFT DESIGN CALCULATIONS

Summary of Results

Table 43: Shafting Summary of Results

SHAFTING SUMMARY OF RESULTS

SHAFT D	D	Outside diameter	in	8.55
SHAFT di	d	Inside diamter	in	7.70
SHAFT_fb	f _b	Vibratory bending stress	psi	12275.73
_			psi	
SHAFT_nb	n _b	Bending factor of safety		1.16
SHAFT_Fts	f _s	Torsional shear stress	psi	10,023.6
SHAFT_nts	n _s	Torsional shear factor of safety		3.92
SHAFT_fa	f _a	Axial tension stress	psi	0.00
SHAFT_na	n _a	Axial tension factor of safety		0.00
SHAFT_W	W_{shaft}	Shaft weight	lb	14.98
SHAFT_MS	MS	Margin of Safety		1.00
		PAS	S	
		Type of Shaft Analysis		0
SHAFT n	n	Type of Shaft Analysis Shaft speed	rpm	0 476.3
SHAFT_n SHAFT_Nc	n N _c	Type of Shaft Analysis Shaft speed Critical speed	rpm rpm	•
_	N _c	Shaft speed	•	476.3
SHAFT_Nc	N _c b N _{sub}	Shaft speed Critical speed	rpm	476.3 278,752.2
SHAFT_Nc SHAFT_Ncsu	N _c b N _{sub}	Shaft speed Critical speed Subcritical avoid speed	rpm rpm	476.3 278,752.2 195,126.5
SHAFT_Nc SHAFT_Ncsu SHAFT_Ncsu	N _c b N _{sub}	Shaft speed Critical speed Subcritical avoid speed Supercritical avoid speed Type of operation	rpm rpm rpm	476.3 278,752.2 195,126.5 306,627.4 0
SHAFT_Nc SHAFT_Ncsu SHAFT_Ncsu	N _c b N _{sub} p N _{sup}	Shaft speed Critical speed Subcritical avoid speed Supercritical avoid speed Type of operation	rpm rpm	476.3 278,752.2 195,126.5 306,627.4 0
SHAFT_Nc SHAFT_Ncsu SHAFT_Ncsu SHAFT_Type	N _c b N _{sub} p N _{sup}	Shaft speed Critical speed Subcritical avoid speed Supercritical avoid speed Type of operation	rpm rpm rpm	476.3 278,752.2 195,126.5 306,627.4 0
SHAFT_Nc SHAFT_Ncsu SHAFT_Ncsu SHAFT_Type	N _c b N _{sub} p N _{sup}	Shaft speed Critical speed Subcritical avoid speed Supercritical avoid speed Type of operation	rpm rpm rpm	476.3 278,752.2 195,126.5 306,627.4 0
SHAFT_Nc SHAFT_Ncsu SHAFT_Ncsu SHAFT_Type SHAFT_Critic	N _c b N _{sub} p N _{sup}	Shaft speed Critical speed Subcritical avoid speed Supercritical avoid speed Type of operation ACCEF	rpm rpm rpm TABLE SUBCRITICA	476.3 278,752.2 195,126.5 306,627.4 0 L SPEED

User Inputs and Selections

Table 44: Shafting User Inputs and Selections

USER INPUTS AND SELECTIONS

	D	Ditabalian stand for an A	• -		40.00	
SHAFT_DA	D _A	Pitch diameter of gear A	in		10.00	
SHAFT_DB	D _B	Pitch diameter of gear B	in		10.00	
SHAFT_nmin	n _{min}	Minimum Margin of Safety			1.00	
SHAFT_Kfb	K _{fb}	Stress factor for bending			1.00	
SHAFT_Kfs	K _{fs}	Stress factor for sheer			1.00	
SHAFT_Materi	al		-		4	
		Material Selected (see Database)		Al	uminum Alloy T707	'5
SHAFT_Analys	sis		_		1	
		Type of Shaft Analysis			Uniform shaft	
SHAFT_Solid		Solid Shafting	-		0 Hollow shaft	
					HONOW SHAIL	
SHAFT_Wgea	r W _{aear}	Weight of gears	lb	0		0
SHAFT_n	n	Shaft speed	rpm		476.3	
_	ρ	Density	slugs/in3	0.283		0.283
	E	Modulus of Elasticity	psi	30.0E+6		30.0E+6
SHAFT_Mome	ntType	Type of Moment Diagram			1	
SHAFT_FyA	Fy _A	Force of Gear A	lb		0.0	
SHAFT_FyB	Fy _B	Force of Gear B	lb		0.0	
SHAFT_FzA	Fz _A	Force of Gear A	lb		110,000.0	
SHAFT_FzB	Fz_B	Force of Gear B	lb		0.0	
SHAFT_xA	Fx _A	Axial Force	lb		0.0	
SHAFT_xB	Fx_B	Axial Force	lb		0.0	
SHAFT_Delta1	ΔX_1	Distance between members	in		3	
SHAFT_Delta2	Δx_2	Distance between members	in		11	
SHAFT_Delta3	Δx_3	Distance between members	in		0	
SHAFT_TQ_A	T _A	Torque at Gear A	in lb		423,433.0	
SHAFT_TQ_B	T _A	Torque at Gear B	in Ib			
SHAFT_MRhp		Main Rotor HP			14,000.0	

Margin of Safety

Table 45: Shafting Margin of Safety Calculations

SUMMARY FOR MARGIN OF SAFETY

D	Outside diameter	in	8.55
d	Inside diamter	in	7.70
f _b	Vibratory bending stress	psi	12,275.7
n _b	Bending factor of safety		1.16
f _s	Torsional shear stress	psi	10,023.6
n _s	Torsional shear factor of safety		3.9
f _a	Axial tension stress	psi	0.0
n _a	Axial tension factor of safety		0.0
W _{shaft}	Shaft weight	lb	14.98
n	Margin of Safety		1.00
	PASS	S	

GEOMETRY LIMITS

D	Outside diameter	in	8.55
d	Inside diamter	in	7.70
D _A	Pitch diameter of gear A	in	10.00
D_B	Pitch diameter of gear B	in	10.00
D _{max}	Max permitted outside diameter	in	9.00
D _{min}	Min permitted outside diameter	in	0.50
d_{max}	Max permitted inside diameter	in	8.10
d _{min}	Min permitted inside diameter	in	0.00
n _{min}	Minimum Margin of Safety		1.00

BENDING STRESS

K_{fb}	Stress factor for bending		1.00
M _{max}	Maximum moment	lb in	259,285.7
f _b	Vibratory bending stress	psi	12,275.7
n _b	Bending factor of safety		1.16

Table 45: Shafting Margin of Safety Calculations (continued)

TORSIONAL SHEAR STRESS

K _{fs}	Stress factor for sheer	
Т	Shaft torque	in lb
f _s	Torsional shear stress	psi
n _s	Torsional shear factor of safety	

1.00
423,433.0
10,023.6
3.92

0.0 0.0 0.00

AXIAL TENSION STRESS

F _x	Axial force	lb	
f _a	Axial tension stress	lb	
n _a	Axial tension factor of safety	psi	

ALLOWABLE STRESSES

	Material Selected (see Database	e)	4
Sut	Ultimate tensile strength	psi	86,000
Sy	Yield tensile strength	psi	78,600
F _{ty}	Tensile yield strength	psi	78,600
F_{sy}	Shear yield stress	psi	39,300
F_{en}	Endurance limit	psi	14,280

Critical Speed (Uniform Shaft)

Table 46: Shafting Critical Speed Calculations (Uniform Shaft) CRITICAL SPEED SUMMARY

	Type of Shaft Analysis		Uniform shaft
n	Shaft speed	rpm	476.3
N _c	Critical speed	rpm	278,752.2
N _{sub}	Subcritical avoid speed	rpm	195,126.5
N _{sup}	Supercritical avoid speed	rpm	306,627.4
	Type of operation	SUB	CRITICAL

ACCEPTABLE SUBCRITICAL SPEED

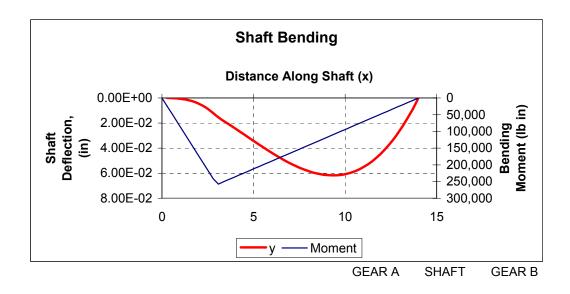


Table 46: Shafting Critical Speed Calculations (Uniform Shaft) (continued)

GEOMETRY & PROPERTIES

L	Shaft length	in		14	
D	Outside diameter	in		8.55	
d	Inside diamter	in		7.70	
I	Cross sectional moment of inertia	in4		90.32	
M _{max}	Maximum moment	lb in		259,285.7	
m	Mass	slugs		20,232.5	
W_{gear}	Weight of gears	lb	0	1,982.8	0
n	Shaft speed	rpm		476.3	
ω	Shaft speed	rad/s		49.9	

MATERIAL PROPERTIES

ρ	Density	slugs/in3	0.098
E	Modulus of Elasticity	psi	10.3E+6

NUMERICAL ITERATION SETUP

р	Number of iteration points		50
dx	Change along x	in	0.28
dw	Segmented change in shaft weight	lb	39.66

RAYLEIGH'S ENERGY METHOD

g	Gravity constant	in/sec ²	386.4
T part	Intermediate value	lb in ²	3.27E+00
V part	Intermediate value	lb in	6.82E+01
Т	Kinetic energy	in Ib	10.54
V	Potential energy	in Ib	34.08
N _c	Critical speed	rpm	856.49

UNIFORM SHAFT METHOD

N _c	Critical Speed for uniform shaft	rpm	278,752.2
N _c	Critical Speed for uniform shaft (thin wall)	rpm	278,366.9

Critical Speed (Nonuniform Shaft)

 Table 47: Shafting Critical Speed Calculations (Nonuniform Shaft)

CRITICAL SPEED SUMMARY

	Type of Shaft Analysis		Nonuniform shaft
n	Shaft speed	rpm	476.3
N _c	Critical speed	rpm	856.5
N _{sub}	Subcritical avoid speed	rpm	599.5
N _{sup}	Supercritical avoid speed	rpm	942.1
	Type of operation	SUBCF	RITICAL



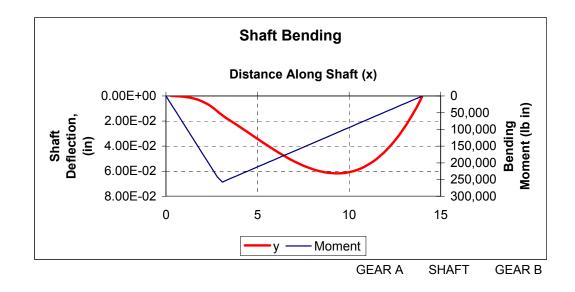


Table 47: Shafting Critical Speed Calculations (Nonuniform Shaft) (continued)

GEOMETRY & PROPERTIES

L	Shaft length	in	14
D	Outside diameter	in	8.55
d	Inside diamter	in	7.70
1	Cross sectional moment of inertia	in4	90.32
M _{max}	Maximum moment	lb in	259,285.7
m	Mass	slugs	20,232.5
W_{gear}	Weight of gears	lb	<mark>0</mark> 1,982.8 0
n	Shaft speed	rpm	476.3
ω	Shaft speed	rad/s	49.9

MATERIAL PROPERTIES

ρ	Density	slugs/in3	0.098	
Е	Modulus of Elasticity	psi	10.3E+6	

NUMERICAL ITERATION SETUP

р	Number of iteration points		50
dx	Change along x	in	0.28
dw	Segmented change in shaft weight	lb	39.66

RAYLEIGH'S ENERGY METHOD

g	Gravity constant	in/sec ²	386.4
T part	Intermediate value	lb in ²	3.27E+00
V part	Intermediate value	lb in	6.82E+01
Т	Kinetic energy	in lb	10.54
V	Potential energy	in lb	34.08
N _c	Critical speed	rpm	856.49

UNIFORM SHAFT METHOD

N _c	Critical Speed for uniform shaft	rpm	278,752.2
N _c	Critical Speed for uniform shaft (thin wall)	rpm	278,366.9

Table 47:	Shafting	Critical Sp	eed Calcul	ations (No	onuniform S	Shaft) (continued)	

ABIIIO								
	BEARING		GEAR		BEA	RING		
	R1	Dx1	А	Dx2	R2			
x		0		3		14		14
Dx			3		11		0	
F		0		0		0		0
Mz		0		0		0		0
Т		0	42	3433		0		0

