Saimaa University of Applied Sciences
Faculty of Lappeenranta
Degree Programme in
Mechanical Engineering and Production Technology

Ma Mingchuan

Automobile Transmission Design

Thesis 2014

Abstract

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The topic of this project is a design of transmissions for a medium-size truck. The adoption of suitable designs and layout enables the effective utilization of engine to improve the truck's motility and economy.

The design was referred to parameters of Beijing Foton truck and relevant books. Via a comprehensive demonstration, the data was collected from various components of the transmission which were modeled by Solidworks.

The transmission is a five-speed transmission, including five forward gears and one reverse gear, applying advanced monolithic structure of the intermediate shaft and the shift lock ring-type synchronizer. The gearbox possesses a compact structure, a small size, high transmission efficiency, and a larger ratio range, with good economy and dynamic performance.

Keywords: five-speed, intermediate shaft, synchronizer.

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1. Introduction

The role of transmission:

Transmission changes the speed of the engine and exports different rotational speed. At low rotational speed, high torque can be obtained. High speed has better efficiency but the torque is low. So, when you start a machine you need to start at low speed and after running then change high rotational speed maintaining a high efficiency.

Specifically, in order to ensure good transmission performance, it should meet the following requirements.

1) Choose the correct gear shift number and transmission gear ratios, and make the optimal matching of engine parameters to ensure that the car has good power and economy.
2) Set neutral to ensure the car engine and the transmission can separate for a long time.
3) Set reverse, so that the car can travel backwards.
4) Set the power output apparatus.
5) Shift quickly, labor-saving, easily.
6) Reliable. During driving, the transmission cannot out-of-mesh, random mesh.
7) Transmission should have a high efficiency.
8) Transmission should be simple, smooth, no noise.

2. Demonstration program

2.1 Select transmission type

There are many types of transmissions.

By the number of forward gears, transmissions can be divided into three, four, five-speed and the multi-speed transmission, five speed transmission was chosen.

According to the shaft of different form, transmission can be divided into fixed shaft and rotating shaft (often with planetary gear transmission) two kinds of transmissions. The main characteristic of fixed axis transmission is easy to realize automatic shift widely used. Two shaft transmission for front engine front-wheel drive. The shaft of rotation type is mainly used for hydraulic mechanical transmission. Fixed shaft type was chosen.

According to the number of different shafts, the transmission can be divided into two-shaft transmission, layshaft transmission and multiple shafts transmission.

2.1.1 Two-shaft transmission

If the transmission ratio is small, we often choose two shaft types. It has the following characteristics.

1. The gearbox output shaft and the main driving gear speed reducer make it an organic whole.
2. When the engine is longitudinally mounted, the main reducer can use spiral bevel gear or hypoid gear, when the engine is held horizontally, it is tapped with a cylindrical gear, thereby simplifying the manufacturing process.

3. The other gears adopt constant mesh gear transmission, except the reverse gear drive.
4. Most of the synchronizer of gears at the end of the output shaft.
5. If low gear transmission is relatively large, the size of the structure increases, it no longer has the above advantages, it can only drive relatively small conditions before using this program.
6. Two-shaft transmission does not have directly gears, therefore, working at high speed, gears and bearings are bearing, gear noise, and easy to be damaged.

2.1.2 Layshaft transmission

From a structural view, there are three shafts: the first and the second shaft are in the same line, and they are directly shifted. When using direct shift transmission, gears, bearings and the second shaft are not loading bearing. The engine torque through the first and second shaft have direct output, and the transmission has high transmission efficiency - up to 90 %. This means that it has less wear and long service life, thus, noise is also smaller. Because the efficiency of direct gear is higher than the other forward gears, it increases the life of the transmission. When the transmission power is transmitted through the first shaft (the intermediate gear shaft and the second shaft), so the distance between them is not too long, but there is still a large transmission gear ratio. High gear uses constant mesh transmission, whereas low speed gear cannot use constant mesh gear. Most transmission schemes except first speed gear shift mechanism are used in synchronization or clutch shift. Few first speed gears are also used to synchronizer type or clutch type shift. Intermediate shaft transmission is widely used in various types of rear-drive cars. That is the reason why the structure is adopted to the design.

Twin intermediate shaft transmissions or multiple intermediate shaft types are the mostly used in heavy vehicles. As it does not match with the design, it is not examined further. (Yu, 2009, p.122)

2.2 Gear selection

There are two transmission gears: spur gears and helical gears. Spur gear is used for sliding. It is applied in reverse gear and the first gear specifically. The structure is simple and easy to manufacture, but when it is shifting, the root of the gear tooth is prone to bring about noise. That intensifies the wear of gears and lowers the life expectancy. And due to the noise, it easily leads into driver fatigue. Helical gears offer smooth transmission, lower noise, lower wear and longer life. The drawbacks are the axial force generated when working and the structure is complex. This drawback can be balanced when making the calculation of the shaft.

By comparing the advantages and disadvantages of the two forms of the gears, reverse and the first speed gear use straight gear, which is considering following factors: the reverse gear and the first speed gear are low usage. Measuring the economy and practicality of the gear structure, the rest of the gears are helical gears, that depends on helical gear has smooth transmission and lower noise. (Yu, 2009, p.126)

2.3 Shift gear structure selection

Transmission shift introduces three kinds of forms: straight teeth sliding, gear meshing and synchronizer shifting.

2.3.1: Straight teeth sliding gear

This form is easy to manufacture, has a simple structure, but includes various disadvantages. It is prone to impacts due to the shift, leading to fast wearing, lower service life and higher noise. Therefore, it reduces the driving safety and comfort of a car. And technical requirements of the driver are too high, which can influence the driving of the car.

2.3.2: Gear meshing

Using a meshing shift increases the number of gear teeth to receive the impact load during the gear shift. In gear meshing, the gear tooth is not involved in the shift, so it is allowing longer life cycle. However, it cannot eliminate shift impact. Therefore, the car safety and ride comfort are affected by a certain amount, and the technical requirements of the driver are too high. In addition, due to adding the mesh and mesh gear, often makes a big moment of inertia of rotating parts of the transmission, so this way of shifting generally is applied to some place without high demand and heavy lorry.

2.3.3 Synchronizer

This shift form can eliminate shift shock and the rapid shift. And the manipulation is light. Also, the driver's request is not high. Eliminating noise and shift shock improves the car ride safety, acceleration, comfort and economy. So, modern cars are generally used in this form but due to its complex structure, manufacturing needs high accuracy requirement. The manufacturing of synchronizer is difficult and synchronous ring is easy to damage but it is still widely used. This design adopts this shifting form. (Yu, 2009, p.130)

2.4 The form of reverse selection

In order to achieve the reverse drive easily. Cars are equipped with a reverse idle gear between the layshaft and output shaft. This program structure is simple and easy to produce.

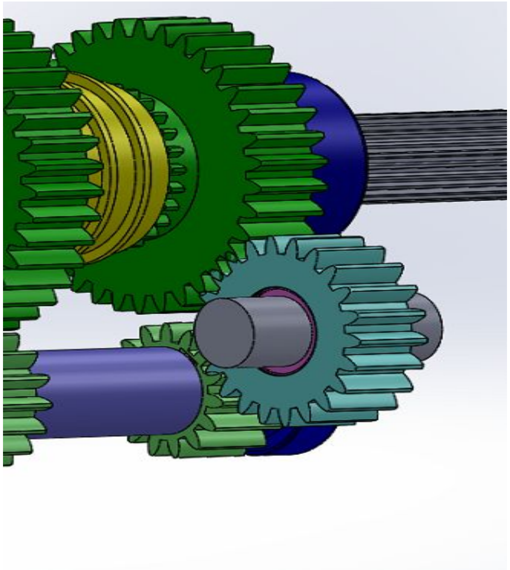

Figure 2.1 Reverse gears (Drawing by the Solidworks).

2.5 Transmission structure

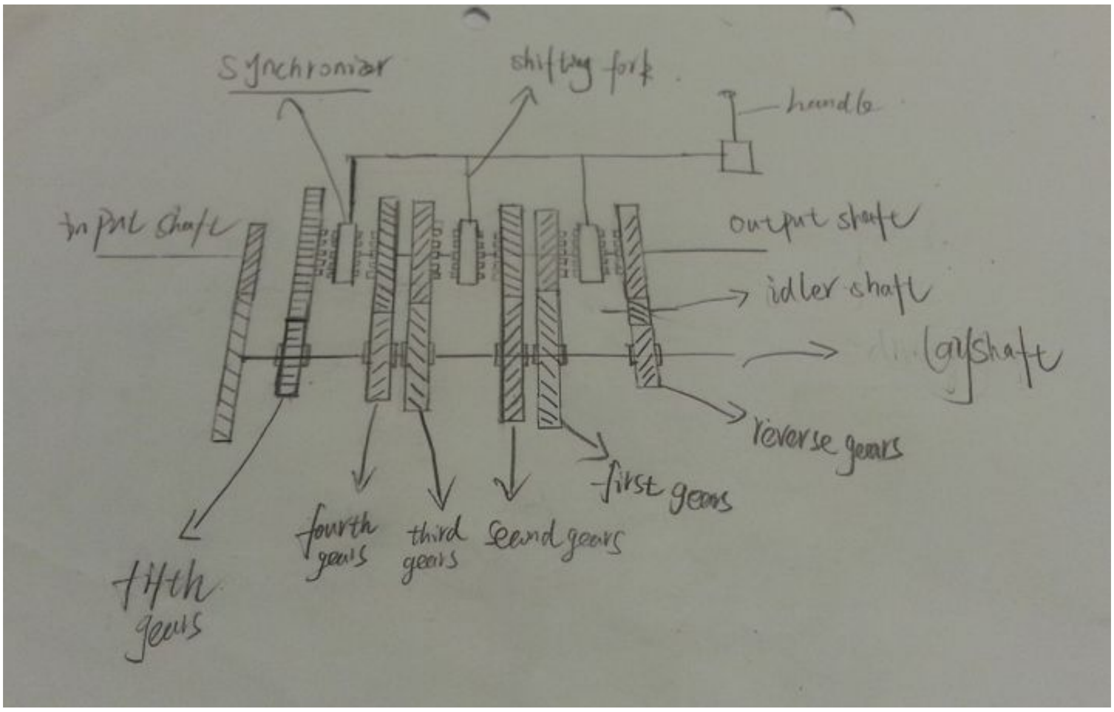

Figure 2.2 Transmission structure (Drawing by hand).

