

Assessment of Planetary Gear Train with Multiple-Pinion Arrangements

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ABSTRACT

The planetary gear train has been used as a core element for most of automatic transmissions (AT) due to its unique advantages: such as high power density, lower noise level, lower support bearing cost, and easier for mass production, compare to lay shaft or other types. In particularly, when multiple-pinion (MP) arrangement is used to accommodate flexible packaging design specifications. However, the planetary gear train does inherit a critical limit that is the lack of flexibility on gear ratio selection, because it needs to comply with certain constraints on tooth combination to meet gear train installation for assembly, load sharing for durability and phasing arrangement for noise concerns.

To fully take the MP advantages for AT transmission applications, while increase the gear ratio selection was the primary motivation that prompted us to seek for some potential solutions on the MP arrangement. This report will provide the method for the proposed approach, and also reveal some of the associated assessment results.

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INTRODUCTION

Recently, pertain to environmental pressure on low emission, and market demands on improving fuel economy and smooth driving, many AT manufacturing companies have considered to apply higher transmission gear ratio ranges from traditional four speeds up to current six-speeds and then further extend to eight or higher speeds to enhance better performance as well as higher efficiency. At the meantime trying to use the same gear train components on many different platforms to achieve both package commonality as well as cost-saving. The approach by using the multiple-pinion (MP) option is the most outstanding one among other alternatives. The MP approach uses common gear train components but various numbers of planetary pinion for different platforms, such as three pinions for compact, four or five pinions for regular, and five, six pinions or more, if space allowed, for heavy duty vehicles, thus enables transmission designer to use many common parts in the planetary assembly. So the costs in design, manufacture and logistics management can be maintained at the lowest level.

As mentioned earlier, planetary gear train design needs to comply with certain rules in order to meet specifications on gear train installation, durability and NVH concerns, these rules have imposed tight limitation on overall ratio selections. To ease these constraints, especially for MP applications, the author has applied the concept of "least-mesh angle (LMA)" which was introduced by O.K. Kelley¹ back to Sixties. The approach of using LMA to include the non-equally spaced pinion arrangement alternatives instead of just using the traditional equally spaced approach could greatly increase the ratio selection options.

To evaluate the advantage as well as the potential risks of using non-equally spaced alternative, some assessments have been conducted and will also be reported in this paper.

METHODOLOGY

The scope of this study covers two separated areas: One is the analytical calculation. that is to find out all the feasible carrier arrangements for establishing different ratios in the working range; while the other is the assessment part that is to evaluate potential benefits as well as risks of the proposed approach. Based on these perspectives, the process of the report can thus be divided into two major sections accordingly.

a. Analytical Calculation

History Background

First of all, let's review some of the conventional three and four speed planetary carrier arrangements as shown on [Table 1](#), and then followed up to the current six to eight speed ones as shown on [Table 2](#). The differences on the development trend are listed as follows:

	Conventional	6/8 speed (Current)
Gear set Type	Simple/double	Ravignaux/Simple
No. of Pinions	3 / 4	3 to 6
Phasing	Counter / Seq. & In-phase	Seq. / Counter & in-phase
Eq. Spaced	All	All



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Table 1. Conventional Planetary Gear Train Arrangement

