Service

Article #20

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“Final Drive Ratios”

by Tony Cripps.

1. Introduction

Final drive ratios for the Australian A series engines are a mixed bag. It’s worth having a look and trying to sort things out.

2. Applications

An examination of various Service Parts Lists and associated drawings show the following part numbers and specifications.

Crown wheel		Pinion						
Part No	**Teeth**	**Part No**	**Teeth**	**Ratio**	**Model**	**Engine**	**Reference**	**Note**
Mini								
22A3	64	2A3641	17	3.765	ADO15	850	AKD1153	A Type Cone
22A3	64	22A399	17	3.765	ADO15	850	HYL2980	A Type Cone & Baulk Ring
22A3	64	22A399	17	3.765	ADO15	850	PUB1011M	A Type Cone
22A401	64	22A399	17	3.765	ADO15	850	PUB1011M	A Type Baulk Ring
22A401	64	22A399	17	3.765	ADO15	850	PUB1011M	B Type Baulk Ring
22A277	62	22A288	18	3.444	ADO50	997	PUB1011M	up to 10447
22A401	64	22A399	17	3.765	ADO50	997	PUB1011M	
22A411	62	22A413	18	3.444	ADO50	997	PUB1011M	
22G101	62	22G99	15	4.133	ADO50	997	PUB1011M	
22A401	64	22A399	17	3.765	ADO50 YDO4 YDO5	997 998 998	PUB1011M	B Type
22A411	62	22A413	18	3.444	YDO6	1275	PUB1011M	B Type
22A1003	72	22G928	22	3.273	YDO5 Auto	998	PUB1011M	Automatic
22G787	70	22G785	24	2.917	ADO16 Auto	1275	PUB1011M	Automatic
22G101	62	22G99	15	4.133	ADO16	1098	PUB1011M	A Type Baulk Ring
22A301	64	22G99	15	4.267	YDO4 YDO5	1098	PUB1011M	B Type Baul Ring
22A401	64	22A399	17	3.765	ADO16	1275 12Y	PUB1011M	to some change point
22G101	62	22G99	15	4.133	ADO16	1275 12Y	PUB1011M	from change point

22A411	62	22A413	18	3.444	YDO6	1275	PUB1011M	YKG2S2 4 speed synchro
22A401	64	22A399	17	3.765	YDO4 YDO5	1098	PUB1011M	YJBAV4 YMGS1 YMG2S3 4 speed synchro
22G940	62	22A399	17	3.647	YDO21 YDO22	1098	PUB1052	
22A411	62	22A413	18	3.444	YDO23	1275	PUB1052	
22G940	62	22A399	17	3.647	YDO21 YDO22	1098	PUB23	
22A411	62	22A413	18	3.444	YDO21 YDO22	998	PUB23	Leyland Mini and Leyland Mini S
unknown	63	unknown	16	3.938	YDO21 YDO22	1275	PUB23	Allegro specs Leyland Mini LS 1275cc
Moke								
22G101	62	22G99	15	4.133	YDO7 YDO8 YDO18	998	PUB1029	
C22G370	64	22G99	15	4.267	YDO7 YDO8 YDO18	998	PUB1029	
22G101	62	22G99	15	4.133	YDO7 YDO8 YDO18	1098	PUB1029	
C22G370	64	22G99	15	4.267	YDO7 YDO8 YDO18	1098	PUB1029	
Unknown	72	22G99	15	4.800	YDO30	1275	PUB1029	
HYL8447	64	22G99	15	4.267	YDO18	1275	PUB27	
DAM3645	65	DAM3647	15	4.333	YDO18	1275	PUB27	99H905AJZH8 99 to 8426 then 8723 on

Table 1 Summary of Final Drive Ratios – Australian Mini and Moke.

Just how are these final drive ratios chosen? It all depends on the desired performance characteristics such as top speed, gradeability, and how this might be achieved with the engine torque curve. Details of this are given in another Article in relation to motor vehicle performance.

For this article, the actual way in which the gears are specified to give the desired final drive ratio is examined.

3. Gear Ratios

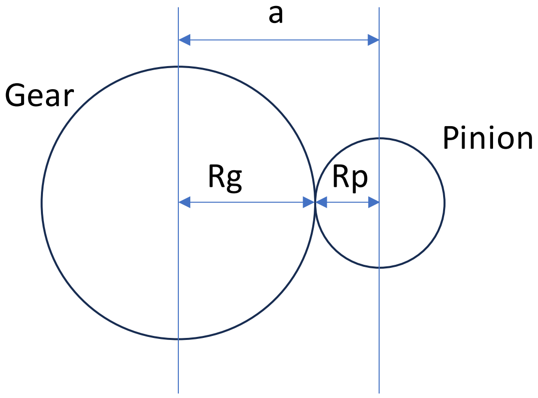
Imagine the final drive gears did not have any teeth at all, but just rubber rings around the edges – much like you see on a cassette tape recorder drive. It is the relative diameter of the pitch circles that gives the drive ratio. This ratio of pitch circle diameters is influenced by the requirement that each gear will have an integral number of teeth (no half-teeth for example) around its circumference, and this will in turn depend on the centre distance between their axes.

In our case, for a Morris Mini, the centre distance is fixed at 4.0” by the transmission case and final drive cover. How then do we determine how many whole number teeth should be on each gear at that centre distance for a desired final drive ratio (FDR)?

To see how this works, we take the centre distance as a fixed quantity – it is the distance from the centreline of the transmission 3rd motion shaft and the centre line of the crown wheel bearings. The desired final drive ratio is to be 3.765:1 (Morris 850). This might have been chosen as a result of previous performance calculations.