2.6 Synchronizer selection

Synchronizer typically has normal pressure type, inertia type and self-servo type. Among them, the inertia type synchronizer is more commonly used.

2.6.1 Normal pressure synchronizer

The structure of the synchronization structure is simple. Because the engagement sleeve axial resistance is caused by the spring pressure, the pressure of limited size is not guaranteed. So this form of synchronizer has been applied only on heavy vehicles. The transmission does not use this synchronizer.

2.6.2 Self-servo synchronizer

The synchronizer is also known as Boshe Er synchronizer. It can ensure that only in the synchronous state shift, as long as there is the angular velocity difference between clutch and gears, the spring of synchronizer supports force to stop synchronizer ring shrinking, so it prevents movement of meshing sets. Only when the angular speed difference is zero, the spring unloads the load, due to losing the resistance of synchronous ring, shifting process can be achieved. Boshe Er synchronizer has high friction torque, simple structure, reliable operation and short axial dimension. They make transmission in trucks very convenient.

2.6.3 Inertia type synchronizer

This form of synchronizer is the same as the normal pressure type synchronizer. It depends on friction effect of synchronization. But it can ensure the joint sets and joint spline gear ring keep distance before the synchronization, and avoid the shock and noise between the teeth. From the structural term, inertia type synchronizer has lock pin type, lock ring type, slider type, chip type and cone type. Although their structures are different, they have the same friction elements, locking elements and elastic elements.

1) The essence of a sliding block type synchronizer is the lock ring synchronizer. It works reliably. It has durable parts but because of the

restrictions on the structure arrangement, the bending moment capacity is not big, and tooth surface wear large. So, to be on the safe side, the car does not use this kind of synchronizer.

2) The locking surface of lock ring synchronizer is on the conical surface of synchronous cone ring. That eliminates the teeth of the synchronized cone ring, thus it makes the shaft size is smaller. Considering the rationality of the structural layout, compactness and cone friction torque factors. It is applied for cars and medium trucks transmissions. So this transmission adopts the lock ring synchronizer.
3) The advantage of locking pin type synchronizer is that it has small number of parts, average friction cone radius is larger and torque capacity is improved. The disadvantage is that the axial size is big. So, it is usually for heavy auto transmission. The design does not use this form of synchronizer.
4) The locking surface of cone type synchronizer is still on the synchronization ring joint tooth, but inserting two auxiliary synchronizations between the two cone surfaces. Since the effective area of the cone friction surfaces is exponentially increasing, the synchronizing torque is increased accordingly, thus having a large capacity and a low torque load. This will not only improve the synchronization performance, increase reliability, but also shift power is greatly reduced. If the shift force remains unchanged, the synchronization time can be shortened. Multi-cone synchronizers are used for heavy vehicles. (Liu, 1996, p.175)

Lock ring synchronizer

Figure 2.3 The real lock ring synchronizer (Automotive Transmission, 2013, p.1).

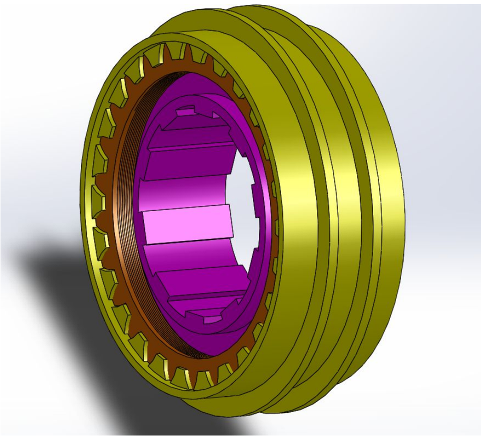

Figure 2.4 The simulative synchronizer (Drawing by Solidworks).

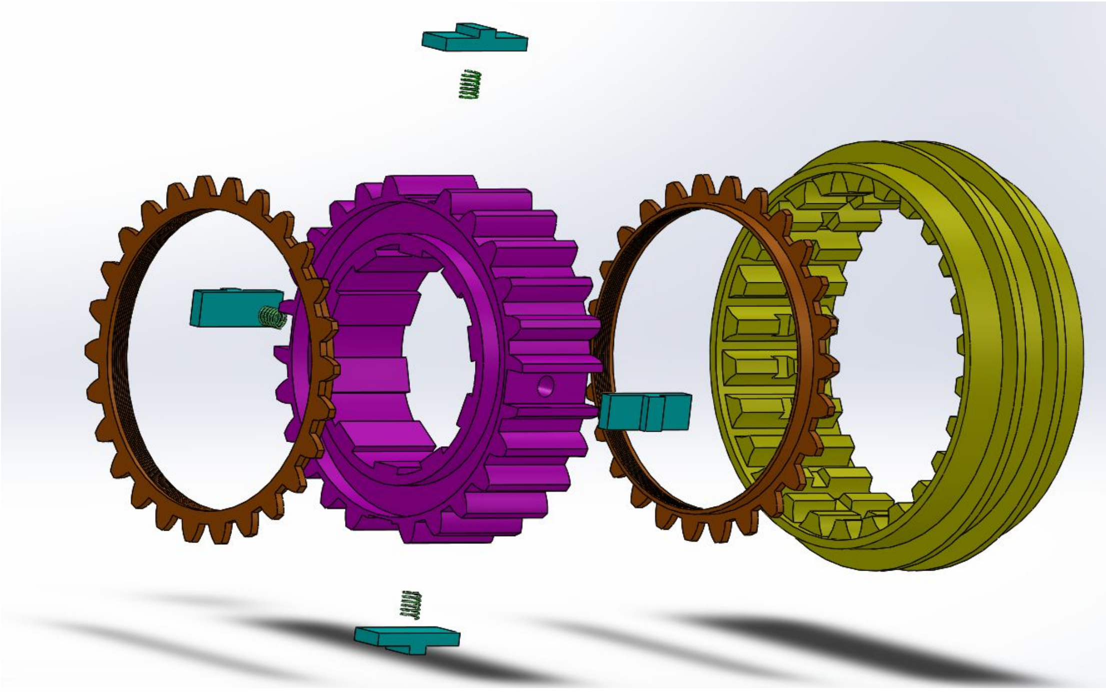

Figure 2.5 The exploded view (Drawing by Solidworks).

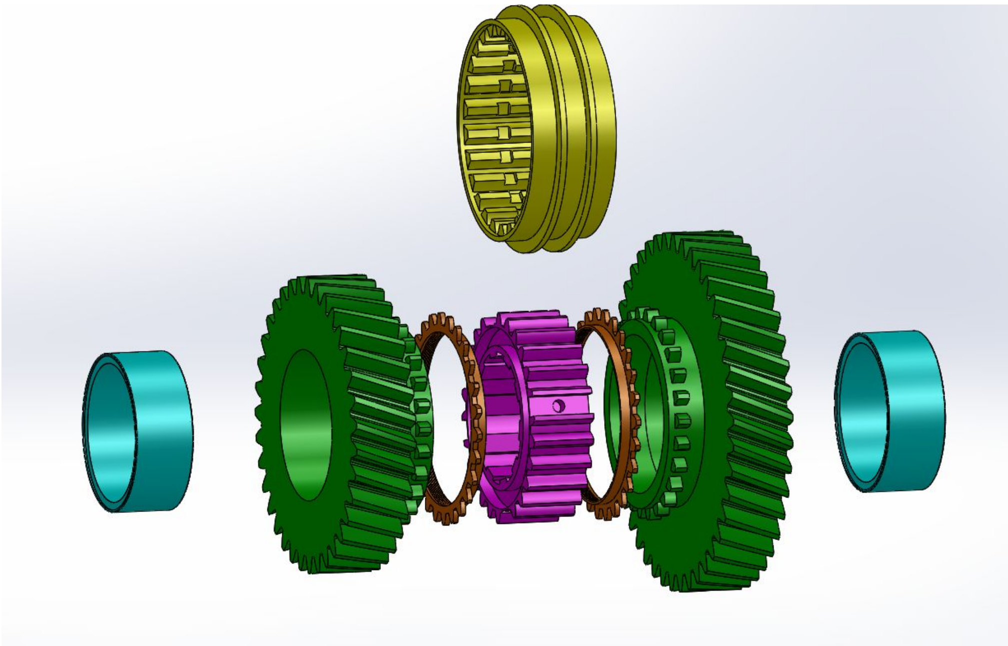

Figure 2.6 The whole view (Drawing by Solidworks).