LOADING FOR Mz

LOADING FOR My

	<u> </u>						
	BEARING	(GEAR		BEARING		
	R1	Dx1 A	A	Dx2	R2		
х	0		3		14		14
Dx		3		11		0	
F	-86428.57		110000		-23571.43		0
My	0		-259285.7		0		0
Т	0		423433		0		0
	_				_		
Mmax	0		259,286		0		0
		3		11		0	
slope		86428.57		-23571.43		#DIV/0!	
xA	FALSE	FALSE	3	FALSE	FALSE	FALSE	FALSE
хB	FALSE	FALSE	FALSE	FALSE	FALSE	FALSE	FALSE
					.,		

	dx	x	М	С	У'	У	w of gears	wi	Ti part	Vi
		0	0		0	0.00	0			
1	0.28	0.28	24,200	2.60E-05	7.28E-06	1.63E-05	0	39.66	1.06E-08	6.47E-04
2	0.28	0.56	48,400	5.20E-05	2.91E-05	1.06E-04	0	39.66	4.46E-07	4.21E-03
3	0.28	0.84	72,600	7.80E-05	6.56E-05	3.43E-04	0	39.66	4.65E-06	1.36E-02
4	0.28	1.12	96,800	1.04E-04	1.17E-04	7.99E-04	0	39.66	2.53E-05	3.17E-02
5	0.28	1.4	121,000	1.30E-04	1.82E-04	1.55E-03	0	39.66	9.53E-05	6.15E-02
6	0.28	1.68	145,200	1.56E-04	2.62E-04	2.67E-03	0	39.66	2.82E-04	1.06E-01
7	0.28	1.96	169,400	1.82E-04	3.57E-04	4.23E-03	0	39.66	7.08E-04	1.68E-01
8	0.28	2.24	193,600	2.08E-04	4.66E-04	6.30E-03	0	39.66	1.57E-03	2.50E-01
9	0.28	2.52	217,800	2.34E-04	5.90E-04	8.95E-03	0	39.66	3.17E-03	3.55E-01
10	0.28	2.8	242,000	2.60E-04	7.24E-04	1.21E-02	0	39.66	5.82E-03	4.80E-01
11	0.28	3.08	257,400	2.77E-04	8.40E-04	1.54E-02	0	39.66	9.37E-03	6.09E-01
12	0.28	3.36	250,800	2.70E-04	9.06E-04	1.82E-02	0	39.66	1.31E-02	7.22E-01
13	0.28	3.64	244,200	2.62E-04	9.55E-04	2.09E-02	0	39.66	1.72E-02	8.27E-01
14	0.28	3.92	237,600	2.55E-04	1.00E-03	2.35E-02	0	39.66	2.20E-02	9.33E-01
15	0.28	4.2	231,000	2.48E-04	1.04E-03	2.63E-02	0	39.66	2.74E-02	1.04E+00
16	0.28	4.48	224,400	2.41E-04	1.08E-03	2.90E-02	0	39.66	3.34E-02	1.15E+00
17	0.28	4.76	217,800	2.34E-04	1.11E-03	3.18E-02	0	39.66	4.01E-02	1.26E+00
18	0.28	5.04	211,200	2.27E-04	1.14E-03	3.46E-02	0	39.66	4.74E-02	1.37E+00
19	0.28	5.32	204,600	2.20E-04	1.17E-03	3.73E-02	0	39.66	5.52E-02	1.48E+00
20	0.28	5.6	198,000	2.13E-04	1.19E-03	4.00E-02	0	39.66	6.35E-02	1.59E+00
21	0.28	5.88	191,400	2.06E-04	1.21E-03	4.27E-02	0	39.66	7.22E-02	1.69E+00
22	0.28	6.16	184,800	1.99E-04	1.22E-03	4.52E-02	0	39.66	8.10E-02	1.79E+00
23	0.28	6.44	178,200	1.92E-04	1.23E-03	4.76E-02	0	39.66	9.00E-02	1.89E+00
24	0.28	6.72	171,600	1.84E-04	1.24E-03	5.00E-02	0	39.66	9.89E-02	1.98E+00
25	0.28	7	165,000	1.77E-04	1.24E-03	5.21E-02	0	39.66	1.08E-01	2.07E+00

Table 47: Shafting Critical Speed Calculations (Nonuniform Shaft) (continued)