Question: what are the dimensions of each gear and how many teeth are on each for this FDR and centre distance?

If the centre distance is a = 4", then the addition of the pitch circle radii must be equal to this number – drawing a line from one centre to the other.



Thus, if R_p is the pitch circle radius of the pinion, and R_g is the pitch circle radius of the gear, then:

$$R_P + R_G = a = 4"$$

Also, the final drive ratio FDR is the ratio of these pitch circle radii, and so:

$$\frac{R_G}{R_p} = FDR$$

Combining these two, we obtain:

$$a = R_P + (FDR)R_P$$
$$a = R_P(1 + FDR)$$

This is the relationship between the centre distance, the final drive ratio, and the radius (or the diameter) of the pitch circle of the pinion.

We next need to determine what whole number of teeth will fit around these pitch circle diameters to give the desired FDR. This is not so easy to do.

Gear teeth are characterised by what is known as the diametral pitch *P*. By definition, *P* is the ratio of the number of teeth *N* and the pitch circle diameter *D*. For the pinion, we have:

$$P = \frac{N_p}{D_P}$$

For what is called "conjugate action", (that is, the teeth mesh cleanly without slipping), the diametral pitch of the pinion has to be equal to the diametral pitch of the gear.

And so, for the gear,

$$P = \frac{N_G}{D_G} = \frac{N_P}{D_p}$$

The diametral pitch may also be expressed in terms of what is called the "module". The module is the inverse of the diametral pitch and is expressed in mm and is used for "metric" gears, while diametral pitch has units of inverse inches (in^{-1}) and is used for "imperial" gears.

The diametral pitch is a number that characterises the size or spacing of the gear teeth relative to the gear's diameter. The actual pitch, *p*, the distance of the arc between two similar point on the gear tooth, is

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

So, what we do is match the diametral pitch with the centre distance until we get a whole number of teeth.

Start with the pinion.

The pinion radius and diameter can be calculated from the centre distance *a* and the desired final drive ratio FDR using the equation above:

$$\begin{aligned} a &= R_P(1 + FDR) \\ R_p &= \frac{a}{1 + FDR} \\ &= \frac{4}{1 + 3.764} \\ &= 0.851" \\ D_P &= 1.702" \end{aligned}$$

For tooling purposes, the diametral pitch *P* for BMC gears is 10 (in inverse inch units). And so the number of teeth that would result is:

$$\begin{aligned} N &= P(2R_p) \\ &= 10(2)(0.862) \\ &= 17.021 \end{aligned}$$

Now, it is obvious we cannot have 17.021 teeth on the pinion, so round this to the nearest whole number and we settle on 17 teeth for the pinion.

Using 17 teeth, and staying with 10 as the diametral pitch, the new pitch circle diameter for the pinion becomes:

$$\begin{aligned} D_p &= \frac{17}{10} \\ &= 1.700" \end{aligned}$$

What now for the gear?

Working backwards from the centre distance, the desired pitch circle radius for the gear is thus:

$$\begin{aligned} R_g &= a - \frac{D_p}{2} \\ &= 4 - \frac{1.7}{2} \\ &= 3.15" \\ D_g &= 6.3" \end{aligned}$$

Using the same value of diametral pitch *P* = 10 as for the pinion, we get the number of teeth for the gear as:

$$\begin{aligned} N_g &= 10(6.3) \\ &= 63 \end{aligned}$$

Hold up! You say. 63 teeth? This would give a final drive ratio of:

$$FDR = \frac{6.3}{1.7}$$
$$= 3.705$$

Which is different to the target of 3.765. Plus, we thought that the gear would have 64 teeth. What gives?

Well, we did the above calculations on the basis that everything was perfect, and that there were no clearances between any of the gear teeth. In practice, a gear design engineer allows for a certain amount of backlash to account for thermal expansion of the parts, and space for the oil, and other manufacturing tolerances. The effect of machining the gear teeth to create these clearances effectively widens the centre distance – as if perfect gears were placed a little bit further apart than the ideal geometrical centres. In our case, for BMC, the effective centre distance is increased by 0.05” which makes the effective centre distance equal to 4.050”.

When you do these calculations again with that centre distance, you will find that the number of teeth on the gear comes to exactly 64 teeth, giving a final drive ratio of 3.765 as advertised.

While the above applies to spur gears, it applies as well to helical gears when we consider the geometry in the transverse plane (looking face on to the gear). We need only concern ourselves with the helical nature of the gears when it comes to designing the pressure angle, line velocity, gear loading in transverse and normal directions, and so on.

Shown below is the data taken from engineering drawings for the pinion and gear shown above.

		Pinion	Gear	
	PART Nº	22A309	22A 402	
	NUMBER OF TEETH.	17	64	N
*	TRANSVERSE PITCH.	10	10	P_t
	PITCH CIRCLE DIAMETER.	1·7"	6·300"	D
*	HELIX ANGLE.	30° L.H	30° R.H.	φ
*	NORMAL DIAMETRAL PITCH	11.547	11·547	P_n
	ADDENDUM.	·0866"	·0866"	
*	NORMAL PRESSURE ANGLE	17° 29' 43"	17° 29'-43"	
*	TRANSVERSE PRESSURE A.	20°	20°	
	OUTSIDE DIAMETER.	1·868" / 1·873"	6.47"	
			CENTRES	

Items marked * refer to the cutter

4” centres

This article introduces you to the concepts of gear design at BMC, and indeed, in any industrial application. I hope you can appreciate some of the detail in which engineers consider to obtain the final drive ratio desired for a particular model.

Tony Cripps

October 2023