ELECTRIC POWERTRAIN FOR LIGHT COMMERCIAL VEHICLE

[1]Bhushan Mehta, [2]Hitesh Raiyani,

[1]Masters of Engineering Student, [2]Assitant Professor
[1]CAD-CAM Engineering,
[1]L J Institute of Engineering & Technology, Ahmedabad, India

Abstract : The major problems in today's mobility is the exponential increase in pollution level, the rise in fuel prices and the fast depletion of fossil fuels. This demands for an alternative source to be used for transportation because transportation as an industry is huge and the entire economy of any country relies on speedy and timely transportation of goods. One of the proposed solution is the advancement of electric vehicles to reduce carbon emission and even save on fuel expenses. This study has been undertaken to develop a powertrain for electric conversion of a light commercial vehicle. Selection methods for gear design, bearings, shafts and materials have been presented. As powertrain is one of the major component in a vehicle for it's performance, optimization has been done through analytical method.

***IndexTerms* - Component,formatting,style,styling,insert.**

I. INTRODUCTION

An electric vehicle uses electric motor instead of the combustion engines used in current vehicles. This motor is powered through a power source comprising of a battery and a battery management system. This power is transmitted to wheels through a powertrain which plays a very important role. Powertrain consists of a gearbox coupled with drive shaft and tyres. The depleting levels of fossil fuels is an alarming sign that the entire mobility industry needs a paradigm shift towards alternative resources. The increasing level of global warming also hints at reducing the carbon emission. Light and heavy commercial vehicles in India contribute to the emission of carbon content and thus electric vehicles in this segment is the best fit.

Electric vehicle is gaining popularity for commercial use due to lower noise, no tail pipe emission and less cost per kilometer. However the mass acceptance of electric vehicles is restricted due to it's less range which depends on the battery pack used. India has a dependency of li-ion. Lifepo4 battery packs on other countries as we only produce lead acid batteries which are 3 times heavy in weight compared to li-ion or lifepo4 batteries. The motors widely used for electric vehicles in light commercial segment is 3 Phase AC induction motor with voltage ranging from 48V to 108V. So it becomes very much essential that all the components of a powertrain other than motor and battery be highly efficient.

Gear design plays an important role for development of a highly efficient powertrain. Gear design is a complex process which involves continuous improvement for accuracy, quieter running and less expensive manufacturing. This work focuses on analytical methods of designing gears with taking input parameters and selection of gears to meet the design requirement.

II. REVIEW OF LITERATURE

On review of research papers and other literature material it was found that there are no electric powertrain available for the light commercial vehicle segment. Thus this research study includes design and development of a gearbox for electric light commercial vehicle. All prior efforts have been made at development of gearbox for passenger cars. All research work has mainly been done in foreign countries.

Research Gap (Problems Identified)

Based on the study of research papers on electric power train, following areas have scope of improvement

1. There is a need for improvement in power train which can carry heavier loads with single motor.
2. There is a scope to reduce the gear whine.
3. Since previous attempts for powertrain optimization were limited to vehicles with less load capacity. Powertrain for Electric LCV have a scope of optimization.
4. The powertrains that were studies lacks the flexibility to be compatible with several vehicle models.
5. The papers shows that in an attempt to optimize the powertrain efficiency, there was a considerable increase in it's cost.

III. EXISTING SYSTEM AND DISADVANTAGES

Currently the light commercial vehicles available have a speed of 25kmph and a Brushless DC motor of 1.5KW. this vehicle has a top loading capacity of 500-1000 kg and comes with a cost between 1.3 to 1.5 lacs. These vehicles are mainly used as auto rickshaws due to it's less speed and short range. It's acceptance as a light commercial vehicle is not viable due to the following disadvantages

DISADVANTAGES

1. Very less speed (25 kmph)
2. Lead acid batteries need 8-10 hours of charging
3. 70-80 kms of mileage per charge.
4. Gard ability is less than 10%

IV. OBJECTIVE OF THIS STUDY

The objective of this research are as follows

1. To study the vehicle dynamic and power train components.
2. To review the currently existing electric power train for light commercial vehicle & identify the gaps thereof.
3. To design the powertrain components like gears, shaft and gearbox casing etc.
4. To develop the prototype of a powertrain.
5. To test the powertrain on a vehicular setup.

