Transmission Techniques

The Gearbox—Its Job and Its Design

PART ONE

By PHIL IRVING, *M.I.Mech.E., M.S.A.E., M.I.P.E.*

IN some ways, a petrol engine is about as bad a choice as could be made for propelling a road vehicle. It cannot start itself, it must be rotating at a reasonable speed before it can develop any usable power, and if overloaded sufficiently it will stall.

To overcome these shortcomings, it must be provided with a transmission system which will enable the drive to be disconnected or taken up smoothly at will by some sort of clutch, and which will also provide some means whereby the engine speed can be varied in relation to the driving wheels to suit varying road conditions.

There has been a prodigious number of solutions to this problem, including simple two-speed engine-shaft gears, epicyclic multi-speed gears together with a friction clutch built into the rear hub, and almost infinitely variable drives with V-belts running over expanding pulleys—to mention only a few. But in the end the system which has won the day is the countershaft box mounted between engine and rear wheel and containing any number of ratios from two to four on touring motorcycles and from four to six on racing models; in fact even a seven-speeder has recently made its appearance.

As most racing is done on fairly level surfaces and with the machine as lightly loaded as possible, it may seem odd that more ratios are employed than is usual with touring machines which are expected to climb steep gradients or traverse mud or sand when heavily laden. The underlying reason, however, is quite simple.

Torque-converter

In a tourer the gearbox is there primarily to act as a torque-converter, and the provision of a bottom gear with, say, a 3 : 1 speed reduction in the box, multiplies the torque of the engine at any given speed by 3. The pull exerted at the rear wheel is thus increased by a similar amount and simultaneously the engine is able to turn over at a speed high enough to be developing a reasonable torque. The two effects taken together enable the machine to climb a hill, even at slow speed, in a way which would be quite impossible in a higher gear.

It is worth noting here that the clutch is in no sense a torque-converter. When it is being slipped it is really acting as an energy-absorber.

The only reason why slipping the clutch will sometimes enable a machine to be extricated from a bad situation is, that it allows engine speed to rise well above the stalling speed to a point at which the motor is capable of developing perhaps three or four times the horsepower, power being proportional to torque multiplied by speed. Then, even if half the power developed is wasted as heat through friction in the slipping clutch, the net result is more pull applied at the rear wheel—and also, of course, a burnt-out clutch if the process is continued for any length of time.

Choosing the Ratios

The highest gear is normally chosen to suit the weight and bulk of the machine, not necessarily to give the highest maximum speed, but rather to give a comfortable cruising speed with the engine turning over at around 4,000 r.p.m., or somewhat less in the case of large-capacity power units.

This gear is so much higher than any useful bottom gear could be that the gap between the two is too great either to suit the torque curve of the engine or to permit easy changing, so it is necessary to insert at least a middle gear to bridge the gap and to provide a ratio which is useful on normal hills.

For many years, three speeds were considered ample—and, provided the engine has a torque curve which does not droop badly at low engine speeds, that number is still sufficient for a bread-and-butter mount, especially where weight and bulk are items to be considered.

The snag with only three ratios is that there is bound to be a fairish gap between top and second and often, when hill-climbing, one has to choose between slogging up at a slow rate in top gear or going at an almost equally slow rate with the engine screaming round in second.

This situation can be surmounted by including a third gear between top and second. Then, if the two upper ratios are fairly close—somewhere in the region of 1.25 : 1—it is possible to over-gear a little in top to keep the cruising revolutions down, and rely upon third for moderate up-grades or for routine acceleration to get past a slower vehicle quickly.

