A Novel Design of 7 Speed Manual Gear Box

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Abstract
Almost all automobiles as of today are driven by internal combustion engines because of the high power to weight ratio, relatively good efficiency and compact energy storage associated with it. The major disadvantage with this engine is its incapability of producing torque from the rest, which is the characteristic of ideal traction hyperbola. This difference between characteristics curve of the combustion engine and the ideal traction hyperbola can be approximated by the use of a gear box.

In this paper a gear box with 7 forward gears and 2 reverse gears has been described. A normal 7 speed MT requires 8 gear pairs to provide 7 forward and 1 reverse but this paper presents a novel 7 speed MT designed with just 6 gear pairs giving 7 forward and 2 reverse gears thereby saving both the material cost as well as meeting the space constraints.

Keywords

I. Introduction
Automobiles are one of the most trusted and popular means of transportation available in the world. With the invention of IC engines, the application of automobile for transportation has increased significantly contributing greatly towards a developing civilization. Cheap transportation cost with safety in driving, makes automobile as one of the best means available for industrial applications. With more development in industrial sector the numbers of automobiles in the road are also increasing rapidly. But with the energy crisis, fluctuating price of the petroleum fuels and environmental damage caused by the automobiles, continuous research and development work is going on to improve the overall vehicle efficiency.

In a developing country like India where people are very much cost conscious, latest technology such as hybrid and electric vehicles are not the immediate solution. If the conventional vehicle is considered, its energy efficiency can be improved by increasing the engine efficiency or by modifying the transmission system of the vehicle that transfers power from the engine to the wheel. Modifying the engine is costlier, at the same time results achieved is also inferior against the transmission modification [1]. Thus a considerable effort is going on in the research area for the development of better transmission that can enhance the vehicle energy efficiency.

Packaging is always an issue for an automobile as the automotive industry is moving towards improving the fuel economy and reduction of weight. Transmission, which is one of the heavy components in an automobile when modified, can help in optimizing the overall vehicle efficiency.

Even after the development of many new technologies in the area of transmission like AT, AMT, DCT still the most efficient transmission with best fuel economy available in the present scenario is manual transmission [1]. Along with the advantage of best fuel economy it also has some disadvantage such as compromise in comfort and also the size of the transmission. The MT uses compound gear train to transfer power from engine to wheel whereas in the advanced technology planetary gear trains are used which are more compact. Thus there is always a scope of improving the MT from the compactness point of view. Thus, a novel approach to reduce the transmission size is proposed here which has the potential to improve fuel economy, reduce the material cost as well as solve the packaging constraint.

II. Gear Ratios for 7 Speed MT
To design a MT for a SUV, following specifications were taken to estimate the gear ratios by referring [2].

Table 1: Specification of SUV

| Max. Torque | 320Nm @ 2700 rpm |
| Wheel radius | 600 mm |
| Rolling resistances | 0.0165 |
| Vehicle mass | 2780 kg |
| Density of air | 1.2251.2kg/m3 |
| Coefficient of drag | 0.3 |
| Grade angle | 15 degree |

Considering requirements of a SUV, the Overall gear ratio was calculated as

Overall gear ratio: \( I_o = 16.81 \)

Final reduction = 4.1

\( I_o = 16.81/4.1 = 4.1 \)

Considering progressive gear steps following results were obtained [2].

Table 2: Intermediate Gear Ratios

<table>
<thead>
<tr>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
<th>5th</th>
<th>6th</th>
<th>7th</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.1</td>
<td>2.553</td>
<td>1.749</td>
<td>1.318</td>
<td>1.093</td>
<td>1.00</td>
<td>0.77</td>
</tr>
</tbody>
</table>

III. Gear Design
1. Gear material : 16Mn5 Cr4 [3]
3. Check for dynamic load on Buckingham equation
4. Check for wear strength and hardness estimation

IV. Shaft Design
1. Shaft material: C-50 hot rolled / forged [5].
2. Calculation of bending moment.
3. Calculation of maximum Twisting moment.
4. Estimation of shaft diameter Based on maximum shear stress theory.
5. Check for allowable deflection of 0.01 x module.

V. Solid Model of Gear Box

A. Parametric Design
Parametric design implies the use of parameters to define a form when what is actually in play is the use of relations. It is also known as relational modeling or variation design or constraint based design. It is a method of linking dimensions and variables.
to geometry in such a way that when its values change, the parts design changes as well. A parameter is a variable to which other variables are related, and these other variables can be obtained by means of parametric equations. In this manner, design modifications and creation of a family of parts can be performed in remarkably quick time compared with the redrawing required by traditional CAD. Parametric modification can be accomplished with a spreadsheet, script, or by manually changing dimension text in the digital model [6].