2.7 Bearing Selection

Transmission requirements increase the ratio of the transmitted power and quality and required work bearing reliability, large capacity, and long service life. Therefore, the selection of the bearing is more important.

Transmission bearing often uses cylindrical roller bearing, ball bearing, needle roller bearing, tapered roller bearing, sliding sleeve, etc. The selection of bearing is restricted by the structure and the load characteristics.

Automotive transmission should have compact structure and small size. Using the big size bearing is limited by the structures, and it often has difficulty in the arrangement. The output shaft fixes on the shell. Because of the axial force, it adopts cylindrical roller bearing. The input shaft fix on the flywheel cavity, it uses angular contact ball bearing to bear radial and axial forces. Layshaft gear works with great axial force, in principle, every side can bear the axial force. If the arrangement of the front section is difficult, the behind section must bear the whole axial force. This design adopts the tapered roller bearing. Due to the limitation of shaft diameter, the gears of the output shaft are fixed on the shaft by the needle roller bearing. (Liu, 1996, p.190)

2.8 The transmission shaft and parts localization

The gears and bearings of output shaft are axial positioned by the snap ring, thrust ring. The axial position of layshaft is through the snap ring and bearing adjustment shim. Synchronizers are located by shaft shoulder and shaft collar. (Liu, 1996, p.182)

2.9 Gearbox shell

There are two forms of gearbox shell: split and integral. The integral is composed of one shell and top cover. The advantage is that the concentricity of the transmission bearing hole is easily to guarantee. The shell is made of cast iron, and the top cover is made of aluminum alloy. The split shell machining precision demand is high, and it is mostly the aluminum alloy die casting. It is mainly used in cars and light vehicle.

The gearbox shell size is as small as possible, at the same time, the weight is smaller and the stiffness is big enough. That ensures the shaft and bearing are not skewed. Transmission gear under transverse section size should be able to guarantee arrangement, and the design should also be noticed that there a 5 - 8 mm gap between the wall of shell and the rotary gear addendum, otherwise it could increase the hydraulic resistance of lubricating oil, produce noise and overheat. There should be a 15 mm gap between the bottom of the shell and the rotary gear addendum.

In order to strengthen the stiffness, the gearbox shell should be set up ribs on the shell. The direction of the ribs is affected by the force of the bearing area. Gearbox shell should not have the large plane that hinders the shell to absorb vibration and noise.

In order to refuel and drain, putting the fuel inlet and the grease outlet on the transmission shell. In order to ensure the internal atmospheric pressure of the transmission, the vent needs to be plugged on the top of the shell. The shell has output hole and the reverse gear inspection hole. In order to reduce the weight of the transmission, it adopts the die casting aluminum alloy, and the wall thickness is 3.5 - 4 mm. When using cast iron casting, it is 5 - 6 mm wall thickness. Transmission thickness increases, and the strength and rigidity of the housing can be improved, but it will increase the weight of the transmission, and also increase the consumption of materials. In this design, the wall thickness of the shell is 6 mm. (Liu, 1996, p.168)

2.10 Automobile control mechanism selection

Transmission control mechanism should be able to ensure that the driver can make accurate and reliable transmission linked into any gear. And it can always retreat to the neutral state. General control mechanism is composed of the shift lever, fork shaft and security devices.

The five-speed transmission generally has three fork shafts. The firsts speed gear and reverse gear share one shaft. The second speed gear and third speed gear share one shaft, and the fourth speed gear and fifth speed gear share one shaft.

In order to ensure the transmission in any case can be an accurate, safe and reliable operation, the following requirements for its control mechanism.

2.10.1 Self-locking device

As shown, there are three pits on the top of the shifting fork shaft, at the pits top there is a steel ball that is pressed by a spring. When the fork is shifting, the steel ball pushes the pits. So it prevents the shifting from removing or changing.

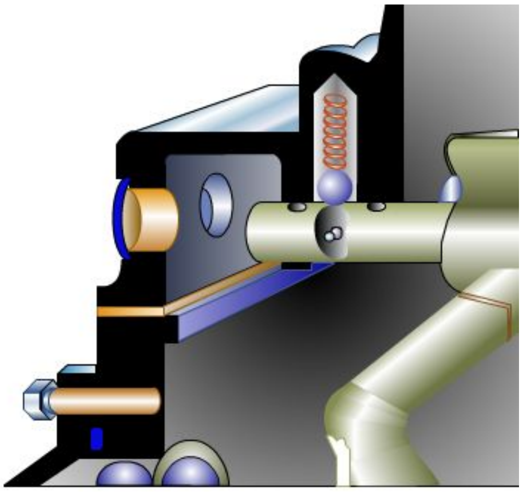

Figure 2.7 Self-locking (www.cxfuwu.com).

2.10.2 Interlocking device

The middle fork shaft moves, and the other two fork shafts are locked by steel ball. That prevents the shifting from putting into two different speed gears and causing damaged. This is the interlock function.

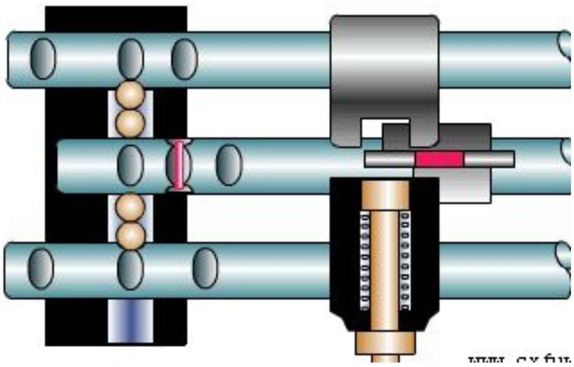

Figure 2.8 Interlocking (www.cxfuwu.com).

2.10.3 The reverse gear lock

When the bottom of the shift lever (red rectangle) is moving to reverse fork shaft, the driver must compress the spring, so the bottom can enter the shifting block groove. That prevents the driver from engaging reverse gear by mistake, and it protects the transmission. That is the reverse gear lock. When the reverse gear fork shaft is moving, the other two fork shafts are locked by steel ball. (http://www.cxfuwu.com/html/2007-04/1280.html)

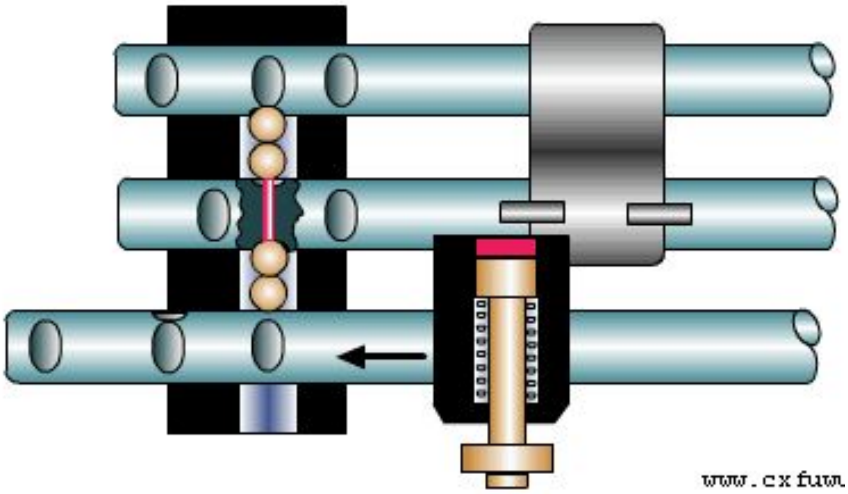

Figure 2.9 Reverse gear lock (www.cxfuwu.com).

3. Engine selection

3.1 The basic parameters of the reference vehicle

Vehicle model:	Beijing Foton 3P78AP4102L medium truck (Figure 3.1)
Weight:	5000 kg
Maximum speed:	85 km/h
Engine position:	Front engine, rear wheel drive
Maximum gradability:	28°
Tire Type:	8.25-16
Vehicle overall length:	5998 mm
Vehicle overall width:	1900 mm
Vehicle overall height:	2600 mm

Figure 3.1 Beijing Foton 3P78AP4102L medium truck (http://rowor.foton.com.cn/sdjg/61_143.html).