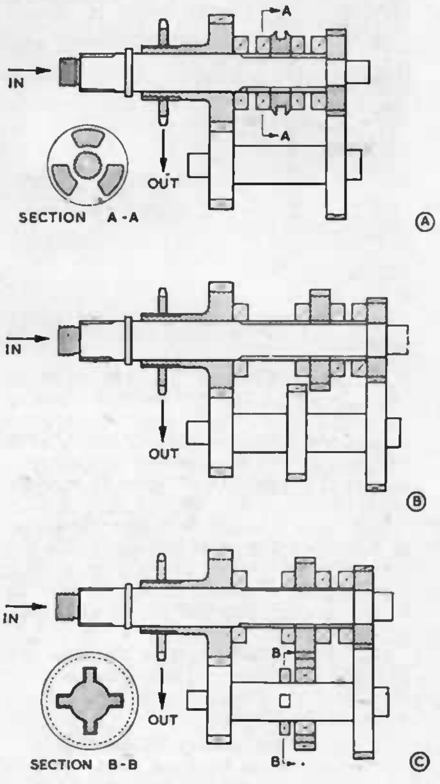
Given four nicely chosen ratios, the low-speed pulling power of the engine ceases to be very important and it becomes possible to use longer valve timings and higher compression ratios, thus providing more top-end power with little or no sacrifice in fuel economy.

In road-racing, the box still acts as a torque-converter to improve acceleration at low speeds, but the power-to-weight ratio is very high and there is no point in having the lower ratios so low that wheel-spin will set in with the throttle only partly open.

(As a matter of strict terminology, a ratio of, say, 10 : 1 is higher than one of 4 : 1, although the gearing is lower. However, it has come to be colloquially acceptable to refer to a low gear as a low ratio.)

Also, the solo road racer rarely has to make a clutch start from a standstill, so its low-speed performance is not a very vital factor. On the other hand, as more and more power is wrung out of engines by raising their normal speed into five figures, the low-speed end of the power curve virtually vanishes and it becomes necessary to keep the engine turning over at nearly its maximum speed all the time by making full use of the gearbox.

If, for instance, an engine has a usable power-band between 8,800 and 10,000 r.p.m., the widest gap between any two ratios should be not greater than 12%—preferably

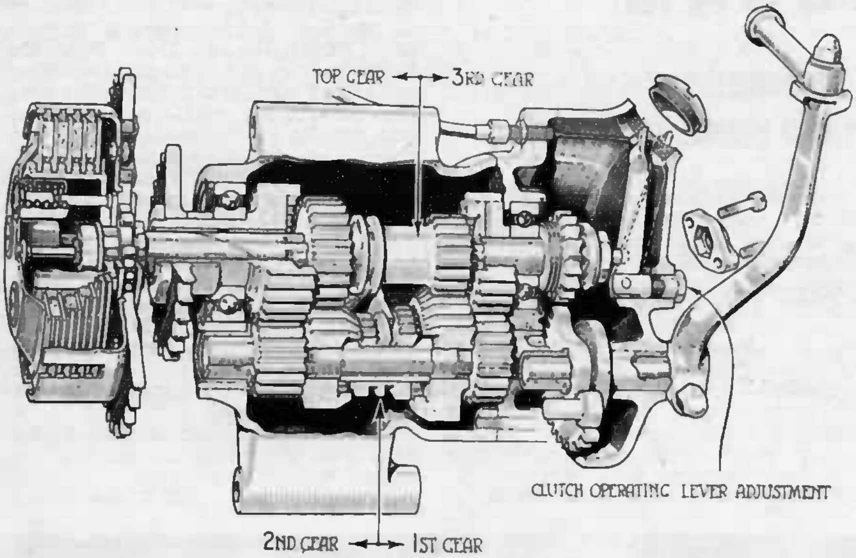


Three basic gearbox layouts. A: Simple two-speed with dog engagement. B: Three-speed, with dogs for first and top and "crash" change for second. C: Three-speed all-constant-mesh, with dog engagement for first and top and short splines for second.

rather less—otherwise, when an up-change is made, the revs. will drop below the minimum unless they are taken up above the permissible figure before the change, a practice which is not to be recommended. It is simple enough to over-rev. an engine inadvertently at such a moment, without the designer going out of his way to make matters worse.