V. PROPOSED SYSTEM

The proposed system is to design a powertrain which is highly efficient and can be used for multiple models of light commercial vehicle.

VI. FINALIZING DESIGN REQUIREMENT

This table contains the comparison between the available specifications and the specifications required for the development of electric powertrain.

Table No 1 Input Parameters

Parameter	Current specs in LCV	Required specs
Torque from motor	37Nm (from engine)	112 Nm
Payload	1000 Kg	1000 Kg
Speed	60-80 kmph	60 kmph
Millage	18-21 kmpl	100 km per charge
Gradeability	21%	21%
Gearbox	4 speed synchromesh gearbox	Constant mesh gearbox with gear ratio 7.6258
Driveline torque required	4162 Nm	4162 Nm
Tractive effort	15627 N	15627 N
Max RPM	3200	7200
Wheel Diameter	21 inch	21 inch

VII. SELECTION OF GEARS

Selection of spur or helical gears?

1. Spur gears are noisy and as this is going to be a commercial vehicle, it needs to be free from noise. This is because spur gears have direct face contact with each other
2. Helical gears have only a point contact and thus it reduces a lot of noise.
3. Spur gears are bulky in weight compared to helical gear of same size.

Thus it was decided to opt for helical gears instead of spur gear for the powertrain

VIII. VEHICLE PERFORMANCE CALCULATIONS

Tractive force is the total resistance faced by the vehicle and that is also the total amount of resistance to be overcome by the power train for functioning of vehicle

Tractive force = Rolling resistance + acceleration force + grad ability resistance (Giri)

Rolling resistance = μ*m*g
= 0.3*2000*9.81
= 5886 N

Acceleration force = a*m
= 1.35 * 2000
= 2700 N

Grad ability Resistance = m*g*sin21°
= 2000*9.81*0.3583
= 7031.179

Thus Tractive force = 5886 + 2700 + 7031.179
= 15617.17 N

Torque on wheel = T.F*Radius of wheel
= 15617.17 * 0.2667
= 4165.10 Nm

Torque From motor = 112 Nm

Thus gear reduction required = Torque on wheel/torque from motor
= 4165.1/112
= 37.188

Reduction obtained from differential = 4.88

Thus required gear reduction = total gear reduction / differential reduction
= 37.188/4.88
= 7.6205

Now calculating the minimum number of teeth required on pinion

Zmin = 2/sin2α Where α = pressure angle = 20 degree (Bhandari V. B., 2013)

Zmin = 17.09 is the theoretical value for minimum number of teeths

According to V.B Bhandari, the practical minimum number of teeths can be taken as 14

So Zmin (pinion) = 15

Gear ratio = 7.6205

Zgmin = 15*7.6205

= 114.3

This is not practical as it will increase the diameter of gear.

Thus a two stage reduction gear box needs to be designed

To find the two reduction ratios the method followed is

√ gearbox reduction ratio

= 2.7605 for each reduction;

Thus first reduction R1 = 2.93

Z1 = 15

Z2 = 44

Second reduction ratio R2 = 2.6

Z3 = 20

Z4 = 52

The reason for taking Zp2 = 20 is that this pinion on intermediate shaft will have the maximum torque acting on it and thus it needs to be safer than any other gear. To make it safe in bending, the number of teeth were increased to keep the module in check. This has helped me in reducing the overall weight of the gear.