Transmission Model	Gear I.D.	Teeth #	R/S Ratio	Gear set	# of Pinion	Eq. Spaced	Phase	Comment	Transmission Model	Gear I.D.	Teeth #		Gear set	# of Pinion	Eq. Spaced	Phase	Comment
NISSAN/JATCO RL-4F02A	SUN	41	1.829	INPUT	4	Yes	SEQRNTIAL	Non Hunting teeth comb	GM 440-T4	SUN	26	2.385	INPUT	4	YES	COUNTER	Hunting teeth comb
	PINION	17								PINION	18						
	INTERNAL	75								INTERNAL	62						
	SUN	33	2.273	REV	3	YES	IN-PHASE	Hunting teeth comb		SUN	42	1.762	REV	4	YES	COUNTER	Hunting teeth comb
	PINION	21								PINION	16						
	INTERNAL	75								INTERNAL	74						
	TRANSFER	50	PARALLET SHAFT					Non Hunting		SUN	30	2.333	F.D.	4	YES	COUNTER	Hunting teeth comb
	TRANSFER	51								PINION	20						
	FINAL DRIVE	22	PARALLET SHAFT					Hunting teeth comb		INTERNAL	70						
FINAL DRIVE	74																
TOYOTA/AISAN A140E	SUN	39	1.821	1ST	4	YES	SEQRNTIAL	Non Hunting teeth comb	FORD AXOD	SUN	38	2.263	INPUT	4	YES	COUNTER	Hunting teeth comb
	PINION	16								PINION	24						
	INTERNAL	71								INTERNAL	86						
	SUN	27	2.333	REV	4	YES	SEQRNTIAL	Hunting teeth comb		SUN	38	1.842	REV	4	YES	COUNTER	Hunting teeth comb
	PINION	18								PINION	16						
	INTERNAL	63								INTERNAL	70						
	SUN	27	2.407	1ST	4	YES	SEQRNTIAL	Non Hunting teeth comb		SUN	30	2.467	F.D.	4	YES	COUNTER	Hunting teeth comb
	PINION	19								PINION	22						
	INTERNAL	65								INTERNAL	74						
	TRANSFER	55	PARALLET SHAFT					Non Hunting		SUN 1	34	2.8824	RAVEN IAL GEARS ET	3	YES	SEQRNTIAL	Hunting teeth comb
	TRANSFER	52								SHORT PINION	31						
	FINAL DRIVE	19	PARALLET SHAFT					Non Hunting		INTERNAL	98						
	FINAL DRIVE	71								SUN 2	38						
MAZDA EC-AT	SUN 1	30	2.80	RAVEN IAL GEARS ET	3	YES	SEQRNTIAL	Hunting teeth comb	ZF 4HP18	LONG PINION	22		PARALLET SHAFT				Non Hunting
	SHORT PINION	22								TRANSFER	59						
	INTERNAL	84								TRANSFER	53						
	SUN 2	36	2.333							FINAL DRIVE	18		PARALLET SHAFT				Non Hunting
	LONG PINION	24								FINAL DRIVE	77						
	TRANSFER	20								Max/Min (Ring Gear / Sun (R/S) Ratio Range:							
	TRANSFER	39	PARALLET SHAFT					Hunting teeth comb		R/S =	2.882	MAX	To	1.76	Min.		
	FINAL DRIVE	74															

Table 2. Current Planetary Gear Train Arrangements

Trans. Model	Gear I.D.	Teeth #	Ratio	Gear Set	# of Pinion	E _q . Spaced	Phasing	Comment	Trans. Model	Gear I.D.	Teeth #	Ratio	Gear Set	# of Pinion	E _q . Spaced	Phasing	Comment
GM 6T-65/70 E	Sun	41	2.171	Input	5	Yes	Seq.	Non-Hunting	ZF 8 HP70	Sun	36	2.0	1st	3/4	YES	In-Phase	Non-Hunting
	Pinion	24								Pinion	17						
	Internal	89								Internal	72						
	Sun	34	2.882	Reaction	3/4	Yes	Seq.	Non-Hunting		Sun	36	2.0	2nd	3/4	YES	In-Phase	Non-Hunting
	Pinion	32								Pinion	17						
	Internal	98								Internal	72						
	Sun	41	2.171	Output	4/5	Yes	Seq.	Non-Hunting		Sun	69	3.1	3rd	3/4	YES	In-Phase(3) C'phase (4)	Non-Hunting
	Pinion	29								Pinion	20						
	Internal	89								Internal	111						
Pinion Gear	69 68	0.98551	Transfer Gear set					Non-Hunting	Sun	23	3.7	4th	3/4 Potential 5	YES	Seq.	Non-Hunting	
Pinion	21		Pinion	31													
Gear	59	2.80952	Final Drive					Non-Hunting	Internal	85							
GM 6L-70E	Sun	37	Ravignaux Gearset	1st & 2nd	3 POTENTIAL 4	YES	Sequential	Non Hunting	Toyota Aisin AA80E	Sun	30	Ravignaux Gearset	1st & 2nd	4	YES	C'Phase	Hunting
	P _{Sp}	25								P _{Sp}	17						
	P _{RP}	25								P _{RP}	20						
	P2	25								R1/2	74						
	R1/2	97								S2	34						
	S2	47	Ps	20													
	S3	49	2.82	3rd	3 POTENTIAL 4 / 5 / 6	YES	Seq.	Non-Hunting		S3	38	2.16	3rd	4	YES	C'Phase	Hunting
	P3	20								Ps	16						
	R3	89								PR	19						
										R3	82						

