# Gearing for Gearheads

## Part 1

by Phillip A. Miller

### Foreword

This series of articles is an engineering-driven examination and appraisal of the Rolls-Royce Merlin propeller speed reduction units (PSRUs). In order to recover dimensions and details of the original Merlin gearsets by “reverse engineering”, it was necessary to derive involute gear relations from basic principles.

**Part 1** looks at production R-R Merlin PSRUs and extends this historical examination to include Unlimited Racer efforts and hypothetical possibilities of modified ratios. Particular threats facing these PSRUs in racing applications are also explored.

**Part 2** will include a description of the principles of involute spur gearing and derivation of relations for the determination of critical dimensions and performance factors. Also included is a look at prototypical gear failure, its root causes and possible alleviation techniques.

**Part 3** will include gearset dimensional specifics, calculated comparative stresses and performance factors for several ratios with discussion and summary.

### Instigation

Graham White, our *Torque Meter* editor, has imagination. He thinks that AEHS gearheads anxiously await articles on Ram Air Induction, Engine/Propeller gearing and who knows what else? I surely hope that he is correct as I seem to end up attempting to write these articles.

However, he IS a kind and helpful editor who promptly put me in contact with Dave Birch of the R-R Heritage Trust and folks in the Unlimited Race community for pertinent tech information and history.

Dave Birch is also very polite, kind, helpful and cooperative, but early communication hinted at a totally unexpected problem. I had reviewed applicable *Torque Meter* articles and illustrations and tentatively jumped to conclusions about probable mechanical problems of this small pinion/large driven gear PSRU. I had by then also hastily preconceived some notions about probable diametral pitches and shifted addendum/dedendum tricks to strengthen pinions.

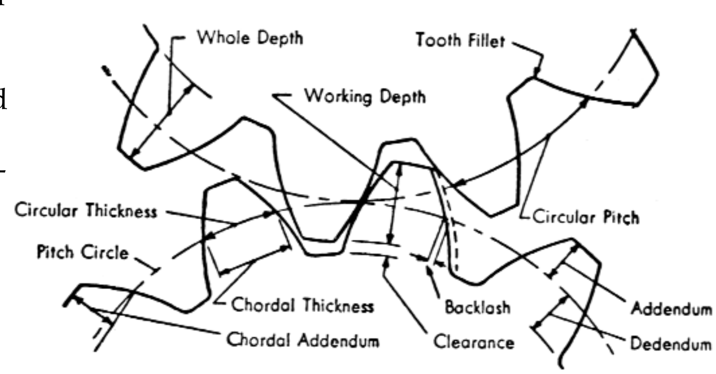
My initial questions were answered with pitch diameters and diametral pitches that were not much different from the preconceived notions but didn’t involve whole numbers or even simple fractions and hence didn’t appear to have been chosen with deference to the economics of allegedly standard shop tooling let alone the suspiciously neat and clean textbook examples that had ALWAYS raised suspicion. Dedendums, however, were the real clincher. It was revealed that in present day Britain it was believed that the term referred only to ancient Roman Forts.

Further denial was useless. I was dealing with an Alien Culture and even years of youth and middle age misspent wrestling with Lucas Electrics, Austin/MG gearboxes, Burman gearboxes, Triumph gearboxes, Triumph and other oil leaks was not likely to help.

“Not to worry” says Dave, “The drawings will explain everything!” He sent copies of the remaining (it’s been well over half a century and there was certainly destruction in that interval) available drawings, the gear and pinion drawings for the Merlins manufactured at R-R, a gear drawing (but not the pinion) for the favored ratio gearset later used by Packard (but engineered at R-R) and assembly drawings of each of the two stock PSRUs based on these gearsets.

### References and Textbooks

An early Post War copy of Spott’s *Elements of Machine Design* (Second Edition) and 1927 and 1940 editions of *Machinery’s Handbook* were readily at hand along with a Second Edition 1974




*Definitions of Gear Tooth Parts (Courtesy of Machinery's Handbook)* copy of Shigley's *Mechanical Engineering Design* and a 1980s Eighth Edition of *Mark's Standard Handbook for Mechanical Engineers*. *Machinery's Handbook* was published in Britain as well as the US and the 1940 issue contained British as well as Modular (inch and metric) gear system standards along with "American Standard Spur gear Tooth Forms". Also discovered from this volume was the fact that as recently as sixty some years ago the term "dedendum" was still in use in Britain to denote the radial distance between pitch and root diameters (the latter a term more popular in Britain than the US). No significance to ancient Roman Forts was mentioned.

Finally, a replacement (cheap, used, Dover reprint, 1949) copy of my long lost and missing *Analytical Mechanics of Gears* by Earle Buckingham was ordered on the Internet. This is not another "cookbook". Its utility is expected in the area of support and verification of the GENERAL derivations deemed likely necessary for this project.