IX. DESIGN OF HELICAL GEARS FOR FAILURE ANALYSIS

Considerations taken are (Bhandari V. B., 2013)

Module (m) = 4

Normal module (mn) = m*cosΨ where Ψ = helix angle = 20 degree

= 4*cos (20) = 3.758

Minimum face width = b ⩾ (∏mn)/sinΨ = (3.14*3.758 / sin 20)

= b ⩾ 34.50 mm

So b = 35 mm

BEAM STRENGTH OF HELICAL GEARS FOR GEAR PAIR 1

Beam strength Sb = (mn*b*σb*Y)/cosφ (Bhandari V. B., 2013)

Y = Lewis form factor = 0.289 σb= bending strength = yield strength / 3

= 473 / 3 = 157.6 Mpa

Sb = (4*35*157.6*0.289)/cos20

= 6785.72 N/mm2

EFFECTIVE LOAD ON GEAR TOOTH

Effective load = Peff = Cs*Pt + Pd

Cs = service factor = 1.5 Pt = tangential load Pd = dynamic load

Pt = (2* Motor Torque / Diameter of pinion)

= (2*112/63.85)*1000

= 3508.18 N

Pd = (21v (cebcos2Ψ+Pt)cosΨ)/(21v+ √(cebcos2Ψ+Pt))

V= pitch line velocity C= deformation factor e = sum of errors

v = (Pi*D*N/60)

D=63.85 mm, N = 3000 rpm, v = 10.029 mm/min, C = 11400 N/mm2

e = Eg+Ep

= 24.33

For Grade 6, value of e= 8+0.6Φ and grade 6 has been selected as gear manufacturing was done through hobbing

Thus value of Pd comes to be 7423.6 N

Peff = Cs*Pt + pd

= 1.5*3508.18 + 7423.6

= 12685.87 N

FACTOR OF SAFETY

FOS = (Peff / Sb) (Bhandari V. B., 2013)

FOS = (12685.87 / 6785.72)

FOS = 1.869

Thus this proves that the pinion is safe in bending with a yield strength achieved from material testing. The FoS of 1.869 is well above the desired limit of 1.5

WEAR STRENGTH OF HELICAL GEAR

According to buckingham's equation of wear strength

Sw = wear strength = (b*Q*Dp*K)/Cos2Ψ (Bhandari V. B., 2013)

b = face width, Q = ratio factor = 1.491, Dp = pitch circle diameter of pinion, K = load stress factor

Sw = (35*1.491*63.85*K)/cos20*cos20

Sw = 3773.41K

FACTOR OF SAFETY

FoS = Sw/peff

For the same factor of safety 1.869

Sw = FoS * Peff

3773.41K = 1.869 * 12685.87

K = 6.2834

To find the hardness of material

K = 0.16(BHN/100)^2

BHN = 636 (G. Thendral1, 2014)

Figure 1 Pinion 1

Figure 2 Gear 1

BEAM STRENGTH OF HELICAL GEARS FOR GEAR PAIR 2

Beam strength Sb = (mn*b*σb*Y)/cosφ (Bhandari V. B., 2013)

Y = Lewis form factor = 0.320 σb= bending strength = yield strength / 3

= 473 / 3 = 157.6 Mpa

Sb = (4*52*157.6*0.320)/cos20

= 11163.07 N/mm2

EFFECTIVE LOAD ON GEAR TOOTH

Effective load = Peff = Cs*Pt + Pd (Bhandari V. B., 2013)

Cs = service factor = 1.5 Pt = tangential load Pd = dynamic load

Pt = (2* Motor Torque / Diameter of pinion)

= (2*112*2.933/85.134)*1000

= 7709.25 N

Pd = (21v (cebcos2Ψ+Pt)cosΨ)/(21v+ √(cebcos2Ψ+Pt))

v = pitch line velocity C= deformation factor e = sum of errors

v = (Pi*D*N/60)

D=85.134 mm, N = 1022.84 rpm, v = 4.5594, C = 11400 N/mm2

e = Eg+Ep

= 24.836

Thus value of Pd comes to be 8882.43 N

Peff = Cs*Pt + pd

= 1.5*7709.25 + 8882.43

= 20,446.305 N

FACTOR OF SAFETY

FOS = (Peff / Sb) (Bhandari V. B., 2013)

FOS = (20446.305 / 11163.07)

FOS = 1.831

Thus this proves that the pinion is safe in bending with a yield strength achieved from material testing. The FoS of 1.831 is well above the desired limit of 1.5