From Table 2, we have observed that many companies were more interested in increasing the pinion number selecting options in order to cover more design alternatives as primary purpose, while paying less concern on the phasing constraints which may affect the gear train NVH quality. However, all the traditional as well as the current gear train designs have applied the equally spaced planet arrangement on the carrier design.

Planetary Gear Train and its Design Considerations

As mentioned earlier, planetary gear train traditionally has imposed many design constraint rules to ensure that there is no major problem on installation, durability and NVH issues. The following listed some of the key design constraints:

1. Equally spaced arrangement: Majority of the planetary gear trains are using equally spaced pinion arrangement as shown on both tables 1 and 2. With this arrangement

each gear mesh theoretically could share the same amount of torque (load) transferring through the power flow path, so there is no abrupt torque change in the entire gear train at each revolution under the steady state condition, that also means there is no abrupt acceleration or deceleration variation (because $F = ma$, and F is constant), thus any noticeable NVH problem will be less likely occurred.

2. Planetary gear train phasing: Three types of phasing terms are commonly applied in planetary gear trains, namely; 1. Counter phasing, 2. Sequential phasing and 3. In phasing. They are depicted as [figure 1](#) shown. Their associated pros and cons are also listed for comparison purpose.

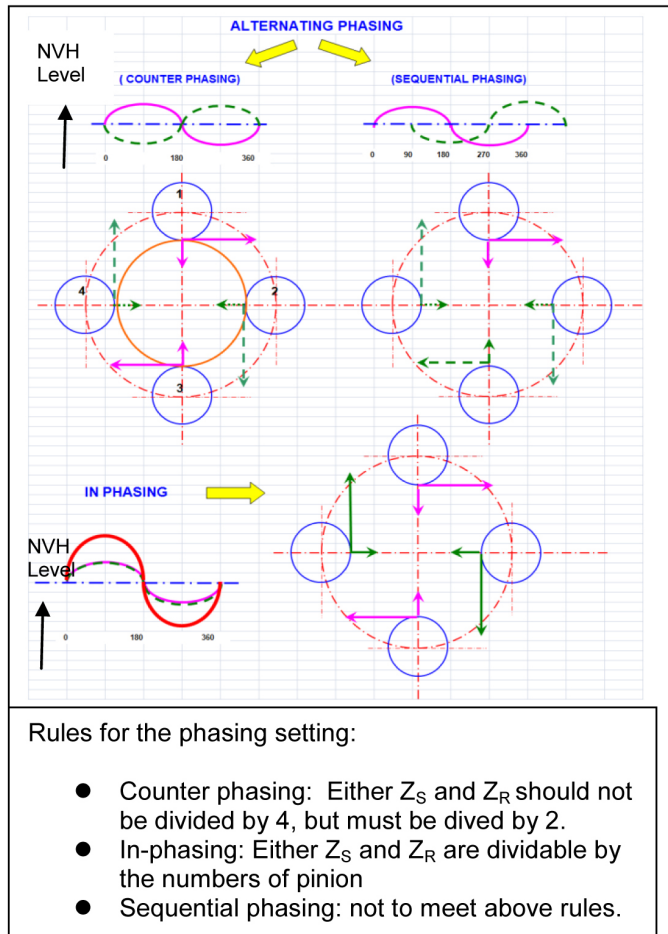


Figure 1. Planetary Gear Train Phasing

3. Balance of gear resultant forces: As shown on [figure 2](#), if we take four pinion carrier arrangement as an example: In order to meet the force balance requirement, all the summations of separate forces, and the tangential forces or the correspond resultant forces generated by gear sets to be equal to zero, no matter the carrier is an equally