3.2 Maximum engine power

$$P_e \max = \eta_r \left(\frac{m_a g f_r}{3600} V_{a\max} + \frac{C_D A}{76140} V_{a\max}^3\right) \tag{3.1}$$

P_e max- Maximum engine power， $Unit$ – Kw

η_r - Drivetrain efficiency, $\eta_r = 0.95$

g - acceleration of gravity, $\mathrm{g} = 9.81\,\mathrm{m/s^2}$

f_r - Rolling friction resistance coefficient, Truck : $f_r = 0.02$

C_D - Air resistance coefficient, $\mathrm{C_D} = 0.9$

$A =$ Orthographic projection area, $A = B * H$

$B -$ Vehicle overall width, $\mathrm{B} = 1.9\,\mathrm{m}$

$H -$ Vehicle overall height, $\mathrm{H} = 2.6\,\mathrm{m}$

$\mathrm{A} = \mathrm{B} * \mathrm{H} = 4.94\,\mathrm{m}^3$

$V_{a\max}$ – Maximum speed, $V_{a\max} = 85\,\mathrm{km/h}$

$\mathrm{m_a}$ - Weight, $\mathrm{m_a} = 5000\,\mathrm{kg}$

$$P_e \max = 0{,}95\left(\frac{5000 * 10 * 0{,}02}{3600} * 85 + \frac{0{,}9 * 4.95}{76140} * 85^3\right)$$

$$= 0.95 * (23.6 + 43.6) = 63.84\,\mathrm{kw}$$

The truck belongs to medium trucks, and medium trucks engine speed is about 1800-2600 r / min. It is based on the above data, and combined with the selected type of truck engine model. The engine model is YZ4102QB (Figure 3.2).

Figure 3.2 YZ4102QB engine
(http://www.360che.com/m30/7735_index.html).

4. The transmission parameter design

4.1 Determining the maximum transmission ratio

4.1.1 From the viewpoint of maximum climbing ability

$$ig_1 \geq \frac{mg * \Psi_{\max} * r}{T_{e\max} * i_0 * \eta t} \quad (4.1)$$

$mg = 5000 * 9.81 = 49050N$

$\Psi_{\max}$ - The maximum friction coefficient of the road, $\Psi_{\max} = (\mathrm{f}\cos\alpha_{\max} + \sin\alpha_{\max})$
f - Rolling friction coefficient, $\mathrm{f} = 0.0076$; $\alpha_{\max}$ - maximum gradability of the automobile

r = tire outer diameter

$T_{e\max}$ - Maximum engine torque

i_0 = final drive ratio

ηt = Automotive driveline efficiency, $\eta t = 0.85$

$$i_0 = 0{,}377 \frac{\gamma_r * \eta_p}{V_{\max} * ig_n} \quad (4.2)$$

r_r = tire outer diameter, $r_r = 0.43\, m$

n_p - Maximum power engine rotational speed

ig_n - Top speed gear transmission ratio $= 1$

$V_{a\max}$ – Maximum speed, $V_{a\max} = 85\ \mathrm{km/h}$

So the final drive ratio is $i_0 = 0{,}377 \frac{\gamma_r * \mathrm{n}_p}{V_{\max} * ig_n} = \frac{0{,}377 * 0{,}43 * 3200}{85 * 1} \approx 6{,}1$

And the maximum transmission ratio is

$$ig_1 \geq \frac{mg*\Psi_{\max}*r}{T_{e\max}*i_0*\eta t} = \frac{49050*0,277*0,43}{225*6,1*0,85} = 5,0$$

4.1.2 From the viewpoint of driving wheel and road surface adhesion conditions

$$ig_1 \leq \frac{G_2*\varphi*\gamma_r}{T_e \max*i_0*\eta t} \quad (4.3)$$

G_2 - When the car loaded with static and pavement drive axle load for the road surface $G_2 = 0.74mg$

φ – Road adhesion coefficient

r_r = tire outer diameter, $r_r = 0.43m$

$T_{e\max}$ - Maximum engine torque

i_0 = final drive ratio

$$ig_1 \leq \frac{G_2*\varphi*\gamma_r}{T_e \max*i_0*\eta t} = \frac{36297*0,5*0,43}{225*6,1*0,85} \approx 6,7$$

So $5 \leq ig_1 \leq 6.7$, we chose $ig_1 = 5.8$. (Liu, 1996, pp.175-176)

4.2 Determining the minimum transmission ratio

Top speed gear transmission ratio generally is 1.

4.3 Determining the transmission ratio

Transmission ratio range: i_{g1}/i_{g5}= 5.8.

Distribution ratio based on a geometric progression:

$$\text{Common ratio} q = \sqrt[4]{\frac{ig1}{ig5}} = 1.55$$

Car primarily uses higher speed gear to run, so the interval ratio between two higher speed gears should be smaller, especially in the highest speed gear and secondary high speed gear. In fact, the transmission ratio is often according to the relationship between the following distributions.

$$\frac{ig1}{ig2} \geq \frac{ig2}{ig3} \geq \frac{ig3}{ig4} \geq \frac{ig4}{ig5} \geq \frac{ig5}{ig6}$$

We already know that i_{g1}=5.8，i_{g5}=1. The q_m is the adjacent gear ratio,

So q_{m1}=1.59, q_{m2}=1.57, q_{m3}=1.55, q_{m4}=1.5.

The ratio of adjacent transmission ratio should be taken into account, it should not be too big (q≤ 1.5 ~ 1.6 or less), in case of the shifting process is difficult.

The finalized i_{g2}=3.65, i_{g3}= 2.32, i_{g4}= 1.5，i_{g5}= 1，i_{g1}= 5.8.

Reverse gear ratio i_R, and reverse transmission ratio smaller than first speed gear transmission ratio, so it selects i_R= 5.4 according to the experience. (Liu, 1996, p.174)

4.4 Center distance

$$A = K_A \sqrt[3]{T_I \max} \qquad (4.4)$$

K_A - Center distance factor， For sedan $K_A = 8.9 \sim 9.3$

For trucks $K_A = 8.6 \sim 9.6$， For multi - shafts transmission $K_A = 9.5 \sim 11$

$T_I\max$ - the output torque of transmission is in first speed gear， $T_I\max = T_e\max * ig_{I*}\eta$

$T_{e\max}$ - Maximum engine torque，N.m

ig_I - first speed gear ratio

ηt = Automotive driveline efficiency， $\eta t = 0.85$

Primaries center distance:

$$A = K_{Ac}\sqrt[3]{T_e max} \qquad (4.5)$$

K_{Ac} - the factor of according to the engine maximum torque to determine center distance *f*or truck $K_{Ac} = 17 \sim 19.5$

So, $A = 18 * \sqrt[3]{225} = 113.4mm \approx 113mm$. (Liu, 1996, p.176)

4.5 Gears parameter selection

The transmission uses a helical gear and spur gear. We need to determine the modulus gear pressure angle, helix angle, tooth width and other parameters.

4.5.1 Gear modulus

Spur gear modulus:

$$m = \sqrt[3]{\frac{2T_1 k_\delta kf}{\pi z k_c y \sigma_w}} \qquad (4.6)$$

T_1 - assumed load

K_δ - stress concentration factor, for spur gear $K_\delta = 1.65$

kf - Coefficient of friction

z - Number of gear teeth

k_c - Tooth width coefficient， $k_c = 4.4 \sim 7.0$

y - Tooth coefficient

σ_w - Gear bending stress

Helical gear modulus:

$$m_n = \sqrt[3]{\frac{2T_1 k_\delta \cos\beta}{\pi z k_c k_e y \delta_w}} \tag{4.7}$$

T_1 - assumed load

K_δ - stress concentration factor, for helical gear $K_\delta = 1.65$

β - Helical gear helix angle

k_c - Tooth width coefficient， $k_c = 7.0 \sim 8.6$

k_e - Overlapping influence coefficient， $k_e = 2$

y - Tooth coefficient

σ_w - Gear bending stress

On the basis of the national standards, helical gear module m_n = 3. Spur gear module m = 4. Synchronizer gear is involute tooth profile, medium-sized truck m = 2 ~ 3.5, so it selects m = 3. (Liu, 1996, p.178)

4.5.2 Pressure angle

When the pressure angle is small, the contact ratio is big, it has smooth transmission and low noise. When the pressure angle is big, it can improve the bending strength of gear contact strength and surface contact strength. For a truck, a larger pressure angle should be chosen. China standard stipulates the pressure angle is 20°. In the same transmission, lower speed gear has larger pressure angle, and higher speed gear has smaller pressure angle. The joint pressure angle between the synchronizer and clutch is 20 °, 25 °, 30 °, it is generally 20 °. (Baike.baidu.com, 2013)

4.5.3 Helix angle

The determination of helix angle is mainly depended on the meshing performance, the influence of the strength and the balance of axial force.

When the helix angle value is increasing, the overlap coefficient of gear meshing is increasing. The transmission has smooth running and low noise. But if the helix angle is too big, it can make the axial force too large enough, adverse to the bearing work and reduced the transmission efficiency. Experiments show that when β > 30, gear bending strength fell sharply, and the intensity of the contact is still rising.

Intermediate shaft helical gear is left-handed, the input and the output shaft helical gears are right-handed. This ensures the axial force of layshaft can be balanced or quits while the transmission is working. So it reduces the intermediate shaft bearing axial load and axial force. For medium trucks, the helical angle value of the transmission is generally 10 ~ 30°. (Baike.baidu.com, 2013)

4.5.4 Tooth width

$$b = K_c * m_n \qquad (4.8)$$

k_c - Tooth width coefficient

m_n = normal module

Spur gear width: b_1 = (4.4~7.0) *4 = 17.6~28

Helical gear width: b_2= (7.0~8.6) *3 = 21~25.8

So based on the data, all teeth are selected 25 mm.