On almost any circuit there is some corner which cannot be negotiated above a fairly modest speed, or a tricky up-hill stretch which demands a lowish bottom gear to avoid excessive, even if temporary, clutch-slipping. It then becomes necessary to fill in the gap between the top gear required to give the highest maximum speed and the essential bottom gear, with enough ratios to suit the characteristics of the engine.

With five or six ratios, it becomes possible to arrange matters so that the highest is used only on down-hill sections or with a strong following wind, thus taking full advantage of the favourable conditions whilst holding the engine revs. below the permitted maximum. This is obviously both faster and safer than gearing exactly for level-road, still-air conditions and trusting to the rider to "roll it back" when the revs. go too high, especially as he may be rather loath to do so if it means losing the lead into the next corner.



A straightforward example of the four-speed box—the Burman—with bottom and second gears on the layshaft and top and third on the mainshaft.

General Design

Of course, it is one thing to say how many ratios are needed. It is quite another to translate the needs into concrete form—in the least expensive manner consistent with long life on touring models, and with the emphasis on lightness consistent with adequate strength for racing, where cost is a secondary consideration.

With a separate box and all-chain drive, it is desirable, and nowadays almost standard practice, to locate both chains on the same side, with their respective sprockets co-axial, to reduce twisting moments on the mountings. The rudimentary two-speed box then consists of a clutch-shaft, extending right across to the far side and carrying one loose pinion, a co-axial sleeve carrying another gear inside the box and the final sprocket outside, with another shaft, the layshaft, lying parallel to the clutch-shaft and carrying two gears splined, keyed or even made integral with it.

By cutting dog-teeth in the sides of the gears on the clutch-shaft, splining this component and mounting thereon a double-faced dog-clutch operated by a shifter fork, a direct drive top gear is obtained by engaging the sleeve-dogs, and bottom by engaging the clutch-shaft pinion. By mounting the shafts on ball-bearings and cutting down the "neutral" clearance between the dogs, to reduce the shaft-lengths as much as possible, the box can be made very narrow, quiet-running, easy to change and with a transmission efficiency of 95% or better in bottom gear.

This design can be altered to a three-speed version simply by adding external teeth to the dog-clutch and another gear to the layshaft, so that their respective teeth are slid endways into mesh to obtain a middle ratio by what is called a "crash" change, which demands much more finesse in handling the controls than does the dog-tooth change. This type of box, though popular for a while, was rapidly superseded by several variations in which the second-gear wheels also were constantly in mesh, so providing a "constant-mesh" box which could be changed up or down with ease. This box is naturally wider than the two-speeder, but the change mechanism is still very simple.

Adding a Fourth Ratio

The position alters drastically as soon as a fourth ratio is introduced, and complications in one way or another arise which inevitably make the four-speeder more difficult to produce.

It is possible—in fact it was accomplished in the extinct "Jardine" box—to obtain four speeds with one sliding dog and several concentric components, but the need to keep the overall width of the box down to a figure commensurate with the space available almost enforces the use of sliding members on both the mainshaft and the layshaft—which immediately increases the difficulty of operating these members in the correct sequence.

In one way, the lack of available space has been a blessing in disguise, because it automatically ensures that the shafts are short and thus reduces their deflection under load. Outward deflection, which causes the teeth to run slightly out of mesh, could be a fruitful source of noise and tooth-wear, and everything must be done to avoid it.

Shifter Fork Layout

In the Albion design, changes are made in sequence straight across the box, by means of a double fork, engaging with the components sliding on the main shaft and layshaft. As this is moved from left to right, top, third, second and bottom are brought into action in turn, either by means of face dogs or interrupted splines on the shafts. The idea has recently been extended to incorporate a fifth gear, still with the same relatively simple shifting mechanism.