or non-equally spaced arrangement. Similarly, the same force balance to be achieved for all other 3, 5 and 6 pinion arrangements.

Pros

Cons

- | | | |
|-----------------|---|----------------------------|
| C'phase | Better load sharing | High thrust load on sun |
| | Lower NVH level | |
| Sequent. | Less thrust load on sun gear | High load on each gear set |
| | Lower NVH level | |
| In-phase | 1. Very sensitive to pinion tangential position that could affect load sharing; and;
2. Both NVH amplitudes (or exciting forces) are stack together as shown to aggregate the noise level. | |

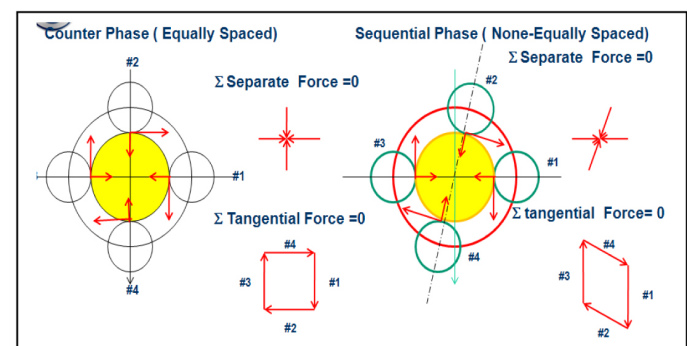


Figure 2. Planetary Gear Train Resultant Force Diagram

4. Planetary gear load distribution: Ideally, the equally spaced arrangement should be used for planetary carrier designs, so the load sharing on each power flow path could be equally distributed, this is one of the primary benefits for the planetary gear design. Nevertheless, in order to increase the gear ratio selection, we have tried applying the least-mesh angle (LMA) concept to explore the potential of non-equally spaced arrangement. The subject of LMA will be discussed in detail on the following section 6. Based on the LMA, very minor of the pinion position/s will be relocated for the assembly reason. The [figure 3](#) of using one of the simple planetary arrangements. In this case, the TT (Sun + Ring gear tooth number) equals to 110 as an example to illustrate the different carrier arrangements.