4.5.5 Distribution of each speed gear

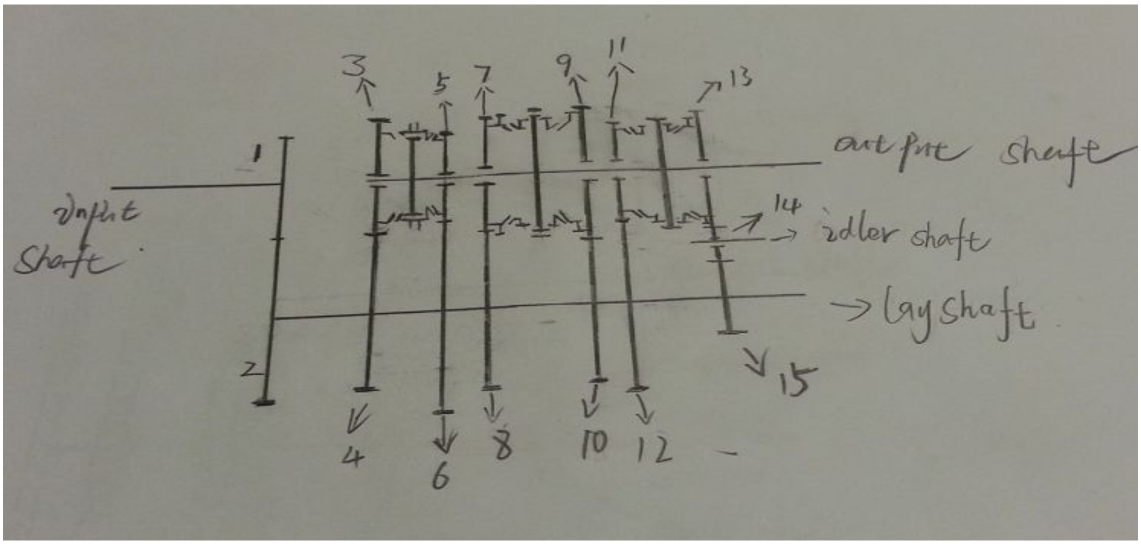

Figure 4.1 Distribution of each speed gear (Drawing by hand).

1. Determine the first speed gear and Constant mesh gear teeth:

$$i_{g1} = \frac{z_2 * z_{11}}{z_1 * z_{12}}$$

The whole number of gear teeth z_h = $\frac{2A}{m}$ = 2×113 / 4= 56.5, so we choose z_h = 56. The smaller first speed spur gear teeth are13 to 17, so we choose z_{10} = 16.

So z_9 = 56 - 16= 40.

（1） $\frac{z_2}{z_1} = i\frac{z_{12}}{z_{11}}$ = 2.32

（2） $\mathrm{A} = \frac{\mathrm{m_n}(z_1 + z_2)}{2\cos\beta}$

Organize（1） and（2）, and get the result

z_1 =21, z_2 =48.

Center distance is adjusted to 111.5 mm.

2. Determine the Second speed gear teeth:

(3) $\frac{z_2}{z_1} = i_{g2} \frac{z_{10}}{z_9}$

(4) $A = \frac{m_n(z_9 + z_{10})}{2\cos\beta}$

Organize (3) and (4) , and get the result z_9= 42, z_{10}= 27.

Using the same method can be calculated:

Third speed gear teeth: z_7= 35, z_8= 35.

Fourth speed gear teeth: z_5= 43, z_6= 27.

Firth speed gear teeth: z_4= 48, z_3= 21.

3. Determine the reverse gear teeth:

The reverse idler gear teeth (Figure 4.2) generally are 21 to 23, we choose z_{14} = 22, the modulus m = 4.

When seeking z_{13} and z_{15}, center distance should be slightly smaller, it is 100mm.

$$Z_\Sigma = \frac{2A}{m} = \frac{2*100}{4} = 50$$

$$\frac{z_2}{z_1} = i_R \frac{z_{13}}{z_{15}}$$

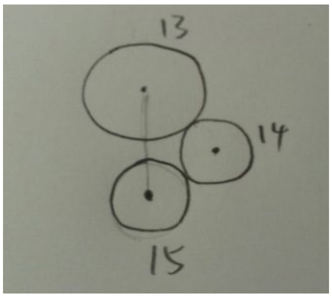

Figure 4.2 Reverse (Drawing by hand).

Get the result z_{13}= 35 , z_{15}= 15.

The distance between the idler shaft and output shaft:

A_1=0.5 * m (z_{13}+z_{14}) = 114 mm.

The distance between the idler shaft and layshaft:

$A_2 = 0.5 * m\ (z_{15}+z_{14}) = 74$ mm.

4.5.6 The gear geometry size calculation

Table 4.1 Truck gear angle (Liu, 1996, p.196).

	Tooth profile	Pressure angle	Helix angle
Truck	common tooth profile	20°	10°—30°

Table 4.2 The coefficients of gear (Liu, 1996, p.199).

Addendum coefficient	f_0	1,0
The radial gap coefficient	C	0,25

I. The formula of helical gear:

Pitch diameter: $d = \dfrac{z * m_n}{\cos\beta}$

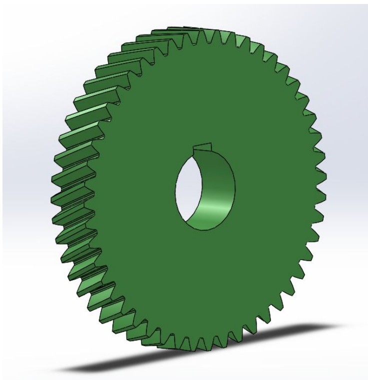

Figure 4.3 Helical gear.

Addendum: $h_a = f_0 m_n$

Dedendum: $h_f = (f_0 + c)m_n$

Tooth height: $h = (2f_0 + c)m_n$

Tip diameter: $d_a = d + 2h_a$

Root diameter: $d_f = d - 2h_f$

z – Teeth Qty

m_n - Helical gear tooth surface modulus

f_0 - Addendum coefficient

β – Helix angle

c – The radial gap coefficient . (Liu, 1996, pp.177-187)

1. Constant mesh gear: z_1= 21, z_2= 48, M_n= 3.

$$\text{Pitch diameter}: d_1 = \frac{z_1 * m_n}{\cos\beta} = \frac{3*21}{\cos 20} = 68, d_2 = \frac{z_2 * m_n}{\cos\beta} = \frac{3*48}{\cos 20} = 155$$

$$\text{Addendum}: h_a = f_0 m_n = 1*3 = 3\text{mm},$$

$$\text{Dedendum}: h_f = (f_0 + c)m_n = 1.25*3 = 3.75$$

$$\text{Tip diameter}: d_{a1} = d_1 + 2h_a = 74, \quad d_{a2} = d_2 + 2h_a = 161$$

$$\text{Root diameter}: d_{f1} = d_1 - 2h_f = 60.5, \quad d_{f2} = d_2 - 2h_f = 147.5$$

2. Second speed gear: z_9= 42, z_{10}= 27. M_n= 3

$$\text{Pitch diameter}: d_9 = \frac{z_9 * m_n}{\cos\beta} = \frac{3*42}{\cos 20} = 136, d_2 = \frac{z_{10} * m_n}{\cos\beta} = \frac{3*27}{\cos 20} = 87$$

$$\text{Addendum}: h_a = f_0 m_n = 1*3 = 3\text{mm}$$

$$\text{Dedendum}: h_f = (f_0 + c)m_n = 1.25*3 = 3.75mm$$

$$\text{Tip diameter}: d_{a9} = d_9 + 2h_a = 142, \quad d_{a10} = d_{10} + 2h_a = 87$$

$$\text{Root diameter}: d_{f9} = d_9 - 2h_f = 128.5, \quad d_{f10} = d_{10} - 2h_f = 79.5$$

3. Third speed gear: z_8= 35, z_7= 35, M_n= 3

$$\text{Pitch diameter}: d_7 = d_8 \frac{z_7 * m_n}{\cos\beta} = \frac{3*35}{\cos 20} = 111.5$$

$$\text{Addendum}: h_a = f_0 m_n = 1*3 = 3\text{mm}$$

$$\text{Dedendum}: h_f = (f_0 + c)m_n = 1.25*3 = 3.75$$

$$\text{Tip diameter}: d_{a7} = d_{a8} = d_7 + 2h_a = 118$$

$$\text{Root diameter}: d_{f7} = d_{f8} = d_7 - 2h_f = 104.5$$

4. Fourth speed gear: z_5= 27, z_6= 43, M_n=3.

$$\text{Pitch diameter}: d_5 = \frac{z_5 * m_n}{\cos\beta} = \frac{3*27}{\cos 20} = 86, d_6 = \frac{z_6 * m_n}{\cos\beta} = \frac{3*43}{\cos 20} = 137$$

$$\text{Addendum}: h_a = f_0 m_n = 1*3 = 3\text{mm}$$

$$\text{Dedendum}: h_f = (f_0 + c)m_n = 1.25*3 = 3.75$$

$$\text{Tip diameter}: d_{a5} = d_5 + 2h_a = 92, \quad d_{a6} = d_6 + 2h_a = 143$$

$$\text{Root diameter}: d_{f5} = d_5 - 2h_f = 78.5, \quad d_{f6} = d_6 - 2h_f = 129.5$$