For example, the references and textbooks at hand contained the involute curve formula only in rectangular coordinates. This is about as handy for dealing with whole gears as a child's (OK, a grandchild's) Etch-a-Sketch is for attempting to draw circles. Efforts to transform Cartesian coordinate parametric relations to polar form resulted in more complication and little utility so direct derivation of the involute curve in polar coordinates was attempted. This was less difficult than feared and resulted in a useful pair of equations. These allow ready calculation of angular values and hence the angular contributions of various tooth features in terms of their radial location on the involute curve. They were of a different and more useful (to us) form but checked out mathematically against the presentation later found in Earle Buckingham's book. It was reassuring to note that a serious (and great!) gearhead had trod down this path before.

## R-R Drawings—The Project Begins

The drawing copies arrived shortly, representing two of several (0.420:1, 0.440:1, 0.477:1, etc.) gear ratios used by various Merlin Marks. The 0.477:1 ratio (65-tooth) gear set of R-R built Merlins was represented by gear Dwg. D.20684/1 with an illegible drawing date but an approval date of April '42. Pinion Dwg. D.20685/1 was labeled as Drawn 18-6-42 (June 18th 1942) and Approved Sept:32(!). This spot of apparent British humor confirmed alien culture worries but served to lighten the atmosphere. The Reduction gear Ass'y (0.477:1 Ratio). Dwg. D.23324 was dated as Drawn 5-7-43 (July 5th, 1942) and Approved April '43. It was stamped that it would not be kept up to date as of 29-5-44 (May 5th. 1944). Tooth profile was a 20° pressure angle involute. These straight cut spur gears had a 2.650" face width.

The 0.420:1 ratio (71-tooth) gearset of the Packard built Merlins was represented only by gear Dwg. D.24537, which was labeled as Drawn 2-11-43 and Approved May '44. The Reduction gear Ass'y (0.420:1 ratio) D.25398 drawing dates were illegible except for the stamped notice that it would not be kept up to date as of 28-7-44. Tooth profile of the 2.625" face width straight cut spur gear was a 25° pressure angle involute.

## Reno and Other Input

Unlimited Racers were reported by stateside contacts to have successfully explored a 0.383:1 ratio 65-tooth 20° gearset but to have suffered pinion tooth breakage with a 0.340:1 ratio based on a 71-tooth 25° set. These modified ratio sets are presumed to have been subjected to the output of Unlimited Race engines. Further prying in this direction was not successful and was dropped out of respect for other folks' "Speed Secrets".

I did gather from old friends (OK, from YOUNGER old friends) some first hand narrative on a variety of PSRU experiences with other powerplants that were highly credible and logical. The implications, where applicable, will be considered later.