B. Input Parameters for Involute Gear Profile
Various input parameters used in 3D modeling of involute gear profile are as follows.
1. Pressure angle : "a"
2. No of teeth : "z"
3. Module : "m"

C. 3D Model of 7 Speed MT for SUV
Pitch circle radius : "r"
Outer circle radius : "Rk"
Root circle radius : "Rf"
Base circle radius : "Rb"

D. Variable Parameters
1. No of teeth : "z"
2. Pressure angle : "a"
3. Module : "m"

E. Parametric Relations: [7-8]
- Pitch circle radius : $r = m \times \frac{z}{2}$
- Base circle radius : $R_B = r \times \cos \left(\frac{\pi}{a}\right)$
- Root circle radius : $R_f = r - 1.25 \times m$
- Outer circle radius : $R_k = R_f + 1 \times m$
- $LAW X \cdot X = R_k \times \sin \left(\frac{T \times \pi \times 1rad}{x}\right) - R_k \times T \times \pi \times \cos \left(\frac{T \times \pi \times 1rad}{x}\right)$
- $LAW Y \cdot Y = R_k \times \cos \left(\frac{T \times \pi \times 1rad}{x}\right) + R_k \times T \times \pi \times \sin \left(\frac{T \times \pi \times 1rad}{x}\right)$

F. Advantage of using the New Concept
The higher the number of gears the better is the approximation between the Engine characteristics curve and Traction hyperbola. As with higher number of ratios the engine traction available touches the traction hyperbola in more number of points. But with the higher number of gears the design and packaging complication also increases thus there is a tradeoff between the more number of gear ratio and packaging complication [2].
Fig. 4: Power Flow Diagram for 7 Speed Gear Box

Table 3: Showing the Exact no Teeth & Gear Reductions for 7 Speed MT

<table>
<thead>
<tr>
<th>Gear name</th>
<th>Teeth on counter</th>
<th>Teeth on output</th>
<th>Final reduction in gear box</th>
<th>By calculation (progressive)</th>
</tr>
</thead>
<tbody>
<tr>
<td>COUNTER A</td>
<td>37</td>
<td>18</td>
<td>2.055600</td>
<td></td>
</tr>
<tr>
<td>COUNTER B</td>
<td>31</td>
<td>24</td>
<td>1.291700</td>
<td></td>
</tr>
<tr>
<td>1st GEAR A</td>
<td>18</td>
<td>37</td>
<td>4.225309</td>
<td>4.1000</td>
</tr>
<tr>
<td>2nd GEAR B</td>
<td>18</td>
<td>37</td>
<td>2.655093</td>
<td>2.5537</td>
</tr>
<tr>
<td>3rd GEAR A</td>
<td>28</td>
<td>27</td>
<td>1.982143</td>
<td>1.7490</td>
</tr>
<tr>
<td>4th GEAR B</td>
<td>28</td>
<td>27</td>
<td>1.245536</td>
<td>1.3186</td>
</tr>
<tr>
<td>5th GEAR A</td>
<td>35</td>
<td>20</td>
<td>1.174608</td>
<td>1.0930</td>
</tr>
<tr>
<td>7th GEAR B</td>
<td>35</td>
<td>20</td>
<td>0.738095</td>
<td>0.7700</td>
</tr>
<tr>
<td>6th GEAR</td>
<td>Direct Drive</td>
<td></td>
<td>1.00</td>
<td>1.0000</td>
</tr>
<tr>
<td>1st REVERSE GEAR</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2nd REVERSE GEAR</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3rd REVERSE GEAR</td>
<td></td>
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</tr>
</tbody>
</table>
VI. Stress Analysis

After designing the 7 speed MT a theoretical analysis of gears was performed to study the stress patterns developed & compare it with theoretical calculations. It is widely agreed that both experimental and theoretical analysis of spur gears can be performed using a 2D approach. This approach presumes that the load is uniformly distributed along the tooth face (on a line parallel with shaft axis) and, in the most cases, only a single tooth is taking into account. Yet many 3D analysis of the spur gears have been accomplished in order to investigate if the development of fillet stress in the axis shaft direction can be ignored or what is the variation of this stress [9-10]. The stress analysis is further subdivided in two parts: Bending stress analysis & contact analysis.