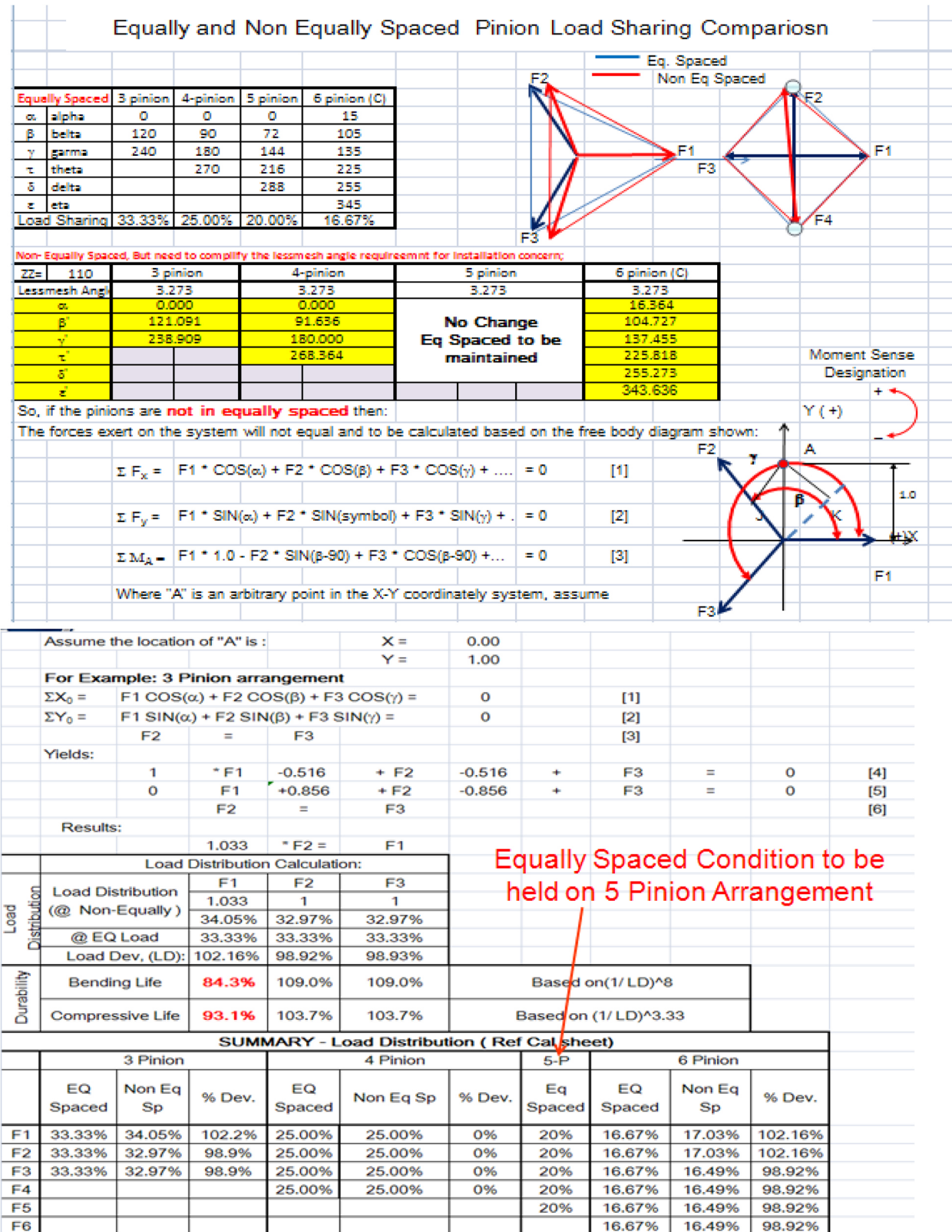


Figure 3. MP Planetary Gear Load Distribution

Notice that in this example, the maximum of total amount of load distribution variation for all none-equally spaced arrangements is less than 3.5%. On the following assessment section, we will further review and discuss the impact of this load distribution variation on both stress and

NVH areas. Also in this study, we have intended to keep the 5 pinion arrangement always in equally spaced condition to simplify both the design calculation and assessment.

5. Rules of assembly: Since all the gear sets in a planetary gear train must be meshed simultaneously together for

installation and load sharing reasons. There are rules governing the number of teeth in the sun and internal ring gear and the center distance of the pinions on the carrier.

- Rule 1. Both sun and ring gear teeth number must be in either even or odd.
- Rule 2. For evenly (equally) spaced pinion arrangement on carrier, for:

Simple Pinion : $(R+S) / N = \text{Integer}$;

Double Pinion : $(R-S) / N = \text{Integer}$

Where: S=Sun, R=Ring gear, N= No. of pinion

6. Apply Least-Mesh Angle (LMA) for alternative design: When we start the design of a pinion carrier, it is necessary to determine the number of pinions and their spacing based on the load requirement and packaging availability. Because in order to assemble both the sun and ring gears, the pinion must in increments of a certain angle as [figure 4](#) shown. The angle defined as “Least-mesh Angles” was first named by O.K. Kelly¹ back to 60s.