## R-R Drawings and Modified Ratio Gearsets - The First Serious Look

### 65-Tooth Production Gearsets

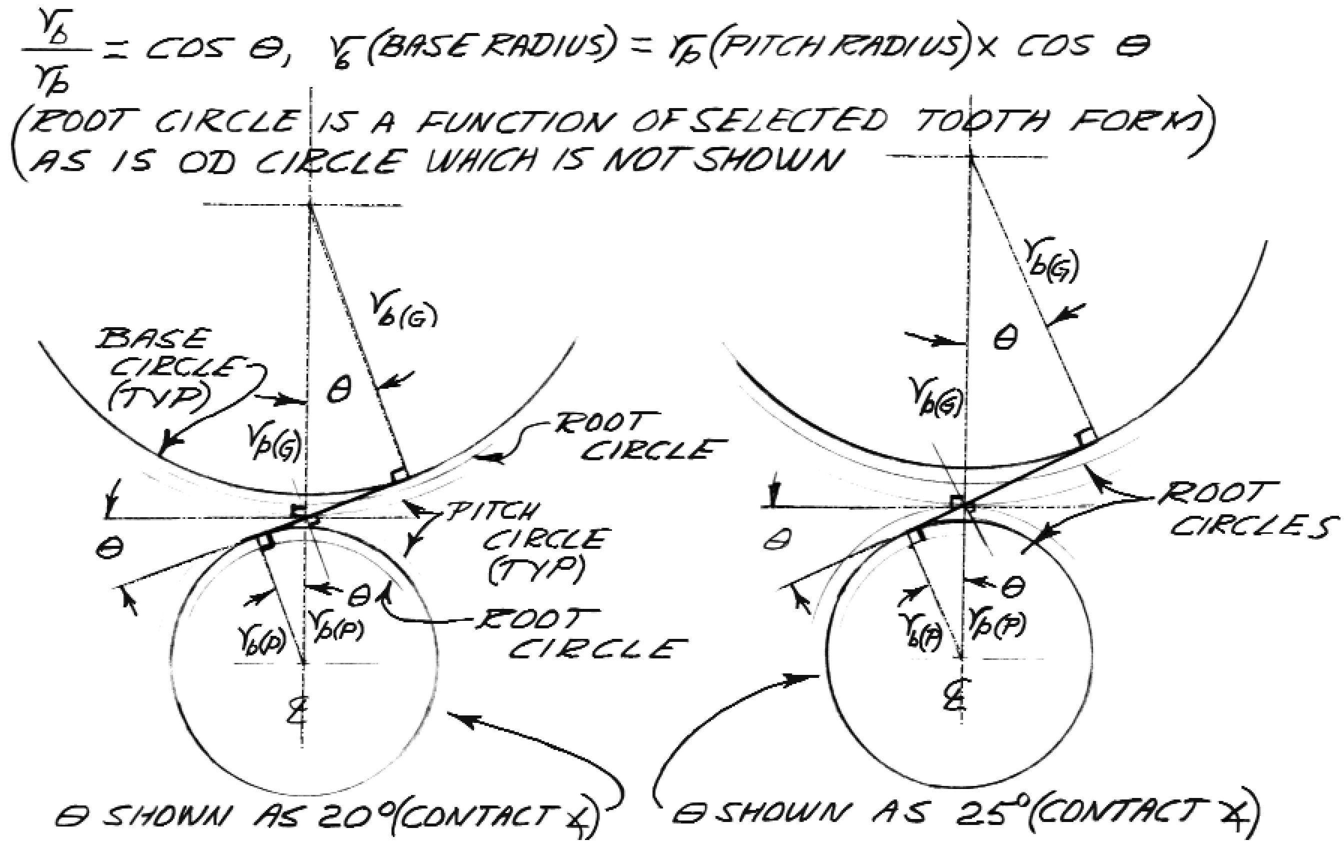
The 0.477:1 Ratio (21 tooth pinion/44 tooth reduction gear) R-R drawings were not initially intimidating. Despite some conveniently overlooked alien culture, a momentarily frightening sixty-one year old drawing error and Kings English to Gringo translation (reference to an "angle of obliquity") it was obvious as noted above that 20° pressure angle involute spur gears were involved. A diametral pitch of 3.7616 teeth per inch and pitch diameters of 5.5872" and 11.6972" (in lieu of the rounded and even numbers of textbooks and frugal shop practice) was also noted.

Detail tooth proportions matched the British

Standard for precision cut or ground gears, "Class B/C", as presented in the 1940 *Machinery's Handbook*. The gear teeth were dimensioned as 0.037-0.038" thin on the pitch circle (tooth widths and intertooth spaces are nominally equal on the pitch circle) while the pinion teeth were dimensioned as 0.012-0.013" thick on their pitch circle, resulting in a nominal 0.025" running clearance. The addendum (pitch circle to OD radial dimension) of both gear and pinion was 0.265". The dedendum dimension of both (as implied by root circle to pitch circle radial distances) were also identical at 0.333" and hence this is not a "shifted addendum/dedendum gearset but rather an elegant one with the desired gear and pinion tooth widths resulting from specially produced or modified generating type tooling and subsequent form grinding.

Tooth filet radii on the pinion are critical to fatigue life because of the pinion's relatively high loading. Hence it was examined closely while the larger less critical gear was not. The specified 0.125", ground, tooth flank filet radii are very nearly "wall to wall" with the angular contribution of the teeth on the root circle though not specified or dimensioned as such. The British Standard for precision cut or ground gears does allow a trace of "flat" between adjacent tooth filet radii. The result is near that of a single filet radius connecting adjacent teeth with the small flat providing dimensional relief for the actual machining. This "largest possible" filet radius practice reportedly started as a successful effort to increase the fatigue life of steel rolling mill gearing (for perhaps an earlier War). By the end of WWII it was accepted as near standard for aircraft engine gearing. It should be noted that single tooth static strength, in contrast to the fatigue strength, may be decreased by the longer effective tooth beam length resulting from a large filet radius.

The root diameter of this pinion is 4.9167" and the base diameter is 5.25025", leaving a bit of the tooth flank undefined except through scaling the 6:1 views on D.6085/1. "Wall to wall" contribution of 0.125" filet radii with 0.030" intervening flats leave room for teeth of 0.5097" arc width at filet radii intersections. Tooth angular width tracked from pitch circle to base circle to root diameter suggests a width of 0.48173" (chordal) and agrees closely with the scaled 6X drawing. A difference of approximately 0.028" is both physically possible and equally dependent upon tooling choices. However, it is likely that the layout




*Basic Geometry of Meshing Involute Spur Gears* specialist, 60-some years ago, was closer to knowledge of R-R tooling than we are today and this suggests that we trust the 6X scale drawing and accept the possibility of near 0.060″ inter-radii flats.