5. Fifth speed gear: z_3= 21, z_4= 48, M_n=3.

$$\text{Pitch diameter}: d_3 = \frac{z_3 * m_n}{\cos\beta} = \frac{3*21}{\cos 20} = 68, d_4 = \frac{z_4 * m_n}{\cos\beta} = \frac{3*48}{\cos 20} = 155$$

$$\text{Addendum}: h_a = f_0 m_n = 1*3 = 3\text{mm}$$

$$\text{Dedendum}: h_f = (f_0 + c)m_n = 1.25*3 = 3.75$$

$$\text{Tip diameter}: d_{a3} = d_3 + 2h_a = 74, \quad d_{a4} = d_4 + 2h_a = 161$$

$$\text{Root diameter}: d_{f3} = d_3 - 2h_f = 60.5, \quad d_{f4} = d_4 - 2h_f = 147.5$$

II. The formula of spur gear:

Pitch diameter: $d = z * m$

Addendum: $h_a = f_0 m$

Dedendum: $h_f = (f_0 + c)m$

Tooth height: $h = (2f_0 + c)m_n$

Tip diameter: $d_a = d + 2h_a$

Root diameter: $d_f = d - 2h_f$

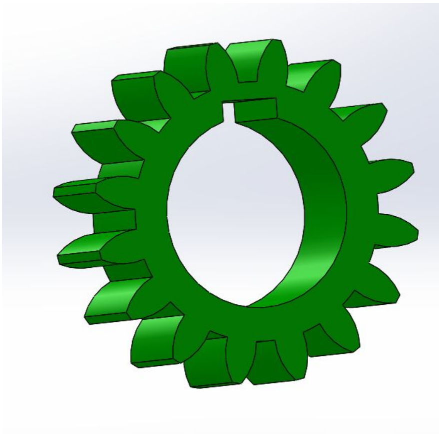

Figure 4.4 Spur gear.

1. The first speed gear: z_{11}= 40, z_{12}= 16, m= 4.

Pitch diameter : $d_{11} = z_{11} * m = 4*40 = 160, d_{12} = z_{12} * m = 4*16 = 64$
Addendum :$_a = f_0 m \;\; = 1*4 = 4\text{mm}$
Dedendum : $h_f = (f_0 + c)m = 1.25*4 = 6$
Tip diameter : $d_{a11} = d_{11} + 2h_a = 167$, $d_{a12} = d_{12} + 2h_a = 72$
Root diameter : $d_{f11} = d_{11} - 2h_f = 147$, $d_{f12} = d_{12} - 2h_f = 52$

2. Reverse gear: z_{13}=35, z_{14}=22, z_{15}=15, m=4.

Pitch diameter : $d_{13} = z_{13} * m = 4*35 = 140, d_{14} = z_{14} * m = 4*22 = 88, d_{15} = z_{15} * m = 4*15 = 60$
Addendum : $h_a = f_0 m \;\; = 1*4 = 4\text{mm}$
Dedendum : $h_f = (f_0 + c)m = 1.25*4 = 6$
Tip diameter : $d_{a13} = d_{13} + 2h_a = 148$, $d_{a14} = d_{14} + 2h_a = 96, d_{a15} = d_{15} + 2h_a = 68$
Root diameter : $d_{f13} = d_{13} - 2h_f = 128$, $d_{f14} = d_{14} - 2h_f = 76, d_{f15} = d_{15} - 2h_f = 48$

(Liu, 1996, pp.177-187)

4.6 The size of the transmission shaft

1. The output shaft and layshaft maximum diameter can be based on the center distance.

   $D = (0.45 \sim 0.60)A$

   D=0.5*113=56.5mm, so we chose the second shaft and the intermediate shaft maximum size is 55mm. And other sizes are depended on the gear size.

2. The length of the shaft is depended on the length of the gears，also need consider the actual installation requirements. (Liu, 1996, pp. 187-190)

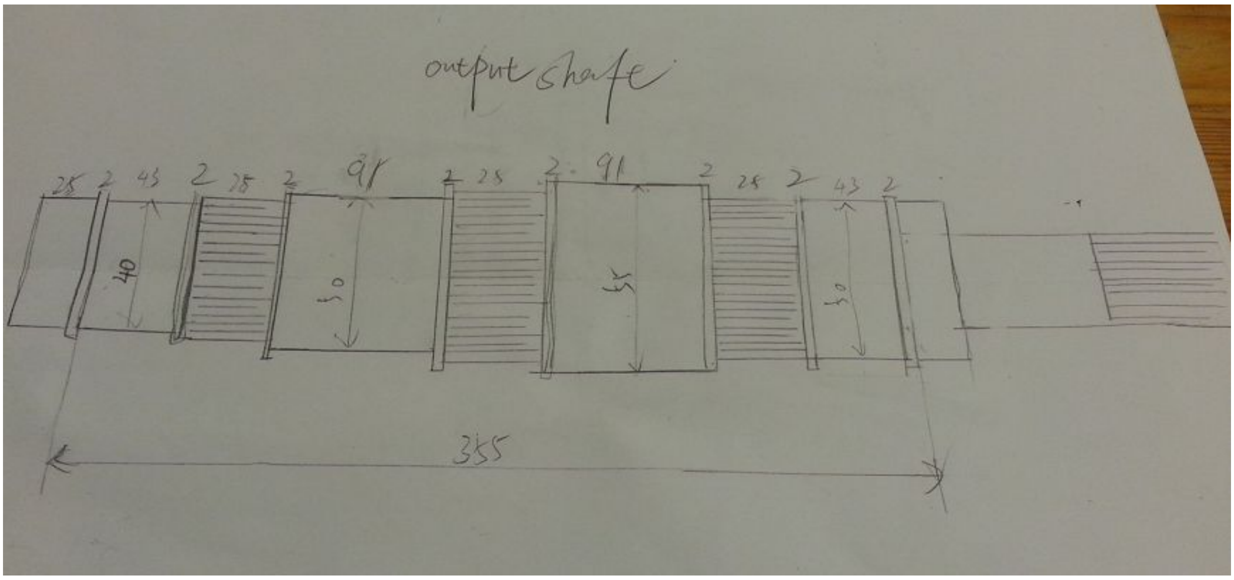

Figure 4.5 Output shaft (Drawing by hand).

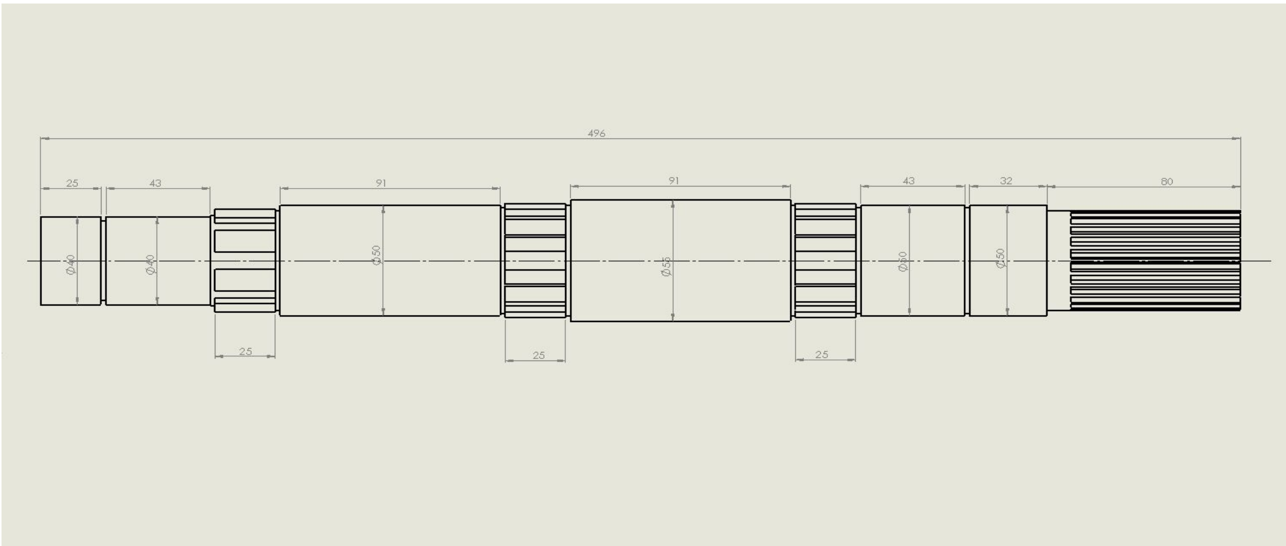

Figure 4.6 Output shaft (Drawing by Solidworks).