General hypothesis followed for stress analysis is as follows [11]

1. The geometry of the tooth and the gear is obtained using mathematical formulations in CATIA V5R10. The tooth profile is involute.
2. The geometry of the gear includes rim geometry with a solid geometry and defined ratio parameters as per standards.
3. The number of the teeth taking into account is three, in order to simulate the single and double pair teeth in contact, over the whole roll angle.
4. The complete gear is also selected for bending analysis.
5. The load applied is modeled both for torque model and force loaded at the profile points.

A. Bending Stress

1. Segmental Analysis

A finite element model with a segment of three teeth is considered for analysis. The boundary conditions used are similar to the one proposed by Von Eiff Et Al. [9]. The geometry was created in CATIA V5R10 & was imported to ANSYS workbench for stress analysis under static load. The 1st gear of the gear box was selected for static analysis as it transmits the highest torque so the bending moment is highest in 1st gear. A standard spur gear with module 3mm, number of teeth 18, pressure angle 20 degrees, is considered for analysis. The element selected for bending stress analysis of the gear teeth is SOLID 186. The gear material having modulus of elasticity equal to $2.1 \times 10^5$ N/mm² and poison’s ratio equal to 0.3 is considered for bending stress analysis. A uniform pressure of 3000 N/mm² was applied on a line along the one of the face of the gear tooth. The bore and the sides of the gear were fixed to make it similar to a cantilever beam. The FEA results of stresses developed are comparable with the theoretical stresses calculated using the Lewis equation [4].

2. Analysis of Complete Gear

A finite element model of 1st gear of the gear box with all teeth is considered for the analysis. The boundary conditions used are similar to the one proposed by Von Eiff et al., [9]. The geometry was created in CATIA V5R10 & was imported to ANSYS workbench for stress analysis under static load.

A standard spur gear with module 3mm, number of teeth 18, pressure angle 20 degrees, is considered for analysis. The FEA results of stresses developed are comparable with the stress calculated using the Lewis equation [4].

B. Analysis Results

By comparing fig. 6, [10], with the results obtained by our analysis in fig. 7 & fig. 8, we can see that the stress distribution obtained by us around the tooth root is similar to the one obtained by researchers.
VII. Contact Analysis

Gear tooth analysis is performed to achieve high load capacity with reduced weight of gears & reduce rotation delay between driver & driven caused due to elastic deformation, manufacturing defects & assembly misalignment. This leads to very serious tooth impact causing noise, vibration & development of local stresses on or beneath the surface of contact which are major causes of gear failure. Thus a contact analysis of gear teeth was performed in ANSYS so as to ensure smooth reliable power transmission & estimate the surface hardness for gears used in 7 speed MT. ANSYS supports three contact models: node-to-node, node-to-surface, and surface-to-surface. An eight noded Iso-parametric plane stress quadratic quadrilateral element was used to build the finite element models of these two teeth. The type of contact was node to surface. The target surface was chosen in the gear tooth and meshed by 2D target element while the contact surface was chosen in the pinion tooth with 2D contact element. In ANSYS software, these two elements are (Target169) and (Contat175) respectively [12]. By comparing fig. 10 [13], with the plots obtained by our analysis in fig. 9, we can see that the stress distribution obtained by us around the point of contact is similar to the one obtained by researchers. Based on surface fatigue strength calculations the estimated hardness was found to be 475BHN.

VIII. Conclusion

This new design for the 7 speed MT offers benefit from all side ranging from better overall efficiency of the vehicle by higher approximation to the traction hyperbola, improvement of fuel economy as well as reduction of material cost by eliminating 2 gear pairs and finally the space constraints. The detailed 3D parametric model was developed in CATIA so that the design modifications and creation of a family of parts can be performed in remarkably quick time thereby avoiding redrawing as required by traditional CAD. The geometry created in CATIA V5R10 & was imported to ANSYS workbench for performing stress analysis & results were comparable with theoretical calculations. This design can also be used in case of high speed gear boxes for commercial vehicles. The same concept can be developed by offering 4 reductions at counter & 4 reductions after it giving us a compact 16 speed MT having 16 forward gears & 4 reverse gears with just 8 gears pairs for commercial vehicles.

IX. Abbreviations

MT Manual transmission
LMV Light motor vehicle
SUV Sports utility vehicle
AT Automatic transmission
AMT Automated manual transmission
DCT Dual clutch transmission
Z No of teeth
F Wheel resistance
FR Rolling friction coefficient
Fst Gradient resistance
FL Air resistance
Fa Acceleration resistance
F Force
Ia Reduction in gear box
Ig Overall reduction
T Torque transmitted
ηtotal Efficiency of gear box
r_dyn Dynamic radius
m Module
Rk Outer circle radius
Rr Root circle radius
Rb Base circle radius
R Pitch circle radius
A Pressure angle
References


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