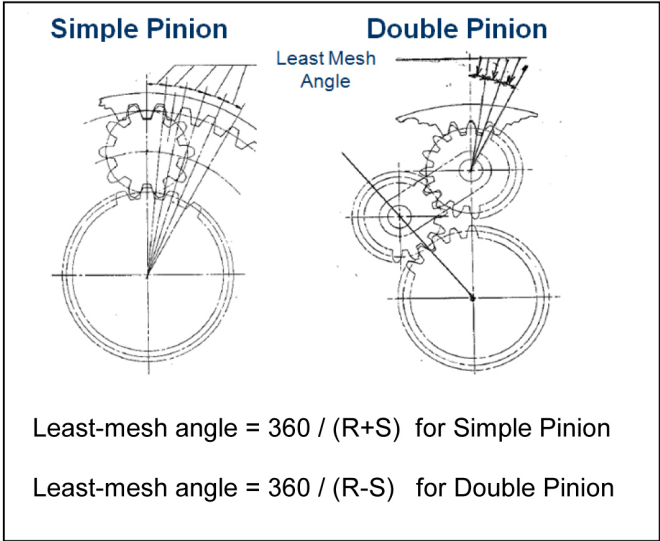


Figure 4. Definition of Least-mesh Angle

The following example ([Table 3](#) and [Figure 5](#)) are used to illustrate how to apply the LMA concept to generate both equally as well as non-equally spaced carrier arrangements on an arbitrary simple planetary gear train to enhance the gear ratio selection:

Table 3. Apply LMA for gear ratio Selections

Sun	Ring	R/S Ratio	Pinion #	Total	Eq. Spaces	Phasing	Total Angle	Least Mesh Angle(°)	#1	#2	#3				#4				
									0	27	28	54	55	56	81	82	83	84	
34	78	2.29	4	112	28	Counter	360	3.214	0		90			180					270
33	79	2.39	4	112	28	Sequential	360	3.214	0		90			180					270
33	77	2.33	4	110	27.5	Sequential	360	3.272	0	88.36	91.63			180				268.36	271.63
32	78	2.43	4	110	27.5	Sequential	360	3.272	0	88.36	91.63			180				268.36	271.63
31	79	2.54	4	110	27.5	Sequential	360	3.272	0	88.36	91.63			180				268.36	271.63
31	77	2.48	4	108	27	Sequential	360	3.333	0	90			180					270	
30	78	2.60	4	108	27	Counter	360	3.333	0	90			180					270	

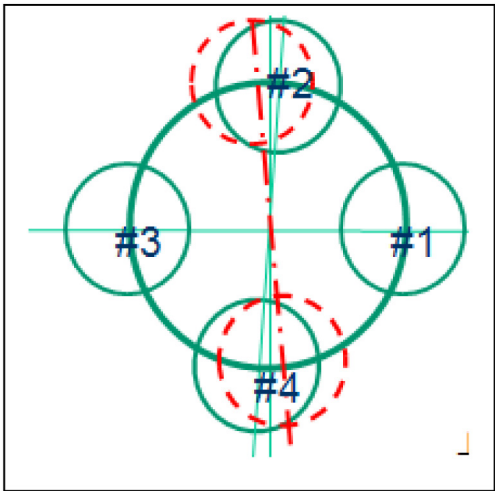


Figure 5. Equal and Non-equally Spaced Arrangements

Alter Carrier Arrangements for Increasing Gear Ratio Selections

The following exercises are processed to illustrate the potential of ratio expanding. Based on the LMA concept, additional non-equally spaced carrier arrangements could be applied on both simple and double pinion planetary gear trains:

a. Simple Planetary Gear Train Arrangements

Although the simple planetary gear train has been used in many applications crossed over various industrial fields, in this report, we are only interested on automobile applications. Based on this respective, the Ring to Sun (R/S) ratio ranges are targeted at around 1.8 to 3.5 and the minimum teeth number for sun gear is set at 30 to accommodate the in-line output shaft arrangement, and the pinion number is 17, to eliminate any severe undercut that could cause tooth bending failure