For various reasons, it is more usual to use two separate shifter forks operated independently. For one thing, it permits the third gear pinions, which are the most-used of all the indirect gear wheels, to be located close to the bearings in the gearbox end-cover, a position in which they are the least likely to suffer the ill-effects of shaft deflection.

Bottom and second gears, which are not used so much and in which silence is not so vital, are located in the centre. They are changed by dogs on the layshaft, while top and third are changed by dogs on the mainshaft, but for a third to second change, or vice versa, one dog must be moved into the neutral position and held there while the other is moved to engage the gear desired.

Gate Change?

This compound movement is simple enough to obtain if something like the gate or ball change used on a car gearbox could be employed. While this is not impossible—and, in fact, some machines have had this feature—it simply is not a very convenient system on a motorcycle with hand-change, and is quite out of the question when a foot-change is employed.

One of the first really successful four-speed boxes which could be operated by a single rod and a lever working in a quadrant was the Rudge-Whitworth, in which the shifter forks were actuated by pegs running in slots formed in a curved plate. This plate was simply mounted on a spindle with an external lever, and only about 90° of movement was required to change right through from top to bottom.

On the other hand, it was a "progressive" change; it was not possible, as with any form of gate, to select any one gear without going through intermediate ones. Not that this really matters, and in practice it is probably a good thing because there is less chance of inadvertently, or through ignorance, endeavouring to effect a change involving an alteration of engine-speed which is too great for the circumstances.

(To be continued)

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The Gearbox—Its Job and Its Design

PART TWO

By **PHIL IRVING**, *M.I.Mech.E., M.S.A.E., M.I.P.E.*

MOST modern gearchange systems employ either a flat plate with what the patent attorneys describe as "arcuate slots therein," or a drum with wavy slots milled in the periphery. In the former design, first adopted in the Sturmey-Archer box, the forks slide on a bar (or two bars), but in the latter they can be carried by the drum, on which they are a sliding fit.

Many successful examples of both types have been built, and it is hard to say which is intrinsically the better. Much depends upon the general design of the box and the facilities available for its manufacture.

One difficulty with a circular cam-plate is that it has to be several inches in diameter and of reasonable thickness—$\frac{3}{16}$ in. or so. It therefore possesses a fair amount of inertia which, when a quick change is made, tends to make the plate overrun the correct position, so that it brings the gears into the next neutral position instead of engaging the next ratio.

By virtue of its much smaller diameter, a cam-drum possesses far less inertia and is almost immune from this trouble.

A cam-plate which moves through only a small angle and obtains the necessary length of fork movement by leverage is an increasingly popular alternative to the circular plate. It is especially suitable for unit or semi-unit designs in which it is not particularly easy to couple a rotary plate to the ratchet mechanism embodied in the foot-change.

Besides making gearchanging so easy that synchro-mesh devices are unnecessary, the constant-mesh principle with dog engagement is a help in accommodating the widest possible teeth in the available space, and also in reducing the distance any dog has to travel.

When not actually transmitting power, mating gear-teeth need not overlap by more than ¼ in. or so, thus reducing power-loss from excessive oil-churning. Clearance between dogs when in neutral can be reduced to 0.060 in. and movement for full engagement need not exceed .250 in., so that the total length of movement from, say, top to third, is .625 in. on a heavy-weight box, and can be less than this if space or weight reduction demands it.

Forming the Dogs

Dogs can be formed in various ways. External ones may be milled into the end-faces or, where pinions with 18 teeth or so are concerned, their own teeth may form the dogs.

Internal dogs may be formed either by drilling holes, by splining or by gear-shaping. It is preferable to make the load-carrying faces on all members of equivalent shape to reduce impact wear; but if there are a number of dogs, so that the load on each is relatively light, it does not matter much if the contacting faces are not quite of the same geometrical form.

Backlash between the dogs is detrimental at low speeds, tending towards transmission-snatch in top or third gear, especially with a single-cylinder engine, but it is desirable in the interests of rapid gearchanging because the greater the backlash the more time is available for the dogs to enter.