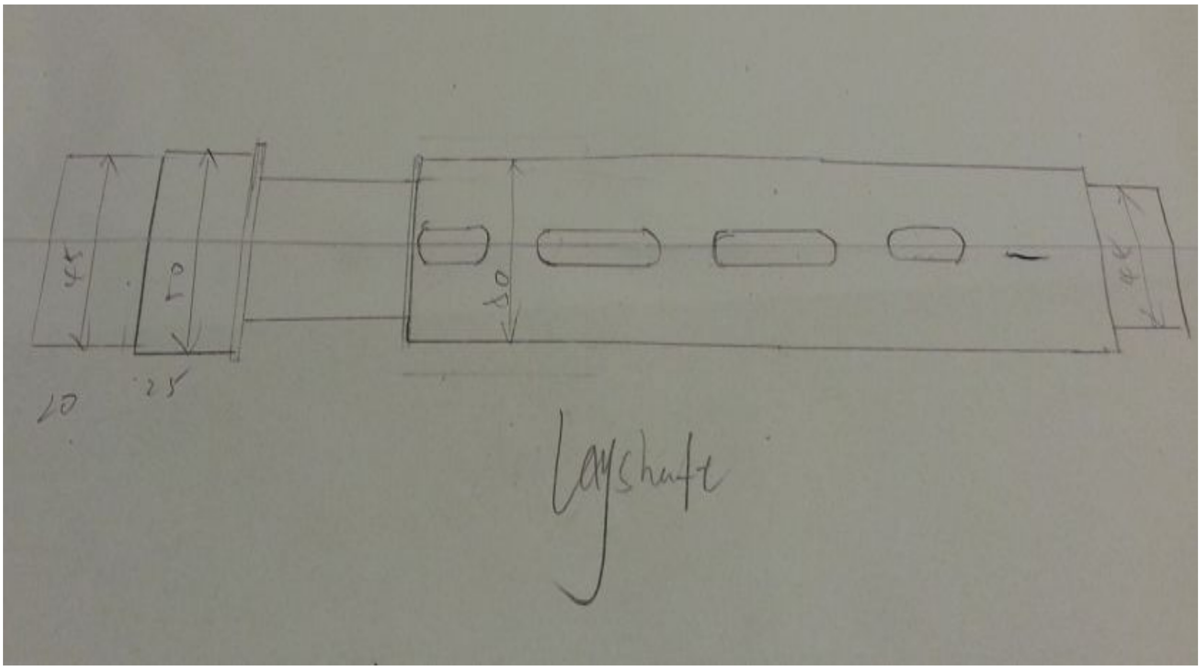

Figure 4.7 Layshaft (Drawing by hand).

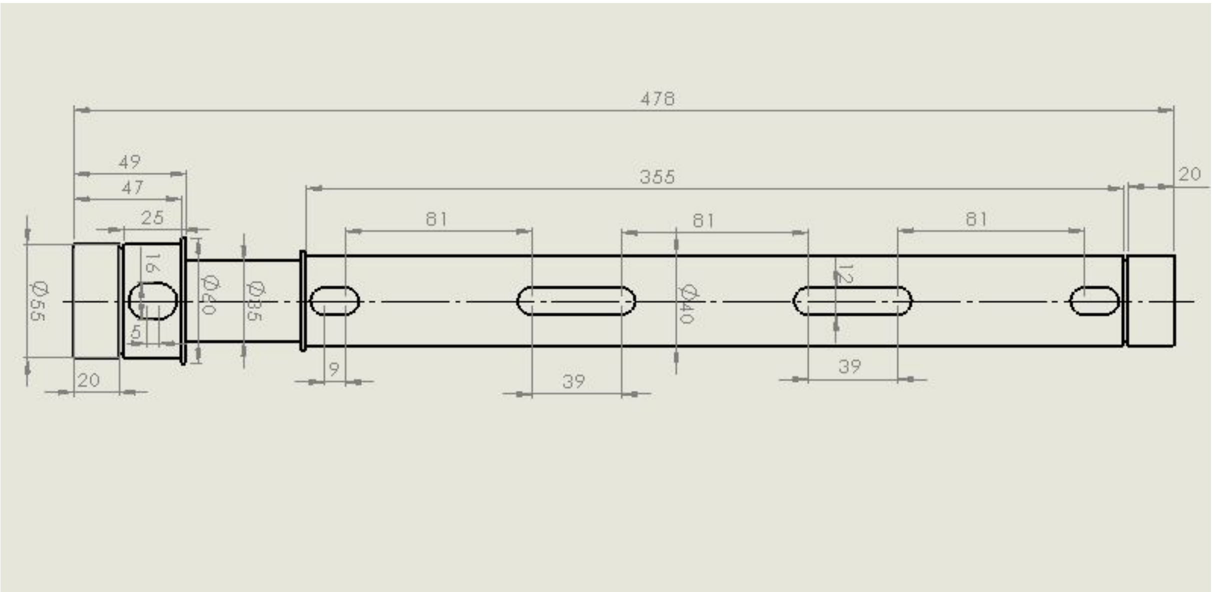

Figure 4.6 Layshaft (Drawing by Solidworks).

5. Summary

The topic of this project is a design of transmissions for a medium-size truck. Transmission is an integral part of the vehicle, with the development of mechanical technology, the design of transmission becomes more and more matured. But for callow students, it is a big challenge for them to use the knowledge to the real practice.

For the design of the gearbox, it has many advantages. The structure of the transmission is simple and easy to produce, so the price and maintenance cost are cheap. The lock ring synchronizer is chosen, so it makes the shifting smooth and no noise. That is also good for the life cycle of the gear. The transmission has five forward gears and one reverse gear and a wide range of transmission ratio that ensures the car can run in different speeds. It still has shortcomings such as the safety factor is not high. So, the designer needs to pay attention to solve the problems.

Despite, that the result of the design is not bad. The successful transmission needs more efforts. It also needs a lot of testings to be produced in the real life.

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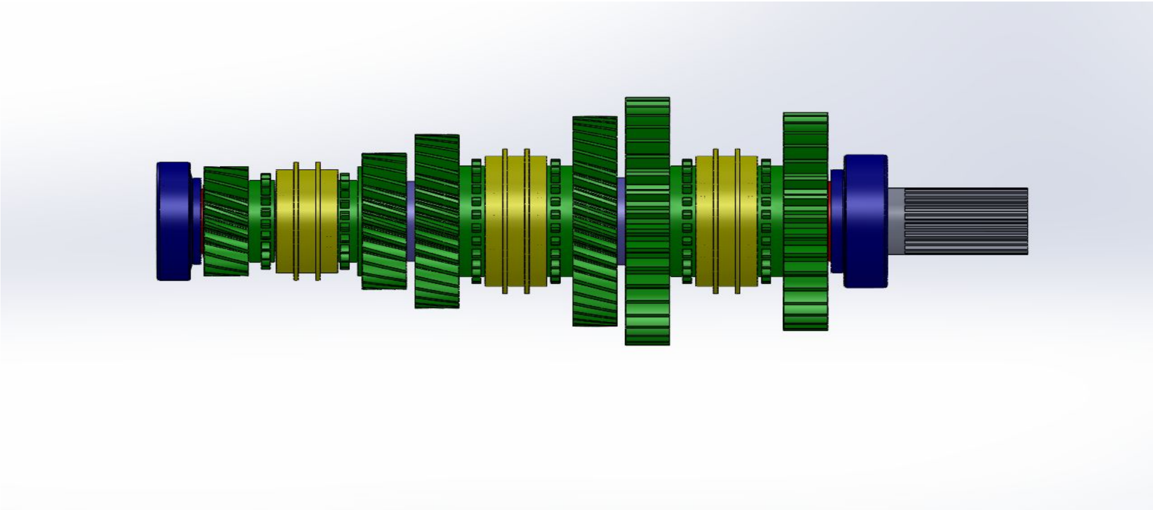
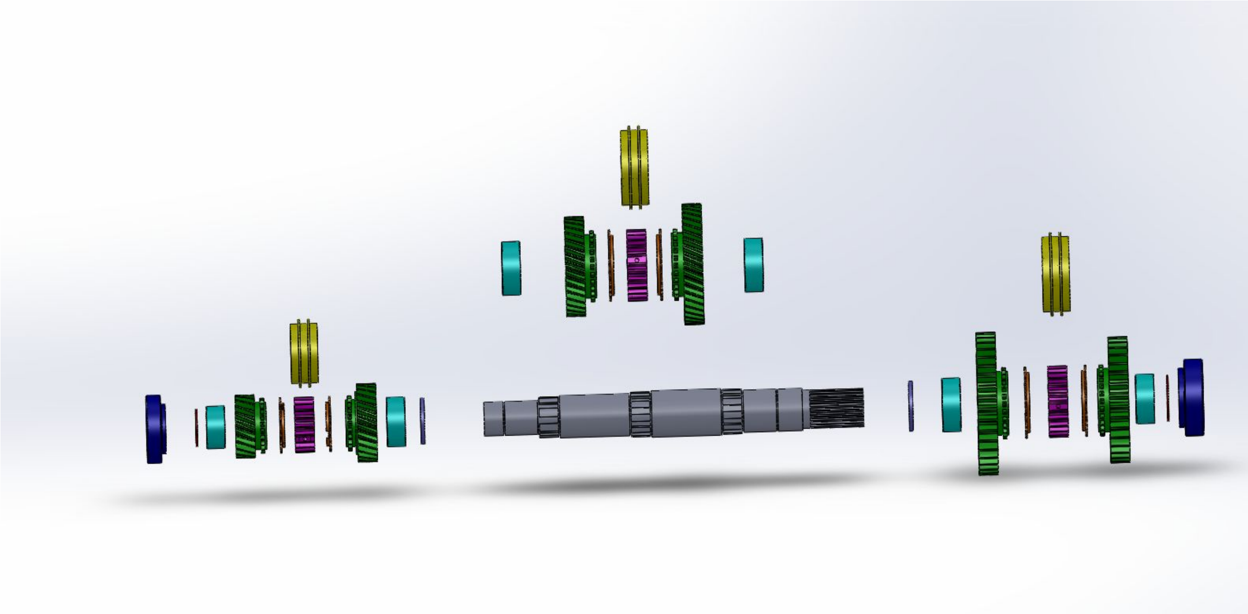
8．Automobile control mechanism selection http://www.cxfuwu.com/html/2007-04/1280.html