Table 47:	Shafting	Critical S	peed	Calculations	(Nonuniform Shaft) ((continued)	
	~~~~~		P		(			

26	0.28	7.28	158,400	1.70E-04	1.24E-03	5.41E-02	0	39.66	1.16E-01	2.15E+00
27	0.28	7.56	151,800	1.63E-04	1.23E-03	5.59E-02	0	39.66	1.24E-01	2.22E+00
28	0.28	7.84	145,200	1.56E-04	1.22E-03	5.75E-02	0	39.66	1.31E-01	2.28E+00
29	0.28	8.12	138,600	1.49E-04	1.21E-03	5.89E-02	0	39.66	1.38E-01	2.34E+00
30	0.28	8.4	132,000	1.42E-04	1.19E-03	6.00E-02	0	39.66	1.43E-01	2.38E+00
31	0.28	8.68	125,400	1.35E-04	1.17E-03	6.09E-02	0	39.66	1.47E-01	2.41E+00
32	0.28	8.96	118,800	1.28E-04	1.14E-03	6.15E-02	0	39.66	1.50E-01	2.44E+00
33	0.28	9.24	112,200	1.21E-04	1.11E-03	6.17E-02	0	39.66	1.51E-01	2.45E+00
34	0.28	9.52	105,600	1.14E-04	1.08E-03	6.17E-02	0	39.66	1.51E-01	2.45E+00
35	0.28	9.8	99,000	1.06E-04	1.04E-03	6.13E-02	0	39.66	1.49E-01	2.43E+00
36	0.28	10.08	92,400	9.93E-05	1.00E-03	6.05E-02	0	39.66	1.45E-01	2.40E+00
37	0.28	10.36	85,800	9.22E-05	9.55E-04	5.93E-02	0	39.66	1.40E-01	2.35E+00
38	0.28	10.64	79,200	8.51E-05	9.06E-04	5.78E-02	0	39.66	1.32E-01	2.29E+00
39	0.28	10.92	72,600	7.80E-05	8.52E-04	5.58E-02	0	39.66	1.23E-01	2.21E+00
40	0.28	11.2	66,000	7.09E-05	7.95E-04	5.33E-02	0	39.66	1.13E-01	2.12E+00
41	0.28	11.48	59,400	6.38E-05	7.33E-04	5.04E-02	0	39.66	1.01E-01	2.00E+00
42	0.28	11.76	52,800	5.68E-05	6.67E-04	4.70E-02	0	39.66	8.78E-02	1.87E+00
43	0.28	12.04	46,200	4.97E-05	5.98E-04	4.31E-02	0	39.66	7.38E-02	1.71E+00
44	0.28	12.32	39,600	4.26E-05	5.24E-04	3.87E-02	0	39.66	5.94E-02	1.54E+00
45	0.28	12.6	33,000	3.55E-05	4.47E-04	3.37E-02	0	39.66	4.51E-02	1.34E+00
46	0.28	12.88	26,400	2.84E-05	3.65E-04	2.82E-02	0	39.66	3.15E-02	1.12E+00
47	0.28	13.16	19,800	2.13E-05	2.80E-04	2.21E-02	0	39.66	1.93E-02	8.75E-01
48	0.28	13.44	13,200	1.42E-05	1.91E-04	1.53E-02	0	39.66	9.31E-03	6.08E-01
49	0.28	13.72	6,600	7.09E-06	9.73E-05	8.19E-03	0	39.66	2.66E-03	3.25E-01
50	0.28	14	0	0.00E+00	1.66E-05	0.00E+00	0	39.66	0.00E+00	0.00E+00
	0		0	0.00E+00	0.00E+00	0.00E+00	0	39.66	0.00E+00	0.00E+00
							Τt	total	3.27E+00	

V total

6.82E+01

# **Total Load and Moments**

# Table 48: Shafting Total Load and Moments Calculations

# TOTAL LOAD AND MOMENTS

				C L	oadir	ng Selection		
	Type of Mon	nent Diagi	ram \prec	)	1	BGB		1
Fy _A	Force of Gea	ar A		lb	2	BGBG		0.0
Fу _в	Force of Gea	ar B		lb	3	GBGB		0.0
Fz _A	Force of Gea	ar A		lb	4	GBBG	110,00	0.0
$Fz_B$	Force of Gea	ar B		lb	5	BGGB		0.0
Fx _A	Axial Force			lb				0.0
Fx _B	Axial Force			lb				0.0
$\Delta x_1$	Distance bet	ween me	mbers	in				3
$\Delta x_2$	Distance bet	ween me	mbers	in				11
$\Delta X_3$	Distance bet	ween me	mbers	in				0
T _A	Torque at Ge	ear A		in lb			423,43	<mark>3.0</mark>
Τ _B	Torque at Ge	ear B		in lb			-423,43	33.0
M _{max}	Maximum m	oment		lb in			259,28	35.7
Т	Shaft torque			in lb			423,43	33.0
Fx	Shaft axial fo	orce	LOADING	FOR N	/I _z			0
	BEARING		GEAR		[	BEARING		
	R1	Dx1	А	Dx2		R2		
x	0		3			14		14
Dx		3		11			0	
F	0		0			0		0
Mz	0		0			0		0
Т	0		423433			0		0

# LOADING FOR M_y

	BEARING		GEAR		BEARING		
	R1	Dx1	А	Dx2	R2		
х	0		3		14		14
Dx		3		11		0	
F	-86428.57		110000		-23571.43		0
My	0		-259285.7		0		0
Т	0		423433		0		0
M _{max}	0		259285.7		0		0

# Material Properties

# Table 49: Shafting Material Properties Database

# MATERIAL DATABASE

			GEAR	SHAFT	GEAR
				Aluminum Alloy T7075	
Sut	Ultimate tensile strength	psi		86,000	
Sy	Yield tensile strength	psi		78,600	
F _{sy}	Shear yield stress	psi		39,300	
S _{en}	Endurance strength	psi		20,000	
F _{en}	Endurance limit	psi		14,280	
ka	Surface factor			1.00	
kb	Size factor			0.70	
kc	Load factor			1.00	
kd	Temperature factor			1.02	
ke	Miscellaneous effects			1.00	
k	Total endurance factor			0.71	
ρ	Density (weight)	lb/in3		0.098	
E	Modulus of Elasticity	psi		10.3E+6	

		General Steel	General Alluminum alloy	Aluminum Alloy 2011	Aluminum Alloy T7075	Steel AISI 4340	Titanium Forging (6 A1-4V)
Sut	Ultimate tensile streng	jth		47,000	86,000	250,000	135,821
Sy	Yield tensile strength			24,500	78,600	230,000	122,642
$F_{sy}$	Shear yield stress	0	0	12,250	39,300	115,000	61,321
S _{en}	Endurance strength	0	20,000	20,000	20,000	100,000	20,000
$F_{en}$	Endurance limit	0	14,280	14,280	14,280	71,400	14,280
ka	Surface factor	1.00	1.00	1.00	1.00	1.00	1.00
kb	Size factor	0.70	0.70	0.70	0.70	0.70	0.70
kc	Load factor	1.00	1.00	1.00	1.00	1.00	1.00
kd	Temperature factor	1.02	1.02	1.02	1.02	1.02	1.02
ke	Miscellaneous effects	1.00	1.00	1.00	1.00	1.00	1.00
k	Total endurance facto	<b>r</b> 0.71	0.71	0.71	0.71	0.71	0.71
ρ	Density (weight)	0.283	0.098	0.098	0.098	0.283	0.161
E	Modulus of Elasticity	30.0E+6	10.3E+6	10.3E+6	10.3E+6	30.0E+6	15.5E+6

# **APPENDIX D: PLANETARY DRIVE CALCULATIONS**

# **Design Calculations**

# **Table 50: Planetary Design Calculations**

### JHL PLANETRARY XMNS DESIGN V3.5

### PERFORMANCE FACTORS

	POWER PLANT	NASA High-T	ech Engine		
	Number of Engines				
HPeng_	Drive System Rated Power	8,773	8,804	8,773 shp	
	Maximum Continuous Power (MCP)	8,773	8,773	8,773 shp	
	Intermediate Rated Power (IRP)	10,397	10,397	10,397 shp	
Hpeng_MRP	Maximum Rated Power (MRP)	11,130	11,130	11,130 shp	
	Contingency Power (CP)	11,512	11,512	11,512 shp	
RPMeng	Engine High Speed Output Shaft Spee		15,000	rpm	

### MAIN ROTOR

			-
	Tip Speed	725 ft/s	1
MRdiam	Diameter	120.4 ft	
MRrpm	Rotor System Speed (100% Speed)	115.0 rpm	
MRPowerReq	Main Rotor Power Required	22,247.0 hp	
		19346	-
	TAIL ROTOR		HPloss
	Tip Speed	725 ft/s	Efficiency
TRdiam	Diameter	29.1 ft	
TRrpm	Tail Rotor System Speed (100% Speed	476.3 rpm	
TRPowerReq	Tail Rotor Power Required	<mark>1,989.0</mark> hp	

### **OVERALL SPEEDS**

Engine High Speed Output Shaft Speed	15,000.0 rpm	AccessPower
Engine Gearbox Output Speed	5,587.7 rpm	OCPower
Main Rotor Drive Shaft Speed	115.0 rpm	OCRPM
Tail Takeoff Drive Shaft Speed	2,264.5 rpm	AccessRPM
Intermediate Tail Drive Shaft Speed	1,421.3 rpm	Nplanets1
Tail Rotor Drive Shaft Speed	476.3 rpm	Nplanets2

### **OVERALL REDUCTION RATIOS**

	ENGINE INPUT GEARBOX REDUX			
EngGBredux	Engine Gearbox Reduction Ratio Bevel Accessory Takeoff	2.684		
	Bevel Accessory Takeoff	5.588		IntTa
				TRre
	MAIN GEAR BOX REDUX	48.589		
	Crown Bevel	4.859		
Stage1Redux	1st Stage Reduction	3.744	3.744	
Stage1Redux Stage2Redux	2d Stage Reduction	2.671	2.6709402	
	Overall Reduction Ratio	130.435		

### EFFICIENCY

HP _{in}	26,350.6 hp
Main Rotor	22,247.0 hp
Tail Rotor	3,200.0 hp
Accessories (all 3)	120.0 hp
Oil Cooler	40.0 hp
HP _{useable}	25,607.0 hp
HP _{Loss}	743.6 hp
Efficiency	97.18%

EFFICIENCY 97.18%

### MISCELLANEOUS

r Accessory Takeoff Power	<mark>40.0</mark> hp
Oil Cooler HP	40.0 hp
Oil Cooler speed	2,000.0 rpm
Accessory Speed	1,000.0 rpm
Nplanets1	5.88
Nplanets2	10.92
Freewheeling Clutch	99.50%
Spur or Helical Gear	99.50%
Bevel Gear	99.50%
Planetary Stage	99.25%

	TR TOTAL REDUX (from Crown)	2.414
	Short Shaft Bevel Takeoff Intermediate Bevel Gear Takeoff	0.508
IntTailRedux	Intermediate Bevel Gear Takeoff	1.593
TRredux	Tail Rotor Gearbox Reduction Ratio	2.984

TR ACCESSORY REDUXS (from sh	ort shaft)
Oil Cooler Spur Takeoff	1.132
Accessory Spur Takeoff	2.000

Table 50: Planetary Design Calculations (continued)

# JHL PLANETRARY XMNS DESIGN V3.5

### ENGINE INPUT GEARBOXES OVERALL DESIGN

### **ENGINE OUTPUTS**

	Gear Redux	RPM	TORQUE	Efficiency	HPin	HPloss	HPout
L High Speed Engine Output Shaft		15,000	36,863.0		8,773.4		8,773.4 CW
C High Speed Engine Output Shaft		15,000	36,991.0		8,803.8		8,803.8 CW
R High Speed Engine Output Shaft		15,000	36,863.0		8,773.4		8,773.4 CW

### L ENGINE INPUT GEARBOX

	Gear Redux	RPM	T _{design} (into)	Efficiency	HPin	HPloss	HPout
L High Speed Engine Output Shaft		15,000.0	36,863.0		8,773.4		8,773.4 CW
L Freewheeling Clutch		15,000.0	36,863.0	99.5%	8,773.4	43.9	8,729.5 CW
L Bevel Input Shaft		15,000.0	36,678.6		8,729.5		8,729.5 CW
L Bevel Input Pinion		15,000.0	36,678.6		8,729.5		8,729.5 CW
L Bevel Input Gear	2.68	5,587.7	98,461.9	99.5%	8,729.5	43.6	8,685.8 CCW
L Bevel Accessory Takeoff	5.59	1,000.0	844.5	99.5%	13.4	0.066667	13.3 CW
L Engine Gearbox Output Shaft		5,587.7	97,818.4		8,672.4		8,672.4 CW
		5,587.7	97,818.4	99.0%		87.6	8,672.4

### C ENGINE INPUT GEARBOX

	Gear Redux	RPM	T _{design} (into)	Efficiency	HPin	HPloss	HPout
C High Speed Engine Output Shaft		15,000.0	36,991.0		8,803.8		8,803.8 CW
C Freewheeling Clutch		15,000.0	36,991.0	99.5%	8,803.8	44.0	8,759.8 CW
C Helical Input Shaft		15,000.0	36,806.1		8,759.8		8,759.8 CW
C Helical Input Pinion		15,000.0	36,806.1		8,759.8		8,759.8 CW
C Helical Idler	1.00	15,000.0	36,806.1	99.5%	8,759.8	43.8	8,716.0 CCV
C Helical Input Gear	2.68	5,587.7	98,310.0	99.5%	8,716.0	43.6	8,672.4 CW
C Engine Gearbox Output Shaft		5,587.7	97,818.4		8,672.4		8,672.4
		5,587.7	97,818.4	98.5%		131.4	8,672.4

### **R ENGINE INPUT GEARBOX**

	Gear Redux	RPM	T _{design} (into)	Efficiency	HPin	HP _{loss}	HPout	
R High Speed Engine Output Shaft		15,000.0	36,991.0		8,773.4		8,773.4 CW	
R Freewheeling Clutch		15,000.0	36,863.0	99.5%	8,773.4	43.9	8,729.5 CW	
R Bevel Input Shaft		15,000.0	36,678.6		8,729.5		8,729.5 CW	
R Bevel Input Pinion		15,000.0	36,678.6		8,729.5		8,729.5 CW	
R Bevel Input Gear	2.68	5,587.7	98,461.9	99.5%	8,729.5	43.6	8,685.8 CCV	N
R Bevel Accessory Takeoff	5.59	1,000.0	844.5	99.5%	13.4	0.066667	13.3 CW	
R Engine Gearbox Output Shaft		5,587.7	97,818.4		8,672.4		8,672.4 CW	
		5,587.7	97,818.4	99.0%		87.6	8,672.4	

# JHL PLANETRARY XMNS DESIGN V3.5

### MAIN GEARBOX DESIGN

#### HPin HPloss HPout RPM TORQUE Efficiency Gear Redux L Engine Gearbox Output Shaft 5,587.7 97,818.4 8,672.4 8,672.4 CW 8,672.4 CW 8,672.4 CW L Crown Bevel Pinion 5.587.7 97,818.4 8.672.4 C Engine Gearbox Output Shaft 5,587.7 97,818.4 8,672.4 8,672.4 CW C Crown Bevel Pinion 5,587.7 97,818.4 8,672.4 8,672.4 CW 8,672.4 CW R Engine Gearbox Output Shaft 97,818.4 5,587.7 8,672.4 R Crown Bevel Pinion 5,587.7 97,818.4 8,672.4 26,017.3 CCW Crown Bevel 1,150.0 1,425,871.8 99.50% 26,017.3 43.4 8,629.1 from L Crown Bevel Input 4.86 1,150.0 475,290.6 99.50% 8,672.4 4.86 1,150.0 475,290.6 99.50% 43.4 8,629.1 from C Crown Bevel Input 8,672.4 from R Crown Bevel Input 4.86 1,150.0 475,290.6 99.50% 8,672.4 43.4 8,629.1 91,910.5 99.50% 3,302.3 Short Shaft Bevel Takeoff 0.51 2,264.5 16.4 3,285.8 CW 1st Stage Sun Shaft 1,150.0 1,237,762.5 22,585.0 22,585.0 CCW 1,150.0 1,237,762.5 99.44% 146.5 22,585.0

#### **CROWN BEVEL**

### **1ST STAGE PLANETARY GEAR**

		Gear Redux	RPM	TORQUE	Efficiency	HPin	HP _{loss}	HPout
Sun 1 Shaft			1,150.0	1,237,762.5		22,585.0		22,585.0 CCW
Spur Sun Gear 1			1,150.0	1,237,762.5		22,585.0		22,585.0 CCW
Per Mesh	Sun 1		1,150.0	210,389.9	99.62%	3,838.9	14.4	3,824.5 CCW
Per Mesh	Planet 1		1,318.8	183,460.0	33.0270	5,050.5	14.4	5,024.5 CW
Per Mesh	Planet 1		1,318.8	183,460.0	99.62%	3.824.5	14.4	3,810.1 CW
Per Mesh	Ring 1		0.0	577,309.8	00.0270	0,024.0	14.4	0,010.1
Per Mesh	Carrier 1	3.74	307.2	781,783.4		3,810.1		3,810.1 CCW
Carrier 1 total			307.2	4,599,376.6		22,415.4		22,415.4 CCW
Sun 2 Shaft			307.2	4,599,376.6		22,415.4		22,415.4 CCW
Planet speed about	post is	966.6	307.2	4,599,376.6	99.25%		169.6309	22,415.4

### 2nd STAGE PLANETARY GEAR

		Gear Redux	RPM	TORQUE	Efficiency	HP _{in}	HPloss	HP _{out}	
Sun 2 Shaft			307.2	4,599,376.6		22,415.4		22,415.4	
Spur Sun Gear 2			307.2	4,599,376.6		22,415.4		22,415.4	
Per Mesh	Sun 2		307.2	421,153.0	99.62%	2,052.5	7.7	2,044.8	CCW
Per Mesh	Planet 2		915.6	141,284.2	99.02 /0	2,032.5	1.1	2,044.0	CW
Per Mesh	Planet 2		915.6	141,284.2	99.62%	2,044.8	7.7	2,037.1	CW
Per Mesh	Ring 2		0.0	703,721.5	33.0270	2,044.0	1.1	2,007.1	
Per Mesh	Carrier 2	2.67	115.0	1,116,425.9		2,037.1		2,037.1	CCW
Carrier 2 total			115.0	12,192,392.4		22,247.0		22,247.0	CCW
Main Rotor Shaft			115.0	12,192,392.4		22,247.0		22,247.0	CCW
Planet speed about	post is	572.8	115.0	12,192,392.4	99.25%		168.3569	22,247.0	

# JHL PLANETRARY XMNS DESIGN V3.5

### TAIL ROTOR GEARBOX DESIGN

### **OIL COOLER GEARBOX**

	Gear Redux	RPM	TORQUE	Efficiency	HPin	HPloss	HPout
Short Shaft BevelTakeoff		2,264.5	91,453.2		3,285.8		3,285.8 +
Short Shaft		2,264.5	91,453.2		3,285.8		3,285.8 +
Short Shaft Spur Gear		2,264.5	91,453.2		3,285.8		3,285.8 +
Oil Cooler Spur Takeoff Gear	1.13	2,000.0	1,266.8	99.50%	40.2	0.2	40.0
Accessory Spur Takeoff Pinion	2.00	1,000.0	844.6	99.50%	13.4	0.1	13.3 CCV