When using gear-teeth as dogs, it is possible to allow only 3° or 4° of backlash without making the mating dogs too frail, but a compromise can be effected by cutting back every alternate tooth on both members for nearly half their length of engagement. This allows about 20° backlash at the commencement of engagement, after which the dogs can slide right home and then be almost free of backlash.

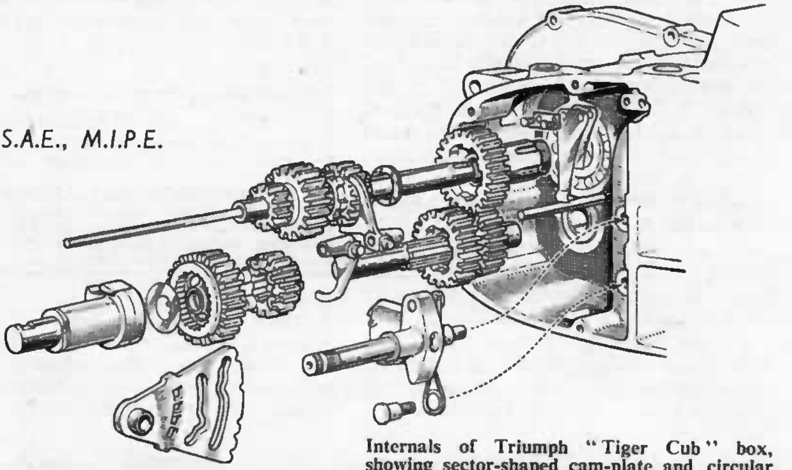
However, if the change is missed, or the "proud" dogs become rounded, there is a chance that the dogs will in effect just bounce over the top of each other and it will be difficult to complete the change at all. For fast work, therefore, it is better to cut away completely each alternate tooth, though obviously there must be an even number of teeth originally, a fact which places some limitation on the design. In the lower ratios, backlash is not detrimental when running slowly and it is best to allow plenty, especially in bottom gear, to facilitate engagement from neutral with the model stationary.

Face-dogs are sometimes undercut at an angle of 2° or so in order to prevent them working out of engagement. This scheme adds to manufacturing difficulty and, if overdone, may make disengagement difficult when the clutch does not free perfectly.

In any case, it cannot be applied to dogs formed by gear-shaping, which are not very prone to slipping-out until badly worn and rounded.

With both drives on the same side, the loads on the main bearing are partially balanced, and there is no relative motion between the clutch shaft and sleeve-gear when in top, and even in bottom the relative speeds are at worst in the region of 1,500 r.p.m. It is usual to employ a long bronze bush to centralize the clutch-shaft, but in some racing boxes needle-roller bearings are fitted. For the layshaft, either bronze bushes or ball or roller bearings are used, the choice depending on the space available and the lengths to which one is prepared to go to reduce friction.

The Velocette box reverses conventional practice in that the clutch-sprocket lies *inside* the final drive, this situation being actually an inheritance from the original two-stroke layout. Its retention is justified on the score that overhang of the engine sprocket is reduced to the barest minimum, and also because it permits the final drive ratio to be altered very quickly and easily. The clutch throw-out mechanism is unusual, and is sometimes maladjusted by those who fail to follow the stipulated sequence of operations; that, however, is no fault of the design.



Internals of Triumph "Tiger Cub" box, showing sector-shaped cam-plate and circular bottom-gear dogs engaging in elongated slots.

Sometimes, the overhang of the clutch-sprocket dictated by the combination of a proprietary conventional box and a wide primary chain-line is very excessive. In fact, there seems to be little point in retaining this type of box when semi-unit construction is adopted, except that it enables existing components to be employed.