We see from this that inter-tooth spacing sets dimensional limits on the choice of radii. In example, British Standard for precision cut and ground gears, "Class A", specifies a larger dedendum (smaller root diameter) with less peripheral room and smaller resulting permissible filet radii. Thus poorer fatigue performance could be expected in a relatively small pinion. It is suggested that this explains the choice of "Class B/C" proportions.

Actual practice is not always so straightforward. Deeper teeth are more flexible and compliant, thus better sharing loads with adjoining teeth. Chain sprocket teeth provide an old though extreme example. Years ago, radial inward slots between sprocket teeth ending in "crack stopper" circular holes permitted Dirt Bike competition with worn out chain and sprockets (sometimes brittle weld repaired) SEASONS beyond their normal life. I found the resulting patent (US 3,173,301) to be a good resume stuffer but otherwise not saleable. Gearing application of increased compliance is similar but in an inverse (perverse?) manner. Increased compliance in the gear will benefit the pinion as well and is more easily accomplished.

### 71-tooth Production Gearsets

The 0.420:1 ratio R-R gear drawing alone represented 71-tooth gearsets. It shows a straight cut spur gear of 2.625″ face width with an OD of 12.5399″ and a pitch diameter (for the D.25398 PSRU) of 12.1693″. This latter with its 50 teeth (of the 71-tooth set) implies a diametral pitch of 4.1087 teeth per inch though such is not called out. This gear is apparently also used against an idler gear for the reverse rotation Merlins with decreased proportional center-to-center distance, a smaller effective pitch diameter, larger diametral pitch and decreased effective pressure angle. It is labeled as a "Dual Pitch Gear" and that is strictly true in view of multiple assignments, but the term is a bit mind boggling to a Gringo concerned with only the D.25398 PSRU application.

The 25° nominal pressure angle of this gear (and set) is acknowledged as advantageous in textbooks and handbooks of the era, but is not presented in a Standard with agreed upon tooth proportions and tables of "Lewis" or "Form Factors", etc. These things are found in later (post WWII) texts and handbooks, but the later data does not match the proportions or discernable antecedent proportions of the 50-tooth gear on R-R Dwg. D.24537. The short addendum (0.1853″) and relatively long dedendum (0.3159″) suggest a shifted addendum/dedendum geometry (or an alien equivalent thereof) to provide an increased diameter, thicker-toothed and stronger pinion, although no standard starting point was found. Gear teeth are called out as '0.03625″ thin' which adds to the notion of shifted addendum/dedendum manipulation. A minimum running clearance of 0.025″ is called out on the D.25398 0.420:1 ratio reduction gear drawing, which would be provided with 0.00525-0.01125″ thicker than nominal pinion teeth.

Fortunately, the lack of a standard starting point does not prevent a reasonable definition of the mating pinion. A pitch diameter of 5.1111″ matches the required ratio and we know that the dedendum of the pinion must equal the addendum of the gear plus a reasonable clearance. A clearance value of 0.061″ is assumed here as it is proportionally equivalent (0.25/diametral pitch) to that of the earlier and successful 0.477:1 ratio gearset. This results in a pinion dedendum of 0.2463″ while a similar but inverse manipulation results in a pinion addendum of 0.2549″, a pinion OD of 5.6209″ and a root diameter of 4.6185″.

Tooth filet radius of the 50-tooth gear is called out as 0.090″ The angular contributions of the features deduced for a matching 21-tooth pinion were evaluated with our polar coordinate involute equations. A summing up and some Pythagorean theory activity at filet radius/involute tangencies revealed just sufficient inter-tooth space for 0.100″ filet radii and small flats if pinion tooth thickness is held to near the apparent minimum increase over nominal at the pitch line of 0.00525″. This follows the general practice seen in the 0.477:1 ratio gearset where the pinion filet radius was larger than that of the gear (0.100-0.125″, gear to pinion).

### 71-tooth, 25°, 0.340:1 Ratio Modified Gearset

Pinion and gear dimensions for the failed 0.340:1 ratio 71-tooth gearset from a custom built racing application were derived by assuming the 25° pressure angle and tooth forms common to the 71-tooth 0.420:1 ratio production gearsets. The results can be regarded as reasonably probable estimates but it is cautioned that the basic assumptions are not verified.