Tail Takeoff Shaft Segment		2,264.5	89,961.4		3,232.2		3,232.2 CCV
		2,264.5	89,961.4	99.99%		0.3	3,232.2

### INTERMEDIATE TAIL ROTOR GEARBOX

	Gear Redux	RPM	TORQUE	Efficiency	HPin	HPloss	HPout
Tail Takeoff Shaft Segment		2,264.5	89,961.4		3,232.2		3,232.2 CCW
Intermediate Bevel Pinion		2,264.5	89,961.4		3,232.2		3,232.2 CCW
Intermediate Bevel Gear	1.59	1,421.3	143,332.6	99.50%	3,232.2	16.2	3,216.1 CW
Tail Intermediate Shaft Seg.		1,421.3	142,615.9		3,216.1		3,216.1 CW
		1,421.3	142,615.9	99.50%		16.2	3,216.1

### TAIL ROTOR GEARBOX

	Gear Redux	RPM	TORQUE	Efficiency	HPin	HPloss	HPout	
Tail Intermediate Shaft Seg.		1,421.3	142,615.9		3,216.1		3,216.1 CV	Ν
Tail Rotor GB Bevel Pinion		1,421.3	142,615.9		3,216.1		3,216.1 CV	Ν
Tail Rotor GB Bevel Gear	2.98	476.3	425,560.8	99.50%	3,216.1	16.1	3,200.0 CC	CW
Tail Rotor Drive Shaft		476.3	423,433.0		3,200.0		3,200.0 CC	CW
		476.3	423,433.0	99.50%		16.1	3,200.0	

# Weight Equations

# Table 51: Planetary Drive Weight Equations**BOEING VERTOL**

	a _{mr}	MR adjustment factor	1
	a _{tr}	TR adjustment factor	0.9
	HP _{mr}	Drive sys rating	26,350.6
	$HP_{tr}$	TR HP required	3200
	rpm _{mr}	MR speed	115
	rpm _{tr}	TR speed	476.3
BO_z	Z _{mr}	Number of stages in main drive	4
BO_kt	<b>k</b> t	Configuration factor	1
BO_W_mr	W _{dsmr}	Weight of MR drive sys	12,023.9
BO_W_tr	W _{dstr}	Weight of TR drive sys	1,337.5
BO_W_Dsys	$W_{ds}$	Weight of drive sys	13,361.4

# RTL

	HP _{xmmr}	Transmission rating	27,000.0	26,350.6
	rpm _{mr}	MR speed	115	
	T _{mrgb}	Ratio of Xmns rating to MR rpm	234.8	
	HP _{tr}	TR HP required	1,989.0	
	rpm _{tr}	TR speed	476.3	
	T _{trgb}	Ratio of TR hp to its rpm to Tmrgb	1.8	
RTL_ngb	n _{gb}	Number of gearboxes	7	
RTL_Ldr	L _{dr}	Horiz distance in ft b/w rotors	45	
RTL_ndsh	n _{dsh}	Number of drive shafts	10	
RTL_Wgb	$W_{gb}$	Weight of gearboxes	15,835.2	
RTL_Wdsh	$W_{dsh}$	Weight of drive shafts	784.9	
RTL_Wds	$W_{ds}$	Weight of drive sys	16,620.1	

# **Modified Solid Rotor Volume Weight Estimation**

# Table 52: Planetary Drive Solid Rotor Volume Weight Estimations

13,729.8

lb

### WEIGHT ESTIMATION (SOLID ROTOR VOLUME METHOD)

TOTAL GEARBOX WEIGHT

	Kplan	600	
	Kbv	600	G 0.25
	Kspur	600	
GEARBOX WEIGHT SUMMARY	K	600	
Engine Input Gearbox Left	348.9	lb	
Engine Input Gearbox Center	424.5	lb	
Engine Input Gearbox Right	348.9	lb	PLANETARY TOTAL WEIGHT
Oil Cooler Gearbox	4.9	lb	Solid Rotor Volume 7,868.4 lbs
Intermediate Tail Rotor Gearbox	431.8	lb	Optimal Weight 7,785.5 lbs
Tail Rotor Gearbox	1,571.5	lb	
Main Rotor Gearbox	10,599.3	lb	
-	13,729.8		

Name	<u>m</u> e	<u>RPM</u>	<u>TQ</u>	<u>ĸ</u>	Vol	<u>G</u>	Weight	Sys Weight
ENGINE INPUT GEARBOX LEFT								
L Bevel Input Pinion		15,000.0	36,678.6	600	167.8	0.25	42.0	
L Bevel Input Gear	2.6844	5,587.7	98,461.9	600	1,209.3	0.25	302.3	344.3
L Bevel Input Gear		5587.743741	<del>151.1</del>	<del>600</del>	0.6	<del>0.25</del>		
L Bevel Accessory Takeoff	5.5877	1000	844.5	600	18.5	0.25	4.6	4.6
ENGINE INPUT GEARBOX CENTER								
C Helical Input Pinion		15000	36,806.1	600	245.4	0.25	61.3	
C Helical Idler	1	15000	36,806.1	600	245.4	0.25	61.3	122.7
<del>C Helical Idler</del>		15000	<del>36,622.1</del>	<del>600</del>	<del>167.5</del>	<del>0.25</del>		
C Helical Input Gear	2.6844	5587.743741	98,310.0	600	1,207.4	0.25	301.8	301.8
ENGINE INPUT GEARBOX RIGHT								
R Bevel Input Pinion		15.000.0	36.678.6	600	167.8	0.25	42.0	
R Bevel Input Gear	2.6844	5,587.7	98,461.9	600	1,209.3	0.25	302.3	344.3
R Bevel Input Gear		5587.743741	<del>151.1</del>	600	<del>0.6</del>	<del>0.25</del>		
R Bevel Accessory Takeoff	5.5877	1000	844.5	600	18.5	0.25	4.6	4.6
CROWNWHEEL								
L Crown Bevel Pinion		5587.743741	97,818.4	600	393.2	0.25	98.3	
	# 4 <del>.8589</del>	<del>1150</del>	475,290.6	<del>600</del>	<del>9,282.3</del>	<del>0.25</del>		98.3
C Crown Bevel Pinion		5587.743741	97818.40701	600	393.2	0.25	98.3	
<ul> <li>from C Crown Bevel Input</li> </ul>	ut 4.8589	<del>1150</del>	475290.6013	600	<del>9,282.3</del>	0.25		98.3
R Crown Bevel Pinion			97818.40701	600	393.2	0.25	98.3	
from R Crown Bevel Input	ut 4.8589	<del>1150</del>	475290.6013	<del>600</del>	<del>9,282.3</del>	<del>0.25</del>		98.3
Crown Bevel	4.8589	1150	475,290.6	600	9,282.3	0.25	2,320.6	2,320.6
Crown Bevel		<del>1,150.0</del>	<del>180,979.9</del>	600	<del>1,791.2</del>	<del>0.25</del>		
Short Shaft Bevel Takeoff	0.5078	2264.451662	91910.49964	600	462.0	0.25	115.5	115.5
OIL COOLER GEARBOX								
Short Shaft Spur Gear		2264.451662	1118.894589	600.0	7.0	0.25	1.8	
Oil Cooler Spur Takeoff Ge	ar 1.1322	2000	1266.841356	600.0	9.0	0.25	2.3	2.3
Short Shaft Spur Gear		<del>2264.451662</del>	422.280452	<del>600</del>	<del>2.1</del>	<del>0.25</del>	0.5	
Accessory Spur Takeoff Pir	nion 2	1000	844.560904	600	8.4	0.25	2.1	2.1
Short Shaft Spur Gear	0	2264.451662	422.280452	600	2.111402	0.25	0.527851	0.5

Intermediate Bevel Pinion		2264.451662	89961.37402	600	488.1	0.25	122.0	
Intermediate Bevel Gear	1.5933	1421.262355	143332.5678	600	1,239.0	0.25	309.7	431.8
ROTOR GEARBOX								
Tail Rotor GB Bevel Pinion		1421.262355	142615.9049	600	634.7	0.25	158.7	
Tail Rotor GB Bevel Gear	2.984	476.3	425560.8167	600	5,651.4	0.25	1,412.8	1,571.5
TAGE PLANETARY GEAR								
Spur Sun Gear 1		1150	1237762.546	600		0.25		
Number of Planets	6							
Mo1	3.744							
ms1	0.872							
Spur Sun Gear 1		1,150.0	1,237,762.5	600	1,505.5	0.25	376.4	
Spur Planet 1 total weight		1318.807339	183459.9594	600	6,735.0	0.25	1,683.8	
				r	per planet	286.2		
Spur Ring 1		0.0	577,309.8	600.0	4,534.4	0.25	1,133.6	3,193.7
STAGE PLANETARY GEAR								
Spur Sun Gear 2		307.1581197	4599376.634	600		0.25		
Number of Planets	11							
Mo2	2.671							
ms1	0.3355							
Spur Sun Gear 2		307.2	4,599,376.6	600	5,588.5	0.25	1,397.1	
Spur Planet 1 total weight		307.1581197	4599376.634	600	6,868.6	0.25	1,717.1	
				F	per planet	157.2		
Spur Ring 1		0.0	421.153.0	600.0	6.241.4	0.25	1.560.3	4,674.6

# Table 52: Planetary Drive Solid Rotor Volume Weight Estimations (continued)

# Force Feed Oil Cooling

# Table 53: Planetary Drive Force Feed Oil Cooling LUBRICATION ANALYSIS

SYMBOL	DESCRIPTION	UNITS	<u>IN</u>	MESH	<u>OUT</u>
MAIN	GEARBOX				
HΡ η _{mesh}	Power efficiency	hp	26,017.3	97.9%	25,483.7
P _{loss}	Power dissipated	hp		533.7	
Q C _p	heat generated Specific heat of oil	Btu/min Btu/lb-°F		22,627.9 0.5	
	Oil flow design	lb/min		2 1005.684	
Μ	Oil flow	gal/min lb/min		134.0913 900	134.0913
$M_{manual}$	Oil flow	gal/min lb/min		120 1005.7	
$M_{HLH}$	HLH Design oil flow ( $\Delta$ T=+45°F)	gal/min lb/min		134.1 1508.5	
M _{rec}	Recommended oil flow	gal/min gpm/hp		201.1	
ΔT	Rule of thumb Temperature rise	gpm ⁰F		52.0347 45.0	
t _{in}	Incoming oil temperature	°F		45.0 125	
t _{out} t _{oil}	Outgoing oil temperature Oil temperature (average)	°F °F		170.0 147.5	
-011	- · · · · · · · · · · · · · · · · · · ·	-			

# ENGINE INPUT GEARBOXES

HP	Power	hp	8,773.4		8,672.4
$\eta_{\text{mesh}}$	efficiency			<mark>98.8%</mark>	
Ploss	Power dissipated	hp		100.9	
Q	heat generated	Btu/min		4,278.8	
Cp	Specific heat of oil	Btu/lb-°F		0.5	
	Oil flow design			2	
		lb/min		190.1674	
M	Oil flow	gal/min		25.35565	32.16674
		lb/min	-	900	
M _{manual}	Oil flow	gal/min		120	
		lb/min		190.2	
M _{HLH}	HLH Design oil flow (∆T=+45°F)	gal/min		25.4	
		lb/min		285.3	
M _{rec}	Recommended oil flow	gal/min		38.0	
		gpm/hp		0.002	
	Rule of thumb	gpm		17.5467	
$\Delta T$	Temperature rise	°F		45.0	
t _{in}	Incoming oil temperature	°F		125	
t _{out}	Outgoing oil temperature	°F		170.0	
t _{oil}	Oil temperature (average)	°F		147.5	

# Minimum Weight Solution

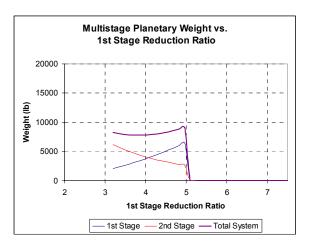
# Table 54: Planetary Drive Minimum Weight SolutionMINIMIZED SOLUTION

$W_1$	1st stage weight	3,145.9	
$W_2$	2nd stage weight	4,639.5	
W _{total}	Total Weight	7,785.5	
M _{o1}	1st stage redux ratio	3.7440	3.7440
M _{o2}	2nd stage redux ratio	2.6709	
b ₁	# Stage 1 planets	5.883185321	
b ₂	# Stage 2 planets	10.92091529	

### Mo'

10.00

Maximum N	umber of Planets	
m	slope	2.7467
С	intercept	2.7333
b ₁	# Stage 1 planets	6
b ₂	# Stage 2 planets	11
Current Setti	ngs	
M _{o1}	1st stage redux ratio	3.744
M _{o2}	2nd stage redux ratio	2.670940171
G	Application factor	0.25
K _{planetary}	K factor	600 lb/in ²
T _{sun1}	Torque at 1st stage sun	1,219,239.2 in lb
		12,192,392.4



# **APPENDIX E: SPLIT TORQUE DRIVE CALCULATIONS**

# **Design Calculations**

# **Table 55: Split Torque Drive Design Calculations**

### JHL SPLIT TORQUE XMNS DESIGN V3.5

### PERFORMANCE FACTORS

POWER PLANT	NASA High-Teo	h Engine			
Number of Engines	3.00				
Drive System Rated Power	8,293	8,293	8,293	24,878.8	
Maximum Continuous Power (MCP)	8,293	8,293	8,293 shp		
Intermediate Rated Power (IRP)	9,827	9,827	9,827 shp		
Maximum Rated Power (MRP)	10,521	10,521	10,521 shp		
Contingency Power (CP)	11,512	11,512	11,512 shp		
Engine High Speed Output Shaft Spee	e	15,000	rpm		

### MAIN ROTOR

Tip Speed	725 ft/s	NumbBevInput
Diameter	120.4 ft	
Rotor System Speed (100% Speed)	115.0 rpm	NumbIdlers
Main Rotor Power Required	22,247.0 hp	

#### TAIL ROTOR

Tip Speed	725	ft/s	SpurCombRedu
Diameter	29.1	ft	IdlerRedux
Tail Rotor System Speed (100% Speed	476.3	rpm	
Tail Rotor Power Required	1,989.0	hp	TRGBRedux
			IntTRGBRedux

### **DESIGN FACTORS**

# Bevel Input Split Per Path	2
# Engines	3
# Idlers per Input Spur	2

### **OVERALL REDUCTION RATIOS**

SpurCombRedux	Spur Combiner	11.70
IdlerRedux	Spur Idler	3.29
	Bevel Input	3.39
TRGBRedux	TR GB Redux	2.00
IntTRGBRedux	IntTRGB Redux	<mark>2.19</mark>
TakeoffBevRedux	Takeoff Bevel Redux	0.64

#NAME?

97.90%

### **OVERALL SPEEDS**

High Speed Engine Output Shaft	15,000.0 rpm
Main Rotor Shaft	115.0 rpm
Tail Shaft	2,090.2 rpm
Intermediate Tail Shaft	952.6 rpm
Tail Rotor Shaft	476.3 rpm

AccessHP Accessrpm

### SHAFT DATA

Tail Shaft	2,009.0	hp
IntTailShaft	2,009.0	hp
Main Rotor Shaft Torque	<i>##########</i> ##########################	n Ib
Tail Shaft Torque	60,578.1 i	n Ib
Intermediate Tail Shaft Torque	132,256.3 i	n Ib
Tail Rotor Shaft Torque	263,190.1 i	n Ib

EFFICIENCY

# **MISCELLANEOUS**

HPloss	602.8 hp	
Accessory Takeoff Power	40.0 hp	
Accessory Speed	1,000.0 rpm	
Freewheeling Clutch	99.50%	
Spur or Helical Gear	99.50%	
Bevel Gear	99.50%	
Combiner pinions per combiner	12	
Combiners per MR Shaft	2	

Combiner pinions per con Combiners per MR Shaft

# JHL SPLIT TORQUE XMNS DESIGN V3.5

### **ENGINE OUTPUTS**

	Gear Redux	RPM	TORQUE	Efficiency	HP _{in}	HPloss	HPout
L High Speed Engine Output Shaft		15,000	34,844.3		8,292.9		8,292.9
C High Speed Engine Output Shaft		15,000	34,844.3		8,292.9		8,292.9
R High Speed Engine Output Shaft		15,000	34,844.3		8,292.9		8,292.9
					8,292.9	0.0	8,292.9

### **BEVEL INPUT**

	Gear Redux	RPM	TORQUE	Efficiency	HPin	HPloss	HPout
High Speed Engine Output Shaft		15,000.0	34,844.3		8,292.9		8,292.9
Freewheeling Clutch		15,000.0	34,844.3	99.5%	8,292.9	41.5	8,251.5
Bevel Input Shaft		15,000.0	34,670.1		8,251.5		8,251.5
Bevel Input Pinion		15,000.0	17,250.6		4,105.6		4,105.6
Bevel Input Gear	3.39	4,421.9	58,517.3	99.5%	4,105.6	20.5	4,085.1
Bevel Accessory Takeoff	4.42	1,000.0	2,533.7	99.5%	40.2	0.2	40.0
Bevel Output Shaft		4,421.9	58,224.7		4,085.1		4,085.1
	3.39			99.0%	24,878.8	248.2	24,630.6
COMBINER BOX							
	Gear Redux	RPM	TORQUE	Efficiency	HP _{in}	HPloss	HPout
Bevel Output Shaft		4421.921	58,224.7		4,085.1		4,085.1
Input Spur		4421.921	58,224.7		4,085.1		4,085.1
Idler	3.29	1345.5	95,676.4	99.5%	2,042.6	10.2	2,032.3
Idler Shaft		1345.5	95,198.0		2,032.3		2,032.3
Combiner Pinion		1345.5	47,599.0		1,016.2		1,016.2
Spur Tail Takeoff Pinion	0.09	1345.5	47,527.4	99.5%	1,014.6	5.1	1,009.6
Combiner	11.70	115.0	6,126,830.4	99.50%	11,179.4	55.9	11,123.5
Main Rotor Shaft		115.0	12,192,392.4		22,247.0		22,247.0
	38.45	115.0	12,192,392.4	99.0%	24,510.6	244.5	24,266.1

# Table 55: Split Torque Drive Design Calculations (continued)

## Tail Takeoff Box

	Gear Redux	RPM	TORQUE	Efficiency	HPin	HPloss	HPout
Spur Tail Takeoff Pinion		1345.5	47,289.8		1,009.6		1,009.6
Takeoff Shaft		1345.5	94,579.5		2,019.1		2,019.1
Takeoff Bevel Pinion		1345.5	94,579.5		2,019.1		2,019.1
Takeoff Bevel Gear	0.64	2090.203	60,882.5	99.5%	2,019.1	10.1	2,009.0
Tail Shaft		2090.203	60,578.1		2,009.0		2,009.0
	0.64	2090.203	60,578.1	199.0%	1,009.6	-999.5	2,009.0