Appendices

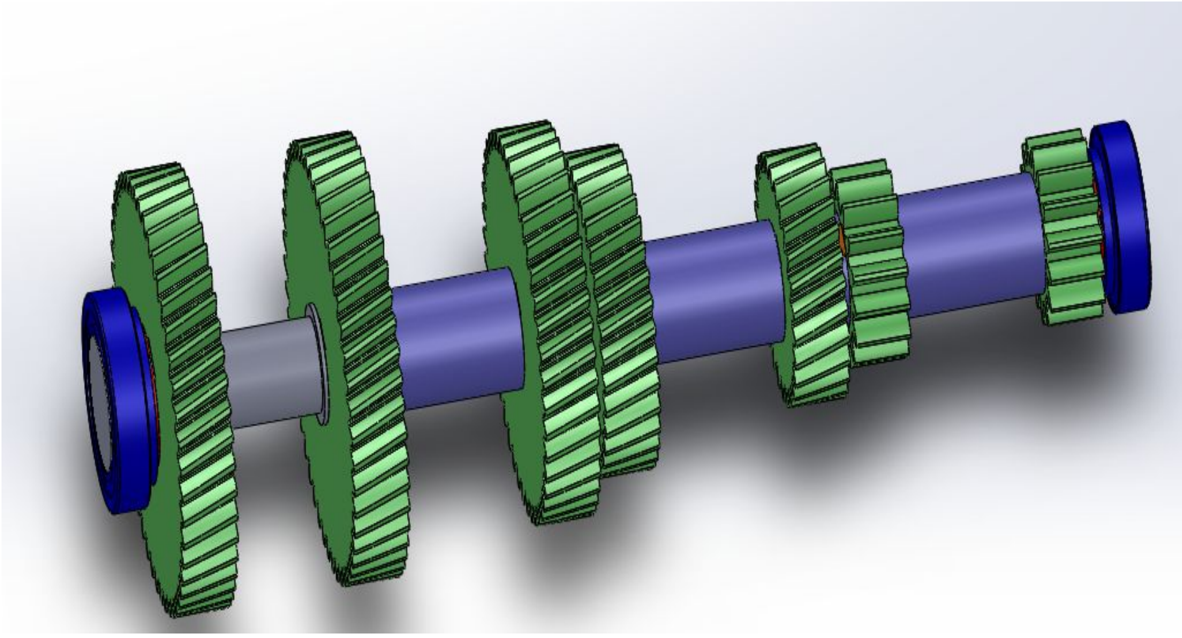
APPENDIX 1. The output shaft (Drawing by Solidworks).



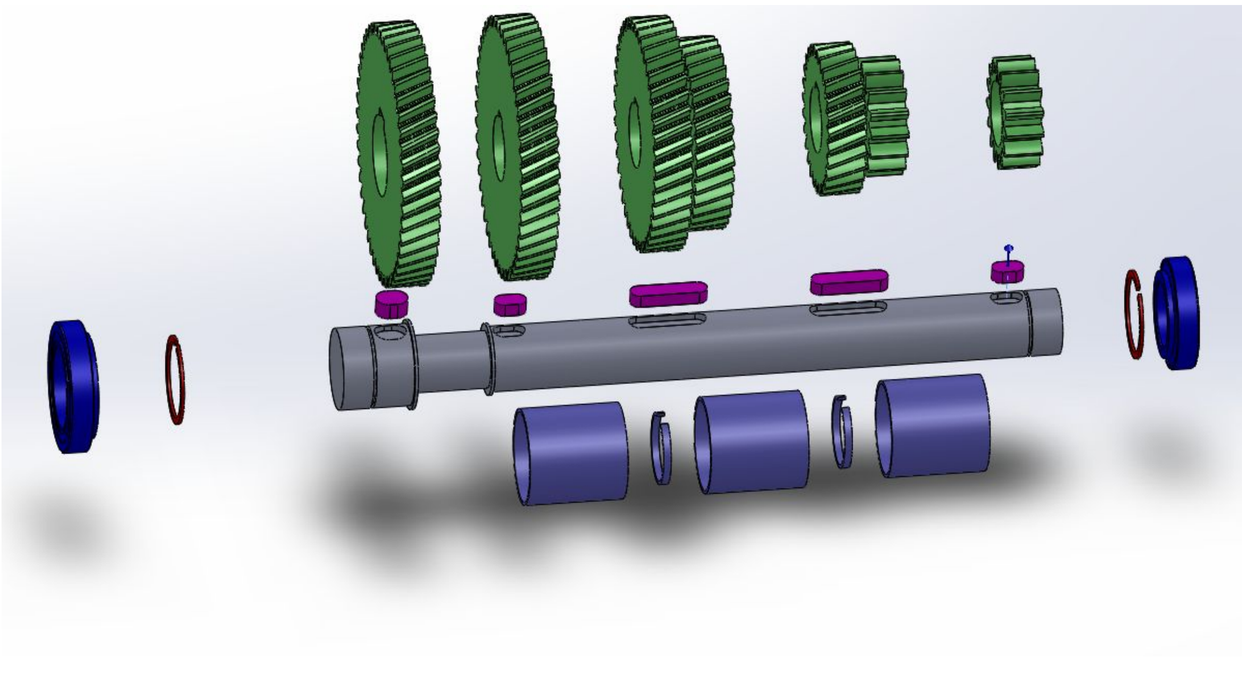
APPENDIX 2. The output shaft exploded view (Drawing by Solidworks).



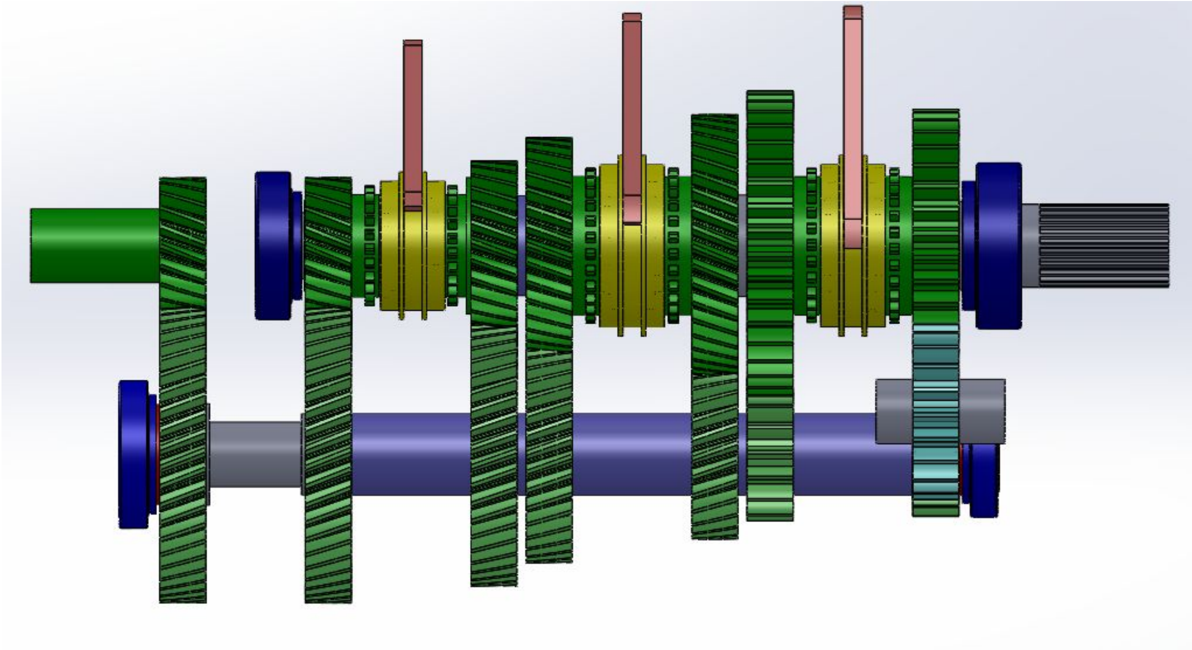
APPENDIX 3. The layshaft (Drawing by Solidworks).



APPENDIX 4. The layshaft exploded view (Drawing by Solidworks).



APPENDIX 5. The whole view (Drawing by Solidworks).



APPENDIX 5. The whole view (Drawing by Solidworks).