The 0.340:1 ratio is satisfied by an 18/53-tooth count. The assumed diametral pitch of 4.1087 teeth per inch yields gear and pinion pitch diameters of 12.8995″ and 4.3809″, and multiplying these by the cosine of 25° yields base circle diameters of 11.5649″ and 4.0960″ (again respectively). Root circle diameters and ODs were based on the calculated pitch diameters with the addendums and dedendums apparent from the 0.420:1 ratio example. As with that production example, it appears that tooth filet radii of 0.10″ and minimal flats are just possible with pinion teeth held to near the nominal plus 0.00525″ minimum increased thickness on the pitch line. Increased gear sizes and decreased pinion diameters lead to interference worries. These were explored with large scale layouts and templates in the Merlin's youth. I have indulged in such practice myself but now with old eyes and stiff pencil and eraser fingers, I prefer to work the problem with analytical geometry and an HP-65 or later HP-48SX.

The interference/undercutting geometry is rearranged with gear OD radius replacing the rack or straight cylindrical gear hobb's straight lines and modified to our new purpose by locating a new "interference point" outboard of the pinion base circle radius at the outboard radial extent of the pinion filet radii tangency. Then, with the aid (again) of Pythagoras, a maximum/limiting gear OD is calculated and compared to the actual gear OD. Rolling interference, on the basis of this exercise, does not appear to be a potential problem for either the 25° 71-tooth 0.420:1 production or 0.340:1 modified ratios.

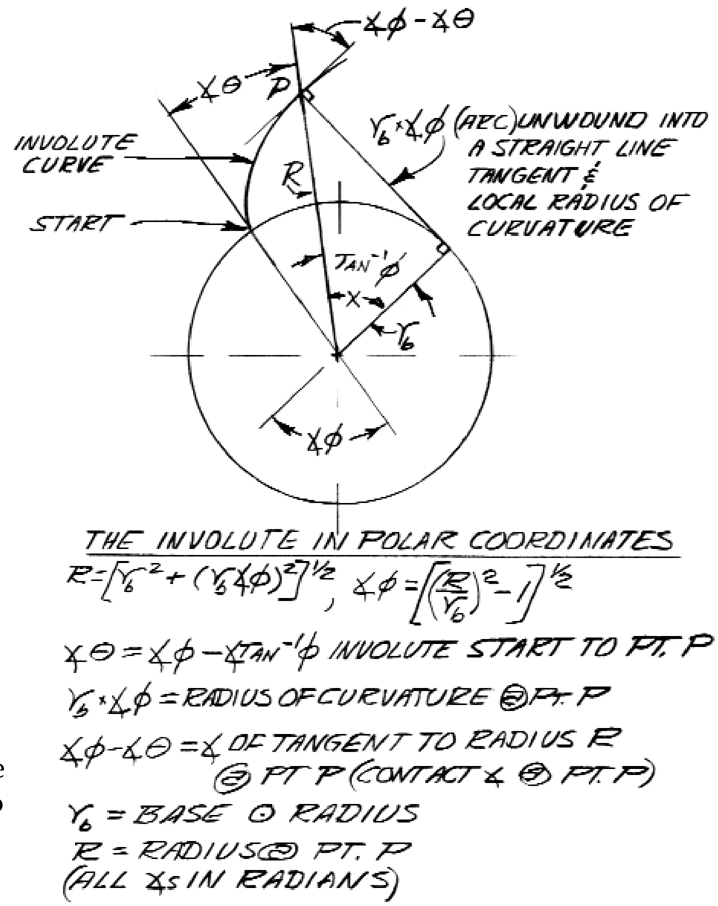
### 65-Tooth, 20°, Modified 0.383:1 Ratio Gearset

Probable pinion and gear dimensions for the successful 0.383:1 ratio 65-tooth gearset were derived by assuming the 20° pressure angle, British Standard B/C tooth form and pinion tooth increment at pitch line (.012-.013″) of the 65-tooth 0.477:1 ratio production gearsets. The availability of 65-tooth gear and pinion drawings was an aid but did not alter the fact that the major assumptions regarding this modified ratio Unlimited Racer gearset are not verified.

The 0.383:1 ratio and 65-tooth total defines an 18-tooth pinion and 47-tooth gear for this gearset. These and the 3.7616 tooth/inch diametral pitch (justified by retention of the 65-tooth total count) give us a 4.7852″ pitch diameter pinion and a 12.4947″ pitch diameter gear. Base circle diameters, determined by multiplying pitch diameters by the cosine of the 20° pressure angle are 4.4966″ and 11.7412″, pinion and gear respectively. British Standard Class B/C addendum (1/diametral pitch) and dedendum (1.25/diametral pitch), as used on the 65-tooth production gearsets, yield root diameters of 4.1212″ and 11.8307″, again pinion and gear respectively. ODs are, similarly, 5.3172″ and 13.0247″. Pinion intertooth spacing, as with the 65-tooth production gearset, is sufficient for a 0.125″ filet radius and a joining flat of generous tolerance. Rolling interference was investigated as with the preceding 71-tooth production and modified ratio gearsets and with similar results. It does not appear as a problem here.