## Intermediate TR Gearbox

	Gear Redux	RPM	TORQUE	Efficiency	HPin	HPloss	HPout
Tail Shaft		2090.203	60,578.1		2,009.0		2,009.0
Intermediate Tail Bevel Pinion		2090.203	60,578.1		2,009.0		2,009.0
Intermediate Tail Bevel Gear	2.19	952.6	132,920.9	99.5%	2,009.0	10.0	1,999.0
Intermediate Tail Shaft		952.6	132,256.3		1,999.0		1,999.0
	2.19	952.6	132,256.3	99.5%	2,009.0	10.0	1,999.0

### **Tail Rotor Gearbox**

	Gear Redux	RPM	TORQUE	Efficiency	HP _{in}	HPloss	HPout
Intermediate Tail Shaft		952.6	132,256.3		1,999.0		1,999.0
Tail Bevel Pinion		952.6	132,256.3		1,999.0		1,999.0
Tail Bevel Gear	2.00	476.3	264,512.6	99.5%	1,999.0	10.0	1,989.0
Tail Rotor Drive Shaft		476.3	263,190.1		1,989.0		1,989.0
	2.00	476.3	263,190.1	99.5%	1,999.0	10.0	1,989.0

# Weight Equations

# Table 56: Split Torque Weight Equations **BOEING VERTOL**

	a _{mr} a _{tr}	MR adjustment factor TR adjustment factor	1 0.9
BO_z BO_kt	HP _{mr} HP _{tr} rpm _{mr} rpm _{tr} Z _{mr} K _t	Drive sys rating TR HP required MR speed TR speed Number of stages in main drive Configuration factor	24,878.8 1989 115 476.3 24 1
BO_W_mr	W _{dsmr}	Weight of MR drive sys	15,619.4

BO_W_mr	W _{dsmr}	Weight of MR drive sys	15,619.4
BO_W_tr	W _{dstr}	Weight of TR drive sys	914.3
BO_W_Dsys	$W_{ds}$	Weight of drive sys	16,533.7

# RTL

RTL_Wds

 $W_{\text{ds}}$ 

	$HP_{xmmr}$	Transmission rating	24,878.8
	rpm _{mr}	MR speed	115
	T _{mrgb}	Ratio of Xmns rating to MR rpm	216.3
	$HP_{tr}$	TR HP required	1,989.0
	rpm _{tr}	TR speed	476.3
	$T_{trgb}$	Ratio of TR hp to its rpm to Tmrgb	1.9
RTL_ngb	n _{gb}	Number of gearboxes	5
RTL_Ldr	L _{dr}	Horiz distance in ft b/w rotors	45
RTL_ndsh	n _{dsh}	Number of drive shafts	13
RTL_Wgb	$W_{gb}$	Weight of gearboxes	14,274.1
RTL_Wdsh	$W_{dsh}$	Weight of drive shafts	834.6

Weight of drive sys

14,274.1
834.6
15,108.6

# **Modified Solid Rotor Volume Weight Estimation**

# Table 57: Split Torque Modified Solid Rotor Volume Weight Estimation

# SOLID ROTOR VOLUME FOR GEARBOXES

Kplan	600
Kbv	600
Kspur	600
К	600
G	0.25

TOTAL GB WEIGHT 13,325.1

<u>Name m_G RPM TQ K</u>	<u>Vol</u>	<u>G</u>	<u>Weight</u>	<u>Component</u> <u>Weight</u>
BEVEL INPUT GEARBOX				
Bevel Input Pinion 0 15,000.0 17,250.6 600	74.5	0.25	18.6	111.7
Bevel Input Gear         3.392191         4,421.9         58,517.3         600	856.7	0.25	214.2	1,285.1
Bevel Input Gear 4421.921 573.0 600	2.3	0.25	<del>214.1827</del>	
Bevel Accessory Takeoff         4.421921         1000         2,533.7         600	45.8	0.201005	9.2	27.6
COMBINER GEARBOX				
Input Spur 4421.921 58,224.7 600	253.1	0.25	63.3	379.7
Idler 3.286452 1345.5 95,676.4 600	2,324.0	0.25	581.0	3,486.0
Combiner Pinion 1345.5 47,599.0 600	146.4	0.25	36.6	878.3
Combiner 11.7 115 6,126,830.4 600	10,079.0	0.25	2,519.7	5,039.5
OC GEARBOX				
Spur Tail Takeoff Pinion         0.08547         1345.5         47,527.4         600	2,012.0	0.25	503.0	1,006.0
Combiner 11.7 115 6,126,830.4				-
INTERMEDIATE TAIL ROTOR GEARBOX				
Takeoff Bevel Pinion         1345.5         94,579.5         600	805.0	0.25	201.3	201.3
Takeoff Bevel Gear         0.643717         2090.203         60,882.5         600	333.6	0.25	83.4	83.4
	004.0	0.05	405.0	405.0
Tail Bevel Pinion         952.6         132,256.3         600           Tail Bevel Gear         2         476.3         264,512.6         600	661.3 2,645.1	0.25 0.25	165.3 661.3	165.3 661.3
	2,040.1	0.20	001.5	13,325.1

# **APPENDIX F: MODEL FIT FOR PLANETARY DRIVE**

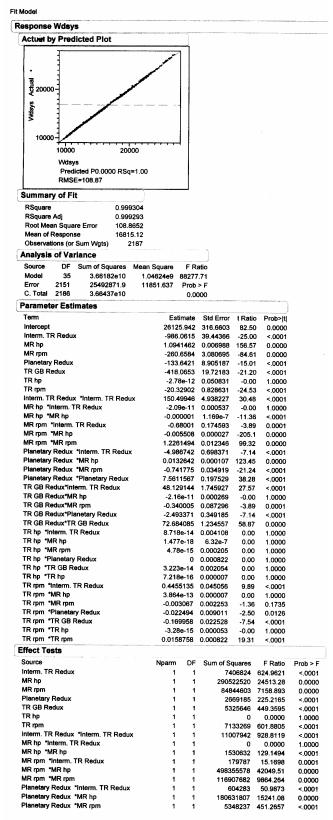


 Table 58:
 RSE Model Fit for Planetary Drive

| esponse Wdsys                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                          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| L<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR npm<br>Planetary Redux<br>TR GB Redux<br>TR dB Redux<br>TR dB Redux<br>TR hp<br>Interm. TR Redux 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR npm 'MR hp<br>MR npm 'MR hp<br>Planetary Redux 'Interm. TR Redux<br>Planetary Redux 'MR hp<br>Planetary Redux 'MR npm<br>Planetary Redux 'MR npm<br>Planetary Redux 'Interm. TR Redux<br>TR GB Redux'Interm. TR Redux<br>TR GB Redux'Interm. 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Estimat<br>6041.81<br>39.554<br>13.008<br>2381.07<br>266.939<br>5.9169<br>1.23e-1<br>197.940<br>10.4994<br>-1.36e-<br>66.1199<br>13.600<br>16.038<br>0.4597<br>4.9337<br>10.0860<br>-74.177<br>9.02828<br>2.809e-                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                        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| Scaled Estimates<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR rpm<br>Planetary Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'MR hp<br>MR rpm 'MR hp<br>Planetary Redux 'Interm. TR Redux<br>Planetary Redux 'MR hp<br>Planetary Redux 'MR pm<br>Planetary Redux 'MR pm<br>TR GB Redux'Interm. TR Redux<br>TR GB Redux'MR pm<br>TR GB Redux'NR pm<br>TR GB Redux'Planetary Redux                                                                                                                                                                                                                                                                                                                                                                                                                                                                 | Scaled<br>16<br>-1<br>-1<br>-1<br>-2<br>21<br>21<br>-1<br>-2<br>-3<br>-3<br>-5<br>-5<br>-5<br>-7<br>-7<br>-7<br>-7<br>-7<br>-18<br>-9<br>-9<br>-9<br>                                                                                                                                                                                                     | Estimat<br>5041.811<br>39.554<br>113.008<br>381.07<br>266.939<br>5.9169<br>1.236-1<br>197.940<br>5.9169<br>1.236-1<br>197.940<br>5.9169<br>1.236-1<br>9.04994<br>1.360<br>0.4597<br>14.0860<br>7.4.177<br>19.0289<br>3.25228<br>2.26096<br>-13.600<br>4.9337                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                             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9011<br>2.851<br>2.855<br>2.855<br>2.857<br>2.857<br>2.851<br>2.851<br>2.851<br>2.851<br>2.851<br>2.851<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3 | i928         17           087         14           087         14           087         4           087         4           087         4           087         4           087         4           087         4           087         4           087         4           087         11           1227         12           1227         12           1227         12           854         12           854         12           854         12           854         12           854         12           854         12           854         12           854         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12      1227         12 <tr td="" td<=""><td>79.28<br/>48.95<br/>42.61<br/>35.15<br/>44.37<br/>75.73<br/>-0.00<br/>11.36<br/>30.48<br/>-0.00<br/>11.36<br/>99.32<br/>-7.14<br/>23.45<br/>99.32<br/>27.57<br/>-0.00<br/>99.32<br/>27.57<br/>43.88<br/>27.57<br/>-0.00<br/>-7.14<br/>58.87<br/>0.00<br/>0.00<br/>0.00</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></tr> <tr><td>Scaled Estimates<br/>Scaled Estimates<br/>Scaled Estimates<br/>Continuous factors centered by mean, sca<br/>Term<br/>Intercept<br/>Intern. TR Redux<br/>MR hp<br/>MR npm<br/>Planetary Redux<br/>TR GB Redux<br/>TR GB Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "MR hp<br/>MR npm "Interm. TR Redux<br/>MR npm "MR hp<br/>Planetary Redux "MR npm<br/>Planetary Redux "MR hp<br/>Planetary Redux "MR hp<br/>Planetary Redux "MR npm<br/>Planetary Redux "MR npm<br/>Planetary Redux "MR npm<br/>Planetary Redux "MR npm<br/>TR GB Redux"MR npm<br/>TR GB Redux"MR npm<br/>TR GB Redux"Hanetary Redux<br/>TR GB Redux"Hanetary Redux<br/>TR GB Redux"TR GB Redux<br/>TR GB Redux"TR GB Redux<br/>TR hp "MR npm<br/>TR hp "MR npm<br/>TR hp "MR npm<br/>TR hp "MR npm</td><td>Scaled 
<br/>16<br/>-1<br/>41<br/>-2<br/>12<br/>21<br/>-4<br/>45<br/>-5<br/>-7<br/>499<br/>-2<br/>43<br/>-3<br/>-3<br/>-4<br/>-2<br/>20<br/>-7<br/>49<br/>-2<br/>-2<br/>20<br/>-7<br/>-8<br/>-8<br/>-9<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2</td><td>Estimat<br/>8041.81<br/>39.554<br/>13.008<br/>381.07<br/>766.939<br/>5.9169<br/>15.9169<br/>1.23e-1<br/>97.940<br/>00.4994<br/>1.3600<br/>116.038<br/>00.4597<br/>11.0860<br/>74.177<br/>11.0860<br/>2.25028<br/>2.809e-<br/>13.609<br/>0.25828<br/>2.809e-<br/>13.609<br/>0.25828<br/>2.809e-<br/>13.609<br/>0.25828<br/>2.809e-<br/>13.600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.3600<br/>1.36000<br/>1.36000<br/>1.36000<br/>1.36000<br/>1.36000<br/>1.36000<br/>1.36000<br/>1.36000000000000000000000000000000000000</td><td></td><td>9015<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3</td><td>i928         17           i087         14           i087         4           i087         -1           i227         -1           i2454         -1           i254         -1           i2454         -1           i254         -1           i254         -1           i254         -1           i254         -1           i254         -1      i254         -1</td><td>79.28<br/>48.95<br/>42.61<br/>35.15<br/>44.37<br/>75.73<br/>0.000<br/>74.65<br/>30.48<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>99.32<br/>21.24<br/>38.28<br/>99.32<br/>21.24<br/>38.28<br/>57.57<br/>-0.00<br/>-3.89<br/>0.00<br/>-3.89<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></tr> <tr><td>Scaled Estimates<br/>Scaled Estimates<br/>Continuous factors centered by mean, sca<br/>Term<br/>Intercept<br/>Intern: TR Redux<br/>MR hp<br/>MR rpm<br/>Pilanetary Redux<br/>TR GB Redux<br/>TR GB Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "Interm. TR Redux<br/>MR rpm "Interm. TR Redux<br/>MR rpm "MR rpm<br/>Pianetary Redux "MR hp<br/>Pianetary Redux "MR hp<br/>Pianetary Redux "MR rpm<br/>Pianetary Redux "MR rpm<br/>Pianetary Redux "MR rpm<br/>Pianetary Redux "MR rpm<br/>Pianetary Redux "MR rpm<br/>TR GB Redux"Haren TR Redux<br/>TR GB Redux"TR RGB Redux<br/>TR hp "Interm. TR Redux<br/>TR hp "MR rpm<br/>TR hp "MR rpm<br/>TR hp "MR rpm<br/>TR hp "MR rpm<br/>TR hp "TR GB Redux</td><td>Scaled 
<br/>16<br/>-1<br/>41<br/>-2<br/>12<br/>21<br/>21<br/>21<br/>21<br/>21<br/>21<br/>21<br/>21<br/>2</td><td>Estimat<br/>0041.81<br/>39.554<br/>13.008<br/>3381.07<br/>260.339<br/>1.23e-1<br/>97.940<br/>5.9189<br/>1.23e-1<br/>1.360-<br/>6.1199<br/>11.360-<br/>6.1199<br/>113.600<br/>0.4597<br/>11.0660<br/>2.25828<br/>2.809e-<br/>113.600<br/>0.4933<br/>10.0289<br/>2.25828<br/>2.809e-<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028<br/>11.6028</td><td></td><td>9015<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491</td><td>i928         17           i087         -           i227         -           i854           8</td><td>79.28<br/>48.95<br/>42.61<br/>35.15<br/>44.37<br/>75.73<br/>30.48<br/>-0.00<br/>11.36<br/>99.32<br/>47.45<br/>30.48<br/>-0.00<br/>11.36<br/>99.32<br/>47.45<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>21.57<br/>-0.00<br/>-3.89<br/>-7.14<br/>21.24<br/>38.28<br/>21.55<br/>0.00<br/>0.00<br/>0.00<br/>0.000</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></tr> <tr><td>L<br/>Scaled Estimates<br/>Continuous factors centered by mean, sca<br/>Term<br/>Intercept<br/>Intern. TR Redux<br/>MR hp<br/>MR pm<br/>Planetary Redux<br/>TR GB Redux<br/>TR fpm<br/>Interm. TR Redux 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR pm 'Interm. TR Redux<br/>Planetary Redux 'MR hp<br/>Planetary Redux 'Interm. TR Redux<br/>TR GB Redux'MR hp<br/>TR GB Redux'MR pm<br/>TR GB Redux'MR pm<br/>TR GB Redux'TR GB Redux<br/>TR hp 'Interm. TR Redux<br/>TR hp 'ITR GB Redux<br/>TR hp 'TR BR Redux</td><td>Scaled 