Cross-over Boxes

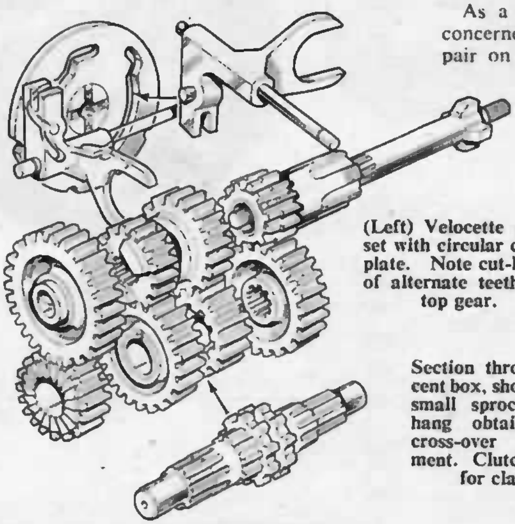
With a cross-over box, there is no partial balancing of the loads due to chain-pull on the bearing adjacent to the sprocket, and robust ball-bearings are needed on both sides, but when the primary and final chain-lines have been settled to suit the engine and rear-wheel layout, the sprocket overhang on both sides can be reduced to a minimum, which is a very desirable thing. Unfortunately, the presence of a chain on the same side as the kick-starter and foot-change complicates the design considerably on touring models; this condition does not occur on pure racing models, and most of the unit-construction Continental racers for many years have had cross-over boxes.

Five Speeds

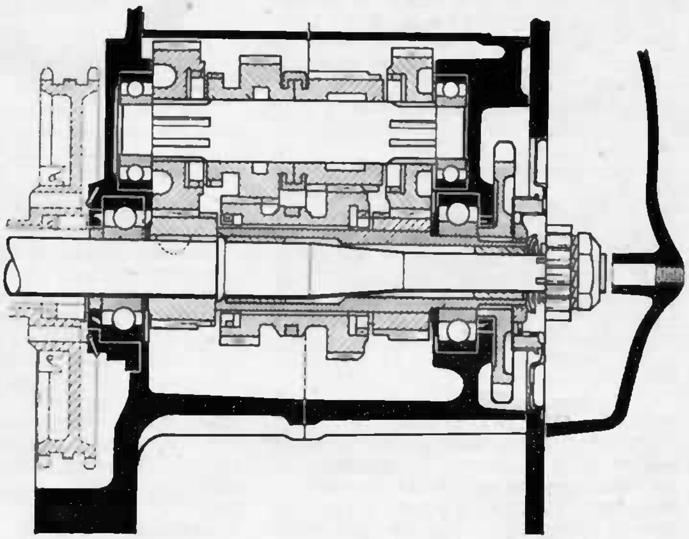
While it appears that five ratios could be obtained from six pairs of wheels by compounding them, the scheme is not very practicable and places limitations on the steps between ratios which may not be acceptable. The usual practice is to add another pair of gears with another shifter fork and a third slot in the cam-plate, in which event fifth gear can be either direct or geared-up to give an overdrive effect.

This raises the question of efficiency. In direct drive, the only power-loss is that due to sliding friction and oil-churning in the box, both being small because none of the rubbing surfaces is carrying load at the time. In the indirects, power is transmitted through two sets of gear-teeth and the overall efficiency in a well-designed unit would be about 96%, but might easily be less.

Though the loss of 4% is not of much moment in touring, in racing it is serious, especially as it is always one of the lower ratios which is in use whenever maximum power is wanted: top gear is strictly for the easy bits. Therefore it is logical to resort to an all-indirect box, which should possess around 98% efficiency in all ratios, and this is the general trend today, especially for five- or six-speeders with reductions of less than 10% between each ratio.



(Left) Velocette gear set with circular cam-plate. Note cut-back of alternate teeth on top gear.



Section through Vincent box, showing very small sprocket overhang obtained with cross-over arrangement. Clutch omitted for clarity.