### Potential 65-Tooth 20° 0.354:1 Ratio Gearset

It appears, from the preceding analytics and the reputed modified ratio results, that the coarser 65-tooth 20° (3.7616 diametral pitch) option may




*Derivation of the Involute in Polar Coordinates* have advantages in this application. Single tooth strength, the rough indicator of torque/power capability and tooth filet radii are somewhat better. These advantages, though, are countered by a lower to be expected contact ratio.

A 65-tooth, 0.354:1 ratio (17/48) 20° pressure angle gearset with pinion teeth 0.013″ thicker than nominal on the pitch line was explored to provide specifics for later evaluation and comparison. The procedure followed that of previous examples. Pitch diameters were 12.7605″ (gear) and 4.5194″ (pinion). Similarly base circle radii were 5.9955″ (gear) and 2.1234″ (pinion). Root radii were 6.0483″ (gear) and 1.9277″ (pinion) while OD radii were 6.6463″ (gear) and 2.5257″ (pinion). Pinion tooth filet radii of 0.150″ with approximately 0.010″ joining flats are possible with care and rolling interference does not appear to be a problem. Tooth flanks between the base circle and the root circle were assumed to be radial straight lines. The ratio (0.354:1) is close to that (0.340:1) of the failed modified 71-tooth gearset and it will be interesting to see the comparisons of complete Merlin contemporary evaluations in later parts of this series.

### Potential 65-Tooth, 25°, 0.354:1 Ratio (17/48) Gearset

Curiosity about hypothetical possibilities led to thought of a coarser (65-tooth) 25° pressure angle gearset of 0.354:1 ratio. Exploration of Shigley's *Mechanical Engineering Design* disclosed an apparent post WWII American Gear Manufacturers' Association (AGMA) tooth form which was basically identical to that of the British Standard Class B/C tooth form of 1940 EXCEPT that it included a 25° pressure angle, though without the Lewis or Form Factor (y) information usually found in more recent tables. The temptation to explore a 25° pressure angle, 65-tooth (3.7616 teeth per inch diametral pitch) gearset with pinion teeth 0.012-0.013″ thicker than nominal on the pitch line was thus reinforced. Unable to resist temptation, the pitch diameter of the gear was soon reckoned (again) as 12.7605″ and that of the pinion as 4.5194″ as with the 20° pressure angle 65-tooth gearset. The pinion base circle radius differed, following as cosine 25° x 4.5194/2 = 2.0480″ and the gear base circle radius, similarly, as cosine 25° x 12.7605/2 = 5.7825″. Addendums (both) were 1/diametral pitch or 0.266″ and dedendums (both) 1.25/diametral pitch or 0.332″ and these led to a gear OD of 13.2925″ and a pinion OD of 5.0514″. Root diameters were 12.0965″ (gear) and 3.8554″ (pinion).

Total (to base circle) pinion tooth and inter-tooth space angular widths were evaluated and tooth filet radii upper limits between the involute flanks investigated. The "pointy toe cowboy boot" profile of the 25° pressure angle teeth limits filet radii to approximately 0.120″ with no tolerance for joining flats. It is probable that the practical maximum would be 7/64 or 0.1094″. Alternately, reduction of pinion tooth width to nominal thickness on the pitch line would JUST render 0.125″ filet radii possible. Evaluation of Lewis or Form Factors for these two options in a later installment of this article will include filet radii stress concentration influences and should lead to interesting comparisons. There is an obvious trade-off between tooth width and filet radii (in the accepted sense) embodied in the pressure angle choice. Is an equally obvious compromise possible?

### Further Potential Threats to Merlin PSRU Gearing

The Merlin is an even firing V–12 with six paired rod crank throws. Each revolution sees six compression strokes, six power strokes, twelve coupled inertia peaks and twelve coupled inertia minimums. That's a lot of lumps plus sums and differences, crank torsional twitches and lesser afflictions like intake and exhaust strokes.

A torsionally flexing hollow quill coupling smoothes power delivery from crank to the pinion gear far end. This works well with both stock and race engines. BUT Merlin gearing is fairly wide so that altered ratios with smaller pinions raise the specter of tangible torque twist of the pinion gear proper with uneven loading across the gear face width as a result.

Buckingham regards the gear rim as the only significant source of torsional resistance and neglects the tooth contribution. This appears reasonable in normal industrial practice but not as applicable with the lightweight thin rim and fairly wide faces of Merlin PSRU drive pinions.

The tooth contribution here approaches that of the rim but both diminish rapidly as the pinion diameter and number of teeth is reduced in the quest for more favorable ratios. There is some room in the Merlin PSRU assembly for thicker pinion rims although practical limits in this direction may involve not only clearances but section depth/heat treat considerations.