<br/>16<br/>-1<br/>-1<br/>-2<br/>-2<br/>-4<br/>-4<br/>-5<br/>-7<br/>-7<br/>-48<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2</td><td>Estimat<br/>8041.81<br/>39.554<br/>139.554<br/>139.554<br/>13.008<br/>13.008<br/>13.008<br/>13.008<br/>13.008<br/>13.008<br/>13.008<br/>13.008<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360<br/>1.360</td><td></td><td>9011<br/>2.851<br/>2.855<br/>2.855<br/>2.857<br/>2.857<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3</td><td>i928         17           i087         14           i087         4           i087         5           i087         5           i087         5           i087         5           i0854         5           i027         5</td><td>79.28<br/>48.95<br/>42.61<br/>35.15<br/>44.37<br/>75.73<br/>0.48<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>74.65<br/>30.48<br/>-2.00<br/>74.65<br/>30.48<br/>-2.00<br/>21.24<br/>43.28<br/>27.57<br/>-0.00<br/>23.89<br/>-7.14<br/>58.87<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.000<br/>0.000<br/>0.000</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></tr> <tr><td>Scaled Estimates<br/>Scaled Estimates<br/>Continuous factors centered by mean, sca<br/>Term<br/>Intercept<br/>Intern. TR Redux<br/>MR hp<br/>MR rpm<br/>Planetary Redux<br/>TR GB Redux<br/>TR GB Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "MR rpm<br/>Planetary Redux "Interm. TR Redux<br/>MR rpm "MR rpm<br/>Planetary Redux "MR rpm<br/>Planetary Redux "MR hp<br/>Planetary Redux "MR hp<br/>Planetary Redux "MR rpm<br/>Planetary Redux "MR hp<br/>Planetary Redux "MR hp<br/>Planetary Redux "MR rpm<br/>TR GB Redux"MR rpm<br/>TR GB Redux"MR rpm<br/>TR GB Redux"MR rpm<br/>TR GB Redux"TR GB Redux<br/>TR GB Redux"TR GB Redux<br/>TR hp "Interm. TR Redux<br/>TR hp "MR hp<br/>TR hp "MR rpm<br/>TR hp "MR rpm<br/>TR hp "TR GB Redux<br/>TR hp "TR GB Redux<br/>TR hp "TR GB
Redux</td><td>Scaled<br/>16<br/>-1<br/>-1<br/>-1<br/>-2<br/>21<br/>21<br/>-2<br/>-2<br/>-3<br/>-3<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7</td><td>Estimat<br/>4041.81<br/>39.554<br/>39.554<br/>39.554<br/>5.9169<br/>1.23e-1<br/>5.9169<br/>1.23e-1<br/>5.9169<br/>1.23e-1<br/>1.36e-<br/>6.1199<br/>7.40.04<br/>90.4994<br/>4.136e-<br/>6.1199<br/>7.40.937<br/>11.0860<br/>7.4.177<br/>19.0269<br/>2.809e-<br/>13.600<br/>4.9337<br/>11.0860<br/>7.4.177<br/>19.0269<br/>2.809e-<br/>13.600<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.177<br/>11.0860<br/>7.4.1777<br/>11.0860<br/>7.4.1777<br/>11.0860<br/>11.0860<br/>11.0860<br/>11.0860<br/>11.0860<br/>11.0860<br/>11.0860</td><td></td><td>9015<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.491<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3.493<br/>3</td><td>i928         17           i087         14           i087         4           i087         -1           i227         -1           i854         -1      i854         -1<!--</td--><td>79.28<br/>48.95<br/>42.61<br/>35.15<br/>44.37<br/>75.73<br/>0.46<br/>-0.00<br/>74.65<br/>30.46<br/>-0.00<br/>74.65<br/>30.46<br/>-3.89<br/>05.06<br/>-3.89<br/>05.06<br/>-7.14<br/>23.45<br/>21.24<br/>21.24<br/>23.45<br/>21.25<br/>7.74<br/>23.45<br/>21.25<br/>7.74<br/>23.45<br/>21.25<br/>7.74<br/>20.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></td></tr> <tr><td>L<br/>Scaled Estimates<br/>Continuous factors centered by mean, sca<br/>Term<br/>Intercept<br/>Intern: TR Redux<br/>MR hp<br/>MR rpm<br/>Pilanetary Redux<br/>TR GB Redux<br/>TR GB Redux<br/>TR GB Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "Interm. TR Redux<br/>MR hp "Interm. TR Redux<br/>MR rpm "Interm. TR Redux<br/>MR rpm "Interm. TR Redux<br/>MR rpm "MR hp<br/>Pianetary Redux "MR hp<br/>Pianetary Redux "MR hp<br/>Pianetary Redux "MR hp<br/>Pianetary Redux "MR rpm<br/>Pianetary Redux "MR rpm<br/>Pianetary Redux "MR rpm<br/>Pianetary Redux "MR rpm<br/>Pianetary Redux "MR rpm<br/>TR GB Redux"Haren TR Redux<br/>TR GB Redux"Haren TR Redux<br/>TR GB Redux"Pianetary Redux<br/>TR GB Redux"Pianetary Redux<br/>TR GB Redux"TR GB Redux<br/>TR hp "Interm. TR Redux<br/>TR hp "MR rpm<br/>TR hp "MR rpm<br/>TR hp "Pianetary Redux<br/>TR hp "TR GB Redux<br/>TR hp "TR Mp<br/>TR rpm "Interm. TR Redux<br/>TR hp "TR Mp</td><td>Scaled
I<br/>16<br/>-1<br/>44<br/>-2<br/>21<br/>21<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-<br/>-</td><td>Estimat<br/>Estimation<br/>139.554<br/>130.008<br/>139.554<br/>130.008<br/>139.554<br/>130.008<br/>139.554<br/>130.008<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169</td><td></td><td>9011<br/>2.851<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/>2.855<br/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        17           i087         14           i087         14           i087         4           i087         -1           i227         5           854         -2           1227         5           854         11           854         -2           1227         5           854         5           854         5           854         5           854         5           854         8           854         8           854         8           854         8           854         8           854         8           854         8           854         8           854         8      854         8      854</td><td>79.28<br/>48.95<br/>48.95<br/>44.37<br/>75.73<br/>-0.00<br/>74.65<br/>30.48<br/>99.32<br/>-7.14<br/>23.45<br/>99.32<br/>-7.14<br/>23.45<br/>38.28<br/>27.57<br/>38.28<br/>27.57<br/>38.28<br/>27.57<br/>38.28<br/>27.57<br/>38.28<br/>0.00<br/>-3.89<br/>-7.14<br/>58.87<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></tr> <tr><td>L<br/>Scaled Estimates<br/>Continuous factors centered by mean, sca<br/>Term<br/>Intercept<br/>Intern. TR Redux<br/>MR hp<br/>MR pm<br/>Planetary Redux<br/>TR GB Redux<br/>TR fpm<br/>Interm. TR Redux 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR trpm 'MR hp<br/>MR rpm 'MR hp<br/>Planetary Redux 'Interm. TR Redux<br/>Planetary Redux 'MR hp<br/>Planetary Redux 'MR trpm<br/>Planetary Redux 'MR trpm<br/>TR GB Redux'MR trpm<br/>TR GB Redux'MR trpm<br/>TR GB Redux'TR GB Redux<br/>TR hp 'Interm. TR Redux<br/>TR hp 'Interm. TR Redux<br/>TR hp 'Interm. TR Redux<br/>TR hp 'TR Ap<br/>TR hp 'TR Ap<br/>TR trpm 'Interm. TR Redux<br/>TR hp 'TR hp<br/>TR trpm 'Interm. TR Redux<br/>TR hp 'TR hp<br/>TR trpm 'Interm. TR Redux<br/>TR hp 'TR hp<br/>TR trpm 'MR trpm<br/>TR trpm 'MR trpm<br/>TR trpm 'MR trpm</td><td>Scaled  <br/>16<br/>-1<br/>41<br/>-2<br/>12<br/>21<br/>-4<br/>-4<br/>15<br/>-5<br/>-7<br/>49<br/>-2<br/>49<br/>-2<br/>25<br/>7<br/>7<br/>8<br/>8<br/>8<br/>5<br/>5<br/>34<br/>11<br/>-2<br/>-4<br/>-4<br/>-2<br/>-2<br/>-7<br/>-7<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2<br/>-2</td><td>Estimat<br/>Estimation 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1,
1,</td><td></td><td>9011<br/>2.851<br/>2.851<br/>2.855<br/>2.857<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3</td><td>i928         17           i087         14           i087         4           i087         -           i227         -           854         -           854         -           854         -           854         -           854         -           854         -           854         -           9227         -           854         -           854         -           854         -           854         -           854         -           <t< td=""><td>79.28<br/>48.95<br/>48.95<br/>44.37<br/>75.73<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>11.36<br/>99.32<br/>-7.14<br/>23.45<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>57.71<br/>-0.00<br/>-3.89<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.0000<br/>0.0000<br/>0.0000<br/>0.000000</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></t<></td></tr> <tr><td>L<br/>Scaled Estimates<br/>Scaled Estimates<br/>Continuous factors centered by mean, sca<br/>Term<br/>Intercept<br/>Intern. TR Redux<br/>MR hp<br/>MR rpm<br/>Planetary Redux<br/>TR GB Redux<br/>TR GB Redux<br/>MR hp 'INR MR hp<br/>MR rpm 'Interm. TR Redux<br/>MR hp 'INR mm<br/>MR rpm 'MR rpm<br/>Planetary Redux 'Intern. TR Redux<br/>MR rpm 'MR rpm<br/>Planetary Redux 'MR rpm<br/>TR GB Redux'INterm. TR Redux<br/>TR GB Redux'INterm<br/>TR hp 'Interm. TR Redux<br/>TR hp 'Interm. TR Redux<br/>TR hp 'INT GB Redux<br/>TR hp 'INT hp<br/>TR hp 'INT hp<br/>TR rpm 'MR hp<br/>TR rpm 'MR rpm<br/>TR rpm 'Intertary Redux</td><td>Scaled<br/>16<br/>-1<br/>-1<br/>-1<br/>-2<br/>-2<br/>-2<br/>-2<br/>-3<br/>-3<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7<br/>-7</td><td>Estimat<br/>Estimation 1997
940<br/>139.554<br/>133.008<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.9169<br/>15.916<br/>15.917<br/>10.0859<br/>11.0860<br/>11.0800<br/>14.9337<br/>11.0860<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0269<br/>11.0</td><td></td><td>9015<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/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        17           i087         -           i227         -           854         -           i227         -           854         -           i227         -           854         -           i227         -           854         -           1227         -           854         -           1227         -           854         -           1227         -           854         -           1227         -           854         -           854         -           854         -           854         -           854         -           854         -           854         -           854         -           854         -      854<td>79.28<br/>48.95<br/>48.95<br/>44.37<br/>75.73<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>11.36<br/>99.32<br/>-7.14<br/>58.87<br/>0.00<br/>-3.89<br/>-7.14<br/>58.87<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></td></tr> <tr><td>L<br/>Scaled Estimates<br/>Continuous factors centered by mean, sca<br/>Term<br/>Intercept<br/>Intern. TR Redux<br/>MR hp<br/>MR pm<br/>Planetary Redux<br/>TR GB Redux<br/>TR fpm<br/>Interm. TR Redux 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR hp 'Interm. TR Redux<br/>MR trpm 'MR hp<br/>MR rpm 'MR hp<br/>Planetary Redux 'Interm. TR Redux<br/>Planetary Redux 'MR hp<br/>Planetary Redux 'MR trpm<br/>Planetary Redux 'MR trpm<br/>TR GB Redux'MR trpm<br/>TR GB Redux'MR trpm<br/>TR GB Redux'TR GB Redux<br/>TR hp 'Interm. TR Redux<br/>TR hp 'Interm. TR Redux<br/>TR hp 'Interm. TR Redux<br/>TR hp 'TR Ap<br/>TR hp 'TR Ap<br/>TR trpm 'Interm. TR Redux<br/>TR hp 'TR hp<br/>TR trpm 'Interm. TR Redux<br/>TR hp 'TR hp<br/>TR trpm 'Interm. TR Redux<br/>TR hp 'TR hp<br/>TR trpm 'MR trpm<br/>TR trpm 'MR trpm<br/>TR trpm 'MR trpm</td><td>Scaled I<br/>16<br/>1-1<br/>44<br/>12<br/>21<br/>21<br/>21<br/>21<br/>21<br/>21<br/>21<br/>21<br/>21</td><td>Estimat<br/>Estimation 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1,
1,</td><td></td><td>9011<br/>2.851<br/>2.851<br/>2.855<br/>2.857<br/>2.851<br/>2.851<br/>2.851<br/>2.851<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3.491<br/>3</td><td>1928         17           1087         14           1087         14           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12      1227         12</td><td>79.28<br/>48.95<br/>48.95<br/>44.37<br/>75.73<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>11.36<br/>99.32<br/>-7.14<br/>23.45<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>57.71<br/>-0.00<br/>-3.89<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.0000<br/>0.0000<br/>0.0000<br/>0.0000<br/>0.0000<br/>0.000000</td><td>Proc<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.</td></tr> | 79.28<br>48.95<br>42.61<br>35.15<br>44.37<br>75.73<br>-0.00<br>11.36<br>30.48<br>-0.00<br>11.36<br>99.32<br>-7.14<br>23.45<br>99.32<br>27.57<br>-0.00<br>99.32<br>27.57<br>43.88<br>27.57<br>-0.00<br>-7.14<br>58.87<br>0.00<br>0.00<br>0.00                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                                          | 0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0               | Scaled Estimates<br>Scaled Estimates<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR npm<br>Planetary Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "MR hp<br>MR npm "Interm. TR Redux<br>MR npm "MR hp<br>Planetary Redux "MR npm<br>Planetary Redux "MR hp<br>Planetary Redux "MR hp<br>Planetary Redux "MR npm<br>Planetary Redux "MR npm<br>Planetary Redux "MR npm<br>Planetary Redux "MR npm<br>TR GB Redux"MR npm<br>TR GB Redux"MR npm<br>TR GB Redux"Hanetary Redux<br>TR GB Redux"Hanetary Redux<br>TR GB Redux"TR GB Redux<br>TR GB Redux"TR GB Redux<br>TR hp "MR npm<br>TR hp "MR npm<br>TR hp "MR npm<br>TR hp "MR npm | Scaled  <br>16<br>-1<br>41<br>-2<br>12<br>21<br>-4<br>45<br>-5<br>-7<br>499<br>-2<br>43<br>-3<br>-3<br>-4<br>-2<br>20<br>-7<br>49<br>-2<br>-2<br>20<br>-7<br>-8<br>-8<br>-9<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2 |