To minimize parasitic drag, all the "free" gears are sometimes mounted on caged roller bearings, but, even when the gear and dog widths are reduced to the barely safe minimum, the length of the shafts in relation to their diameters begins to become excessive for high powers. However, the scheme works well with moderate powers and high revolution rates, where the actual torque being transmitted is not unduly high.

All-indirect boxes are not very suitable for touring use, because it is difficult to obtain the required difference between the lowest and highest ratios when only one pair of gears is involved in each reduction.

For all ordinary use and most forms of competition, four speeds are sufficient, but the actual ratios depend upon the engine characteristics and the class of work in view. Some models lend themselves to adaptation to various sorts of competition, and the boxes are designed so that different gears can easily be fitted to alter the internal ratios—a procedure which is inevitably simpler with a conventional box than with a cross-over drive.

As one instance of such variations, the following table shows the gear-sets available for the B.S.A. "Gold Star," and is representative of the ratios which have been found suitable for the work listed.

	Top	3rd	2nd	1st
Road Racing ..	1	1.1	1.33	1.75
Standard ..	1	1.21	1.76	2.58
Scrambles ..	1	1.32	1.75	2.34

The figures apply to the *box* ratios only; top gear being fixed by the number of teeth on the sprockets. The choice of ratios in Albion gearboxes is even wider, up to a dozen being available in the heavy-weight types.

Tooth Proportions

As a general rule, where spur gears are concerned the total number of teeth in any pair on parallel shafts always comes to the same number, this being numerically equal to twice the centre-distance multiplied by the diametral pitch (D.P.) of the gear teeth. The D.P. is a nominal figure, not an actual measurement, and is equal to the number of teeth divided by the pitch diameter; common figures are 10, 12 or 14, according to the torque capacity required, the smaller number denoting the coarser (and stronger) teeth.

"Stub" Teeth

Sometimes, "stub" teeth are employed, these being designated by a two-figure symbol such as 10/12. This particular pitch is used, for example, in the Vincent box and denotes that the teeth are spaced at 10 D.P. but only as deep as those of 12 D.P. This system gives a stronger tooth for the same outside diameter, but at the expense of a little greater "separating force" tending to bend the shafts. As to the actual numbers of teeth, 18 is preferably the smallest size, though pinions with only 16 teeth have been used.

Sometimes, in endeavouring to obtain a particular ratio, one is faced with the fact that a reduction of, say, 18 : 30 gives too low a figure, yet 19 : 29 is too high. The way out of this is to cut 18 teeth on a 19-tooth blank and use this pinion in conjunction with a 29-tooth gear, thus obtaining a ratio midway between the other two and, incidentally, a much stronger pinion.

This system will not work with gears cut with a form-cutter in a milling machine, but only with those generated by gear-shaping, and it offers a convenient way of changing a ratio in small steps simply by changing one pinion instead of a pair. In the Vincent box, the second-gear layshaft pinion was changed from 24 to 23 to lower the ratio a little, without having to modify the rather complex double-gear component with which it meshed.

Gear and Shaft Materials

The steel used for the gears can be either a direct-hardening variety, such as 1½% nickel-chrome oil-hardening steel, or a case-hardening steel with high core-strength, such as EN 36, a 3% nickel steel which gives very good results and is not subject to much distortion during hardening if the blanks are stress-relieved or annealed before gear-cutting.

Some makers use a direct-hardening steel, such as KE 805, and heat the parts before final quenching in a carburizing salt bath for about 20 minutes, so forming a thin but very hard skin on the teeth which resists wear yet is unlikely to flake off or "spall," as thicker cases sometimes do, especially when the core is deficient in hardness.

Where the shafts are concerned, strength in bending is the main attribute, and this is a function of the diameter and length, not of the hardness of the steel. Surface hardness is, however, needed for the bearing surfaces, and often good quality case-hardening mild steel will be sufficient for strength whilst providing a dead-hard surface. For more arduous work, 3% nickel C.H. steel may be necessary, and it is desirable for any shaft formed with small projecting dogs or interrupted splines.