### Contemporary PSRU Gear Failure Experience (OK, Near Contemporary)

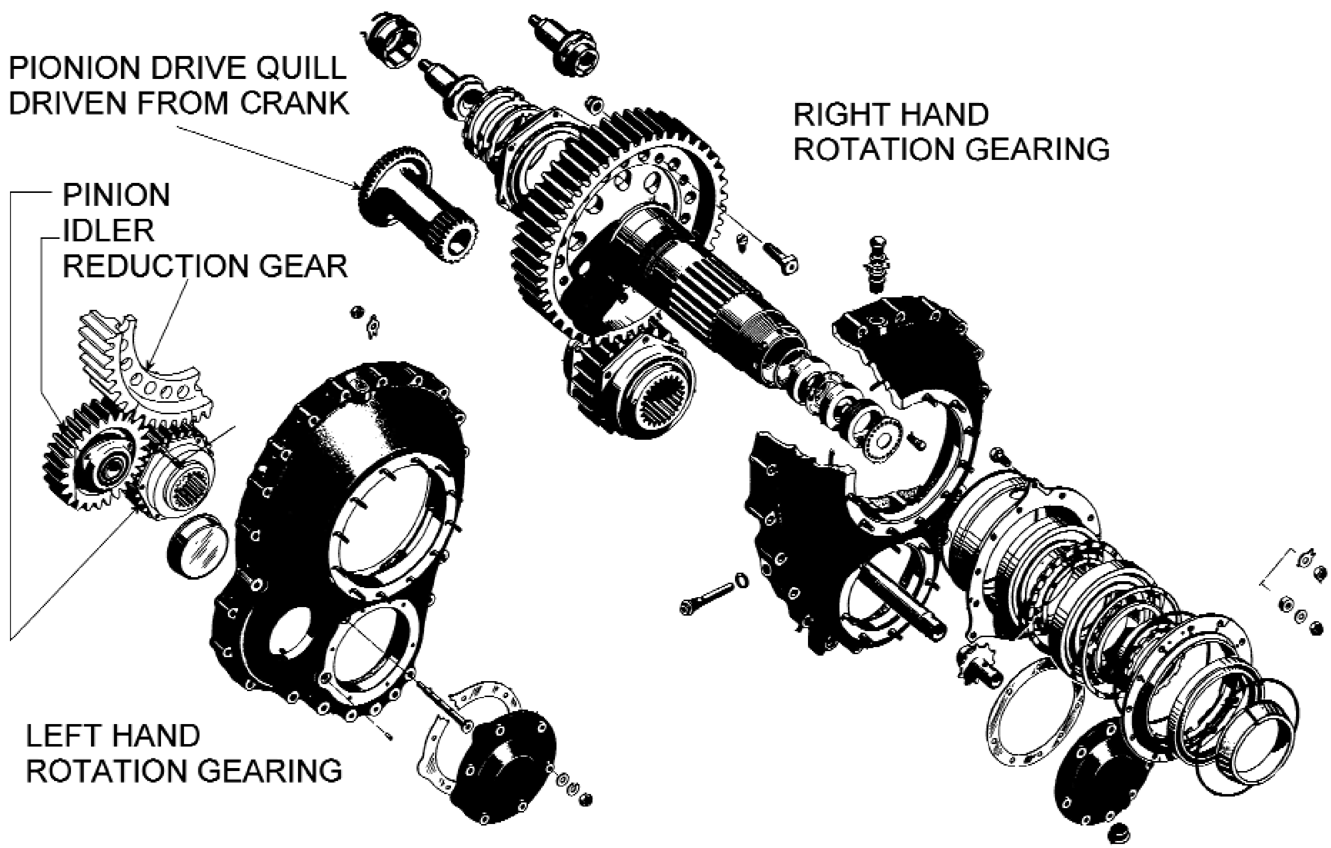
A few years ago, Alaskan bush operators of Helio Couriers experienced a rash of premature PSRU gear failures with the normally very satisfactory Lycoming geared engines. A local and sharp mechanic/engineer tracked the problem to an operational condition common to the bush operation. Namely low intermediate power setting, as on a long descent, scud running, or reduced speed search efforts where the engine/prop combo oscillated between thrust and windmilling, thrust and drag, This was the aero equivalent of “chain snatch”, a descriptive term widely appearing in motorcycle publications not so long ago. The satisfactory field fix consisted of restricting operation to conditions of either positive thrust or negative drag. These conditions were reportedly defined with the aid of instrumentation at the thrust bearing.

The idling and shutdown gear clatter of a Merlin is less likely to be troublesome. Ferrying operations, however, particularly with modified engines and reduction ratios and a gentle throttle hand could unintentionally enter the “chain snatch” zone. This is best avoided as it impact loads gear teeth alternately in each direction.