Estimat<br>8041.81<br>39.554<br>13.008<br>381.07<br>766.939<br>5.9169<br>15.9169<br>1.23e-1<br>97.940<br>00.4994<br>1.3600<br>116.038<br>00.4597<br>11.0860<br>74.177<br>11.0860<br>2.25028<br>2.809e-<br>13.609<br>0.25828<br>2.809e-<br>13.609<br>0.25828<br>2.809e-<br>13.609<br>0.25828<br>2.809e-<br>13.600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.3600<br>1.36000<br>1.36000<br>1.36000<br>1.36000<br>1.36000<br>1.36000<br>1.36000<br>1.36000000000000000000000000000000000000 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| i928         17           i087         14           i087         4           i087         -1           i227         -1           i2454         -1           i254         -1           i2454         -1           i254         -1           i254         -1           i254         -1           i254         -1           i254         -1      i254         -1 | 79.28<br>48.95<br>42.61<br>35.15<br>44.37<br>75.73<br>0.000<br>74.65<br>30.48<br>-0.00<br>74.65<br>30.48<br>-0.00<br>74.65<br>30.48<br>-0.00<br>99.32<br>21.24<br>38.28<br>99.32<br>21.24<br>38.28<br>57.57<br>-0.00<br>-3.89<br>0.00<br>-3.89<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00 | 0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0 | Scaled Estimates<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern: TR Redux<br>MR hp<br>MR rpm<br>Pilanetary Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR rpm "Interm. TR Redux<br>MR rpm "MR rpm<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>TR GB Redux"Haren TR Redux<br>TR GB Redux"TR RGB Redux<br>TR hp "Interm. TR Redux<br>TR hp "MR rpm<br>TR hp "MR rpm<br>TR hp "MR rpm<br>TR hp "MR rpm<br>TR hp "TR GB Redux | Scaled  <br>16<br>-1<br>41<br>-2<br>12<br>21<br>21<br>21<br>21<br>21<br>21<br>21<br>21<br>2 | Estimat<br>0041.81<br>39.554<br>13.008<br>3381.07<br>260.339<br>1.23e-1<br>97.940<br>5.9189<br>1.23e-1<br>1.360-<br>6.1199<br>11.360-<br>6.1199<br>113.600<br>0.4597<br>11.0660<br>2.25828<br>2.809e-<br>113.600<br>0.4933<br>10.0289<br>2.25828<br>2.809e-<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028<br>11.6028 |  |
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TR Redux<br>MR hp<br>MR pm<br>Planetary Redux<br>TR GB Redux<br>TR fpm<br>Interm. TR Redux 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR pm 'Interm. TR Redux<br>Planetary Redux 'MR hp<br>Planetary Redux 'Interm. TR Redux<br>TR GB Redux'MR hp<br>TR GB Redux'MR pm<br>TR GB Redux'MR pm<br>TR GB Redux'TR GB Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'ITR GB Redux<br>TR hp 'TR BR Redux | Scaled  <br>16<br>-1<br>-1<br>-2<br>-2<br>-4<br>-4<br>-5<br>-7<br>-7<br>-48<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2 | Estimat<br>8041.81<br>39.554<br>139.554<br>139.554<br>13.008<br>13.008<br>13.008<br>13.008<br>13.008<br>13.008<br>13.008<br>13.008<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360<br>1.360 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9011<br>2.851<br>2.855<br>2.855<br>2.857<br>2.857<br>2.851<br>2.851<br>2.851<br>2.851<br>2.851<br>2.851<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3 | i928         17           i087         14           i087         4           i087         5           i087         5           i087         5           i087         5           i0854         5           i027         5 | 79.28<br>48.95<br>42.61<br>35.15<br>44.37<br>75.73<br>0.48<br>-0.00<br>74.65<br>30.48<br>-0.00<br>74.65<br>30.48<br>-0.00<br>74.65<br>30.48<br>-2.00<br>74.65<br>30.48<br>-2.00<br>21.24<br>43.28<br>27.57<br>-0.00<br>23.89<br>-7.14<br>58.87<br>0.00<br>0.00<br>0.00<br>0.00<br>0.000<br>0.000<br>0.000 | 0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0 | Scaled Estimates<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR rpm<br>Planetary Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "MR rpm<br>Planetary Redux "Interm. TR Redux<br>MR rpm "MR rpm<br>Planetary Redux "MR rpm<br>Planetary Redux "MR hp<br>Planetary Redux "MR hp<br>Planetary Redux "MR rpm<br>Planetary Redux "MR hp<br>Planetary Redux "MR hp<br>Planetary
Redux "MR rpm<br>TR GB Redux"MR rpm<br>TR GB Redux"MR rpm<br>TR GB Redux"MR rpm<br>TR GB Redux"TR GB Redux<br>TR GB Redux"TR GB Redux<br>TR hp "Interm. TR Redux<br>TR hp "MR hp<br>TR hp "MR rpm<br>TR hp "MR rpm<br>TR hp "TR GB Redux<br>TR hp "TR GB Redux<br>TR hp "TR GB Redux | Scaled<br>16<br>-1<br>-1<br>-1<br>-2<br>21<br>21<br>-2<br>-2<br>-3<br>-3<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7 | Estimat<br>4041.81<br>39.554<br>39.554<br>39.554<br>5.9169<br>1.23e-1<br>5.9169<br>1.23e-1<br>5.9169<br>1.23e-1<br>1.36e-<br>6.1199<br>7.40.04<br>90.4994<br>4.136e-<br>6.1199<br>7.40.937<br>11.0860<br>7.4.177<br>19.0269<br>2.809e-<br>13.600<br>4.9337<br>11.0860<br>7.4.177<br>19.0269<br>2.809e-<br>13.600<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.177<br>11.0860<br>7.4.1777<br>11.0860<br>7.4.1777<br>11.0860<br>11.0860<br>11.0860<br>11.0860<br>11.0860<br>11.0860<br>11.0860 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| i928         17           i087         14           i087         4           i087         -1           i227         -1           i854         -1      i854         -1 </td <td>79.28<br/>48.95<br/>42.61<br/>35.15<br/>44.37<br/>75.73<br/>0.46<br/>-0.00<br/>74.65<br/>30.46<br/>-0.00<br/>74.65<br/>30.46<br/>-3.89<br/>05.06<br/>-3.89<br/>05.06<br/>-7.14<br/>23.45<br/>21.24<br/>21.24<br/>23.45<br/>21.25<br/>7.74<br/>23.45<br/>21.25<br/>7.74<br/>23.45<br/>21.25<br/>7.74<br/>20.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.</td> <td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td> | 79.28<br>48.95<br>42.61<br>35.15<br>44.37<br>75.73<br>0.46<br>-0.00<br>74.65<br>30.46<br>-0.00<br>74.65<br>30.46<br>-3.89<br>05.06<br>-3.89<br>05.06<br>-7.14<br>23.45<br>21.24<br>21.24<br>23.45<br>21.25<br>7.74<br>23.45<br>21.25<br>7.74<br>23.45<br>21.25<br>7.74<br>20.00<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00<br>0. | 0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0 | L<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern: TR Redux<br>MR hp<br>MR rpm<br>Pilanetary Redux<br>TR GB Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR rpm "Interm. TR Redux<br>MR rpm "Interm. TR Redux<br>MR rpm "MR hp<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>TR GB Redux"Haren TR Redux<br>TR GB Redux"Haren TR Redux<br>TR GB Redux"Pianetary Redux<br>TR GB Redux"Pianetary Redux<br>TR GB Redux"TR GB Redux<br>TR hp "Interm. TR Redux<br>TR hp "MR rpm<br>TR hp "MR rpm<br>TR hp "Pianetary Redux<br>TR hp "TR GB Redux<br>TR hp "TR Mp<br>TR rpm "Interm. TR Redux<br>TR hp "TR Mp | Scaled I<br>16<br>-1<br>44<br>-2<br>21<br>21<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>- |
Estimat<br>Estimation<br>139.554<br>130.008<br>139.554<br>130.008<br>139.554<br>130.008<br>139.554<br>130.008<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169 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| i928         17           i087         14           i087         14           i087         4           i087         -1           i227         5           854         -2           1227         5           854         11           854         -2           1227         5           854         5           854         5           854         5           854         5           854         8           854         8           854         8           854         8           854         8           854         8           854         8           854         8           854         8      854         8      854 | 79.28<br>48.95<br>48.95<br>44.37<br>75.73<br>-0.00<br>74.65<br>30.48<br>99.32<br>-7.14<br>23.45<br>99.32<br>-7.14<br>23.45<br>38.28<br>27.57<br>38.28<br>27.57<br>38.28<br>27.57<br>38.28<br>27.57<br>38.28<br>0.00<br>-3.89<br>-7.14<br>58.87<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00 | 0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0 | L<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR pm<br>Planetary Redux<br>TR GB Redux<br>TR fpm<br>Interm. TR Redux 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR trpm 'MR hp<br>MR rpm 'MR hp<br>Planetary Redux 'Interm. TR Redux<br>Planetary Redux 'MR hp<br>Planetary Redux 'MR trpm<br>Planetary Redux 'MR trpm<br>TR GB Redux'MR trpm<br>TR GB Redux'MR trpm<br>TR GB Redux'TR GB Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'TR Ap<br>TR hp 'TR Ap<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'MR trpm<br>TR trpm 'MR trpm<br>TR trpm 'MR trpm | Scaled  <br>16<br>-1<br>41<br>-2<br>12<br>21<br>-4<br>-4<br>15<br>-5<br>-7<br>49<br>-2<br>49<br>-2<br>25<br>7<br>7<br>8<br>8<br>8<br>5<br>5<br>34<br>11<br>-2<br>-4<br>-4<br>-2<br>-2<br>-7<br>-7<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2<br>-2 | Estimat<br>Estimation 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, |  |
9011<br>2.851<br>2.851<br>2.855<br>2.857<br>2.851<br>2.851<br>2.851<br>2.851<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3 | i928         17           i087         14           i087         4           i087         -           i227         -           854         -           854         -           854         -           854         -           854         -           854         -           854         -           9227         -           854         -           854         -           854         -           854         -           854         - <t< td=""><td>79.28<br/>48.95<br/>48.95<br/>44.37<br/>75.73<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>11.36<br/>99.32<br/>-7.14<br/>23.45<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>21.24<br/>38.28<br/>57.71<br/>-0.00<br/>-3.89<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.000<br/>0.0000<br/>0.0000<br/>0.0000<br/>0.000000</td><td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td></t<> | 79.28<br>48.95<br>48.95<br>44.37<br>75.73<br>-0.00<br>74.65<br>30.48<br>-0.00<br>11.36<br>99.32<br>-7.14<br>23.45<br>21.24<br>38.28<br>21.24<br>38.28<br>21.24<br>38.28<br>57.71<br>-0.00<br>-3.89<br>0.00<br>0.00<br>0.00<br>0.00<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.0000<br>0.0000<br>0.0000<br>0.000000 | 0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0 | L<br>Scaled Estimates<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR rpm<br>Planetary Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp 'INR MR hp<br>MR rpm 'Interm. TR Redux<br>MR hp 'INR mm<br>MR rpm 'MR rpm<br>Planetary Redux 'Intern. TR Redux<br>MR rpm 'MR rpm<br>Planetary Redux 'MR rpm<br>TR GB Redux'INterm. TR Redux<br>TR GB Redux'INterm<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'INT GB Redux<br>TR hp 'INT hp<br>TR hp 'INT hp<br>TR rpm 'MR hp<br>TR rpm 'MR rpm<br>TR rpm 'Intertary Redux | Scaled<br>16<br>-1<br>-1<br>-1<br>-2<br>-2<br>-2<br>-2<br>-3<br>-3<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7 | Estimat<br>Estimation 1997 940<br>139.554<br>133.008<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.916<br>15.917<br>10.0859<br>11.0860<br>11.0800<br>14.9337<br>11.0860<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0 |  |
9015<br>2.851<br>2.851<br>2.851<br>2.851<br>2.851<br>2.851<br>2.851<br>2.851<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3 | i928         17           i087         -           i227         -           854         -           i227         -           854         -           i227         -           854         -           i227         -           854         -           1227         -           854         -           1227         -           854         -           1227         -           854         -           1227         -           854         -           854         -           854         -           854         -           854         -           854         -           854         -           854         -           854         -      854 <td>79.28<br/>48.95<br/>48.95<br/>44.37<br/>75.73<br/>-0.00<br/>74.65<br/>30.48<br/>-0.00<br/>11.36<br/>99.32<br/>-7.14<br/>58.87<br/>0.00<br/>-3.89<br/>-7.14<br/>58.87<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00<br/>0.00</td> <td>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0<br/>0.0</td> | 79.28<br>48.95<br>48.95<br>44.37<br>75.73<br>-0.00<br>74.65<br>30.48<br>-0.00<br>11.36<br>99.32<br>-7.14<br>58.87<br>0.00<br>-3.89<br>-7.14<br>58.87<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00<br>0.00 | 0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0<br>0.0 | L<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR pm<br>Planetary Redux<br>TR GB Redux<br>TR fpm<br>Interm. TR Redux 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR trpm 'MR hp<br>MR rpm 'MR hp<br>Planetary Redux 'Interm. TR Redux<br>Planetary Redux 'MR hp<br>Planetary Redux 'MR trpm<br>Planetary Redux 'MR trpm<br>TR GB Redux'MR trpm<br>TR GB Redux'MR trpm<br>TR GB Redux'TR GB Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'TR Ap<br>TR hp 'TR Ap<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'MR trpm<br>TR trpm 'MR trpm<br>TR trpm 'MR trpm | Scaled I<br>16<br>1-1<br>44<br>12<br>21<br>21<br>21<br>21<br>21<br>21<br>21<br>21<br>21 | Estimat<br>Estimation 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, |  | 9011<br>2.851<br>2.851<br>2.855<br>2.857<br>2.851<br>2.851<br>2.851<br>2.851<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3.491<br>3 | 1928         17           1087         14           1087         14           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1087         4           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12           1227         12      1227         12 |
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| Scaled Estimates<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern: TR Redux<br>MR hp<br>MR rpm<br>Pilanetary Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR rpm "Interm. TR Redux<br>MR rpm "MR rpm<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>TR GB Redux"Haren TR Redux<br>TR GB Redux"TR RGB Redux<br>TR hp "Interm. 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| Scaled Estimates<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR rpm<br>Planetary Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "MR rpm<br>Planetary Redux "Interm. TR Redux<br>MR rpm "MR rpm<br>Planetary Redux "MR rpm<br>Planetary Redux "MR hp<br>Planetary Redux "MR hp<br>Planetary Redux "MR rpm<br>Planetary Redux "MR hp<br>Planetary Redux "MR hp<br>Planetary Redux "MR rpm<br>TR GB Redux"MR rpm<br>TR GB Redux"MR rpm<br>TR GB Redux"MR rpm<br>TR GB Redux"TR GB Redux<br>TR GB Redux"TR GB Redux<br>TR hp "Interm. 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| L<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern: TR Redux<br>MR hp<br>MR rpm<br>Pilanetary Redux<br>TR GB Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR hp "Interm. TR Redux<br>MR rpm "Interm. TR Redux<br>MR rpm "Interm. TR Redux<br>MR rpm "MR hp<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR hp<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>Pianetary Redux "MR rpm<br>TR GB Redux"Haren TR Redux<br>TR GB Redux"Haren TR Redux<br>TR GB Redux"Pianetary Redux<br>TR GB Redux"Pianetary Redux<br>TR GB Redux"TR GB Redux<br>TR hp "Interm. TR Redux<br>TR hp "MR rpm<br>TR hp "MR rpm<br>TR hp "Pianetary Redux<br>TR hp "TR GB Redux<br>TR hp "TR Mp<br>TR rpm "Interm. TR Redux<br>TR hp "TR Mp                                                                                                                                                   | Scaled I<br>16<br>-1<br>44<br>-2<br>21<br>21<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-<br>-                                                                                                                                                                                                                                   | 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| L<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR pm<br>Planetary Redux<br>TR GB Redux<br>TR fpm<br>Interm. TR Redux 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR trpm 'MR hp<br>MR rpm 'MR hp<br>Planetary Redux 'Interm. TR Redux<br>Planetary Redux 'MR hp<br>Planetary Redux 'MR trpm<br>Planetary Redux 'MR trpm<br>TR GB Redux'MR trpm<br>TR GB Redux'MR trpm<br>TR GB Redux'TR GB Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'TR Ap<br>TR hp 'TR Ap<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'Interm. 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79.28<br>48.95<br>48.95<br>44.37<br>75.73<br>-0.00<br>74.65<br>30.48<br>-0.00<br>11.36<br>99.32<br>-7.14<br>23.45<br>21.24<br>38.28<br>21.24<br>38.28<br>21.24<br>38.28<br>57.71<br>-0.00<br>-3.89<br>0.00<br>0.00<br>0.00<br>0.00<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.000<br>0.0000<br>0.0000<br>0.0000<br>0.000000 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| L<br>Scaled Estimates<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR rpm<br>Planetary Redux<br>TR GB Redux<br>TR GB Redux<br>MR hp 'INR MR hp<br>MR rpm 'Interm. TR Redux<br>MR hp 'INR mm<br>MR rpm 'MR rpm<br>Planetary Redux 'Intern. TR Redux<br>MR rpm 'MR rpm<br>Planetary Redux 'MR rpm<br>TR GB Redux'INterm. TR Redux<br>TR GB Redux'INterm<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'INT GB Redux<br>TR hp 'INT hp<br>TR hp 'INT hp<br>TR rpm 'MR hp<br>TR rpm 'MR rpm<br>TR rpm 'Intertary Redux | Scaled<br>16<br>-1<br>-1<br>-1<br>-2<br>-2<br>-2<br>-2<br>-3<br>-3<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7<br>-7                                                                                                                                                                                                          | Estimat<br>Estimation 1997 940<br>139.554<br>133.008<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.9169<br>15.916<br>15.917<br>10.0859<br>11.0860<br>11.0800<br>14.9337<br>11.0860<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0269<br>11.0 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| L<br>Scaled Estimates<br>Continuous factors centered by mean, sca<br>Term<br>Intercept<br>Intern. TR Redux<br>MR hp<br>MR pm<br>Planetary Redux<br>TR GB Redux<br>TR fpm<br>Interm. TR Redux 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR hp 'Interm. TR Redux<br>MR trpm 'MR hp<br>MR rpm 'MR hp<br>Planetary Redux 'Interm. TR Redux<br>Planetary Redux 'MR hp<br>Planetary Redux 'MR trpm<br>Planetary Redux 'MR trpm<br>TR GB Redux'MR trpm<br>TR GB Redux'MR trpm<br>TR GB Redux'TR GB Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'Interm. TR Redux<br>TR hp 'TR Ap<br>TR hp 'TR Ap<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'Interm. TR Redux<br>TR hp 'TR hp<br>TR trpm 'MR trpm<br>TR trpm 'MR trpm<br>TR trpm 'MR trpm                                                                                                                                        | Scaled I<br>16<br>1-1<br>44<br>12<br>21<br>21<br>21<br>21<br>21<br>21<br>21<br>21<br>21                                                                                                                                                                                                                                                                   | Estimat<br>Estimation 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1, 1,                                                                                                                                                                                                                                                                                                                                                                                   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Table 58: RSE Model Fit for Planetary Drive (continued)

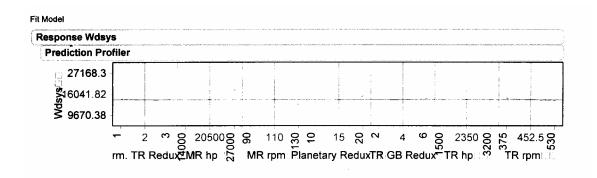


Table 58: RSE Model Fit for Planetary Drive (continued)

# APPENDIX G: MODEL FIT FOR SPLIT TORQUE DRIVE

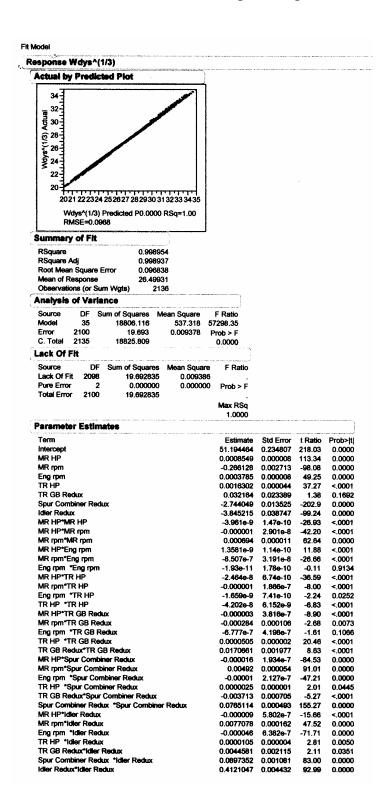


Table 59: RSE Model Fit for Split Torque Drive

# Table 59: RSE Model Fit for Split Torque Drive (continued)

esponse Wdys^(1/3)					
Effect Tests					
Source	Nparm	DF	Sum of Squares	F Ratio	Prob > F
MRHP	1	1	120.46875	12846.52	0.000
MR rpm	1	1	90.20164	9618.902	0.000
Engrpm	1	1	22.74965	2425.972	0.000
TRHP	1	1	13.02919	1389.403	<.000
TR GB Redux	1	1	0.01773	1.8911	0.169
Spur Combiner Redux	1	1	386.02378	41164.71	0.000
Idler Redux	1	1	92.35528	9848.561	0.000
MR HP*MR HP	1	1	6.80222	725.3730	<.000*
MR HP*MR nom	1	1	16.70164	1781.026	<.000
MR rpm*MR rpm	1	1	36.79433	3923.666	0.000
MR HP*Eng rpm	1	1	1.32295	141.0766	<.000
MR rpm*Eng rpm	1	1	6.66401	710.6354	<.000
Eng rpm *Eng rpm	1	1	0.00011	0.0118	0.913
MR HP*TR HP	1	1	12.55514	1338.853	<.000
MR rpm*TR HP	.1	1	0.59999	63.9813	<.000
Engrpm *TR HP	1	1	0.04704	5.0159	0.025
TR HP *TR HP	1	1	0.43763	46.6676	<.000
MR HP*TR GB Redux	1	1	0.74236	79.1636	<.000*
MR rpm*TR GB Redux	1	1	0.06752	7.2002	0.007
Eng rpm *TR GB Redux	1	1	0.02444	2.6059	0.106
TR HP *TR GB Redux	1	1	3.92458	418.5087	<.000
TR GB Redux*TR GB Redux	1	1	0.69893	74.5326	<.000
MR HP*Spur Combiner Redux	1	1	67.00271	7145.019	0.000
MR rpm*Spur Combiner Redux	1	1	77.66891	8282.439	0.000
Eng rpm *Spur Combiner Redux	1	1	20.90294	2229.043	0.000
TR HP *Spur Combiner Redux	1	1	0.03792	4.0441	0.044
TR GB Redux*Spur Combiner Redux	1	1	0.26025	27.7529	<.000
Spur Combiner Redux *Spur Combiner Redux	1	1	226.07964	24108.63	0.000
MR HP*Idler Redux	1	1	2.30065	245.3363	<.0001
MR rpm*ldler Redux	1	1	21.18034	2258.624	0.000
Eng rpm *Idler Redux	1	1	48.22748	5142.87	0.000
TR HP *Idler Redux	1	1	0.07408	7.8996	0.005
TR GB Redux*Idler Redux	1	1	0.04167	4.4440	0.035
Spur Combiner Redux *Idler Redux	1	1	64.59434	6888.196	0.000
klier Redux*klier Redux	1	1	81.08295	8646.505	0.000
Residual by Predicted Plot	)				
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# Table 59: RSE Model Fit for Split Torque Drive (continued)

sponse Wo	tys^(	1/3)											
Scaled Esti									··				
Continuous fa	ctors o	entered t	ov mea	n. sc	aled by rano	e/2							
Term						aled Estimate		Std Error	t Ratio	Prob>iti			
Intercept							25.701599				0.00807	3184.85	0.0000
MR HP							1.7906927	ł			0.00257	696.65	0.0000
MRrpm							-1.644905			- 1	0.00259	-635.05	0.0000
Eng rpm							0.3337001				0.002571	129.79	0.0000
TRHP							0.8465738		1.1		0.002568	329.72	0.0000
TR GB Redu	x						0.2081232				0.002566	81.11	0.0000
Spur Combin	er Red	ux					-2.638655				0.00259	-1018.98	0.0000
Idler Redux							-0.450314				0.00259	-173.86	0.0000
MR HP*MR H	IP						-0.119822		·		0.004449	-26.93	<.0001
MR HP*MR n	pm						-0.134667		L		0.003191	-42.20	<.0001
MR rpm*MR	mqn						0.277587	1			0.004432	62.64	0.0000
MR HP*Eng r	npm						0.0373484			i	0.003144	11.88	<.0001
MR rpm*Eng	rpm						-0.085068				0.003191	-26.66	<.0001
Eng rpm *En	ng rpm						-0.000484		_		0.004445	-0.11	0.9134
MR HP*TR H	IP .						-0.115209				0.003149	-36.59	<.0001
MR rpm*TR I	HP						-0.025376				0.003173	-8.00	<.0001
Engrpm *TR	R HP						-0.007052				0.003149	-2.24	0.0252
TR HP "TR I	HP						-0.030363				0.004445	-6.83	<.0001
MR HP*TR G	B Red	ux					-0.028009		1		0.003148	-8.90	<.0001
MR rpm*TR 0	GB Re	tux					-0.008512				0.003172	-2.68	0.0073
Eng rpm *TF	r gb r	edux					-0.005083			1	0.003149	-1.61	0.1066
TR HP *TR (	GB Re	dux					0.064328				0.003144	20.46	<.0001
TR GB Redu	x*TR G	B Redux					0.0383987		J		0.004448	8.63	<.0001
MR HP*Spur	Comb	iner Redu	x				-0.269699		L		0.003191	-84.53	0.0000
MR rpm*Spu	r Coml	viner Red	lux 🛛				0.2951979		J	1	0.003244	91.01	0.0000
Engrpm *Sp	our Cor	nbiner Re	odux				-0.150662		1		0.003191	-47.21	0.0000
TR HP *Spu	-						0.0063794				0.003172	2.01	0.0445
TR GB Redu							-0.016707	1	_		0.003171	-5.27	<.0001
Spur Combin			Comb	iner f	Redux		0.6886029				0.004435	155.27	0.0000
MR HP*Idler							-0.049961	1	L		0.003191	-15.66	<.0001
MR rpm*idler		-					0.154156		J		0.003244	47.52	0.0000
Engrpm *idi							-0.228848				0.003191	-71.71	0.0000
TR HP *Idler							0.0069166				0.003172	2.81	0.0050
TR GB Redu							0.0066872		-		0.003172	2.11	0.0351
Spur Combin			Redux				0.2692057		L	i	0.003244	83.00	0.0000
Idler Redux*k	dier Re	idux					0.4121047	<u> </u>			0.004432	92.99	0.0000
Prediction i	Profil	er 👘											
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