Impact loading due to unloaded acceleration/deceleration across the tooth running clearance readily approaches levels which may result in Hertzian (the stress/strain distribution resulting from the forced contact of two curved rigid bodies) surface damage and ultimately tooth failure with otherwise satisfactory gearing as in the case of the Alaskan Helio Couriers.

The modified helicopter turbine used in the Piper Enforcer (Turbine Mustang clone) had a surprising and severe gear life limitation at full power. It is surmised, without access to the engineering details, that at least one reduction stage was eliminated in the conversion from helicopter to fixed wing powerplant and a ratio change was incorporated in a remaining planetary stage.

This apparently left the individual planet gears with inadequate contact ratio against the center sun gear. The sun gear typically has multiple planet gears in mesh so may not itself be in jeopardy. Planet gears in mesh with a large internal




*This Packard illustration shows the conventional right had rotation Merlin reduction gear (upper right). The lower left part of the illustration shows the similar route Packard took for opposite rotation in the F-82.*

tooth gear tend to have adequate if not excessive contact ratios. But individual planet gear mesh with the sun gear is the most vulnerable site and this may not improve even with increased planet gear diameter if it is in mesh with a decreased diameter sun gear.

Thus it may escape suspicion to deliver an ugly surprise late in testing or even in service. Kevin Cameron recently wrote about a similar problem in the development of the Wright R-3350. The Enforcer, however, was more than adequately powered so that a limit on full power use was an acceptable though probably not a permanent solution to the gearing problem. That would not be the case with an Unlimited Racer.

This narrative is included as a reminder to watch the contact ratio when juggling reduction ratios with smaller pinions.

### An Additional Hazard

The gyroscopic precession loads generated by large propellers are truly awe inspiring. They can load the propeller shaft roller bearings oppositely in any and all radial directions. These loads are transient but may momentarily distort the PSRU housing sufficiently to misalign the reduction gear mesh.

It would be interesting and perhaps helpful, in the presumed absence of applicable and available R-R factory data, to fill a fixture mounted PSRU housing assembly with dummy shafts and bearings and add a big lever arm to the dummy prop shaft to crank in calculated maximum precession loads from a hydraulic cylinder for static test. Stresscoat (www.stresscoat.com) brittle lacquer could provide a first approximate reading of the results and from this it is possible that external brace development along the lines of the R-R approved "tie bars" could improve any deficiencies discovered in regard to Unlimited Racer use.

### And Last but not Least

Lubrication of the Merlin PSRU gearsets has not appeared to be a problem in either original purpose, commercial or competition use. It was designed with an engine lube oil jet directed into the closing mesh in accordance with the prevailing wisdom of the era. Subsequent experience with REALLY high speed gearing as in gas turbine and rocket turbopump applications has shown that meshing side lube oil jets can fail to reach their objective against prevailing gas and vapor flow ejected from a closing mesh. It becomes imperative at some pitch line velocity to direct the lube oil jet into or laterally through the OPENING mesh where the reduced pressure literally explodes the jet into a spray that coats and cools hot gear faces as the mesh opens.

It may be possible to improve the Merlin PSRU lube scheme. Is PSRU housing drainage or scavenging adequate at higher than stock RPM? It is generally not advantageous to run high speed gears and bearings in a flooded condition. Would an "opening mesh" side lube jet (possibly in addition to the original) and improved scavenging reduce oil heating and improve gear face cooling? Answers to these questions may be helpful in Unlimited Racer applications and could be investigated and to a certain extent tested without great expense. Perhaps this has already been done? The Reno Racers are an ingenious bunch and I write of their R-R Merlin race engine and PSRU development in virtually unlimited ignorance. BUT I don't wish to ignore any advantageous possibilities either.

### That's All for Now

We have defined and developed herein ("Gearing for Gearheads, Part 1") the Merlin production gearsets and a few more of advantageous ratios in addition to noting several PSRU gear problem areas. Analytical tools have been derived, written for presentation in "Gearing for Gearheads, Part 2" and blessed or cursed as the case may be with illustrations which are still being polished for presentation.

It is intended that "Gearing for Gearheads, Part 3" present the results of evaluating the gearsets of Part 1 with the tools of Part 2 to provide a well rounded look at involute spur gearing in general and application to the R-R Merlin in particular. This work is presented on a "best effort" basis and it should be remembered that it has not been extensively nor independently checked. It is essentially an individual and not an engineering office effort. I check my work by repeating it a few times and using my analytical tools as written for presentation. BUT I know that this does not eliminate the pitfalls inherent in checking one's own work so please keep your eyes open and holler when you spot mistakes.

***TM***