WEAR STRENGTH OF HELICAL GEAR

According to buckingham's equation of wear strength

Sw = wear strength = (b*Q*Dp*K)/Cos2Ψ (Bhandari V. B., 2013)
b = face width = 52 mm
Q = ratio factor = 1.444
Dp = pitch circle diameter of pinion = 85.134
K = load stress factor
Sw = (52*1.444*85.134*K)/cos20*cos20
Sw = 7239.38K

FACTOR OF SAFETY

FoS = Sw/peff

For the same factor of safety 1.869
Sw = FoS * Peff
7239.38K = 1.869 * 20446.308
K = 5.278
To find the hardness of material
K = 0.16(BHN/100)^2
BHN = 574 (G. Thendral1, 2014)

Figure 3 Pinion 2

Figure 4 Gear 2

X. MATERIAL

According to the design calculations. The material used in gear should have the following specifications

Table 2 Material property requirement

Parameter	Value
Yield strength	480 MPa
Hardness	650 BHN
Carbon content	0.18%

Materials available are:

Table 3 Material chart

Material name	Tensile strength	Hardness	Carbon content
EN 353	650-700 MPA	175 BHN	0.1-0.2%
EN 24	850-1000 MPA	250-300 BHN	0.36-0.4%

Though the tensile strength and hardness of EN 353 is lesser than EN 24, but due to the high carbon content in EN 24, it makes it unsuitable for production of high speed gears. Thus it was decided to use EN 353 as the gear material. The hardness can be increased beyond 650 BHN through case hardening.

XI. MACHINING PROCESS

GEAR MACHINING

The gear manufacturing involves a series of processes one after the other. These processes are listed below

- Performing the gear blank without teeth.
- Preparation of the blank to the required dimensions by machining (mainly through lathe or CNC)
- Producing teeth on the gear blank by machining
- Surface or full hardening of the machined teeth through case hardening.
- Finishing teeth surface if required by grinding.
- Inspection of finished gears.

Table 4 gear machining steps

Task	Process
Preparing gear blank	This is generally done through casting or forging or metal casting.
Preparation of blank to the required dimensions	This was done on a CNC machine to have precision. As any weight imbalance can lead to vibration and noise.
Production of teeth on gear blank	This was done by hobbing as it is best suitable for manufacturing helical gears.
Surface hardening of gear teeth	This was done by quenching the material between 850 degree Celsius for 2 hours and then dipping it in oil (S Sathish kumar1*, 2014)
Finishing teeth surface	This was done by gear grinding.

GEARBOX CASING MACHINING

The gearbox casing manufacturing involves a series of processes one after the other. These processes are listed below

Table 5 casing machining steps

Task	Process
Preparing Step file	This is done on a 3D modelling software and then fed into the CNC machine
Preparing a wooden pattern	A wooden block is then machined and wooden pattern casings are prepared
Preparation of mould, gates, runer	Sand moulding was used to cast the gearbox casing from liquid LM 24
Post machining	This includes final surface finish of the casing along with boring of various holes for bearing fitment.

XII. SELECTION OF BEARING

On disassembling the tata ace gear box and the differential unit, it was found that they used single row tapered bearing. Following are the reasons for using tapered bearings

- They reduce friction during motion and thus reduce heat generation.
- They are able to transfer the load evenly due to it's shape and thus it is widely used in automobile transmission systems.
- They have high durability and can handle heavy loads.

For Shaft 1 which is attached to the motor

Bearing selected is 33206 (bearing)

Table 6 bearing specs on shaft 1

O.D	I.D	Weight	Max rpm	Pu (fatigue load limit)	C	Co
62 mm	30 mm	0.35 kg	8500	8.5 Kn	79.7 Kn	76.5 KN

For Shaft 2 which is attached to the intermediate gear

Bearing selected is 33108 (bearing)

Table 7 bearing specs on shaft 2

O.D	I.D	Weight	Max rpm	Pu (fatigue load limit)	C	Co
75 mm	40 mm	0.5 kg	7000	11.4 Kn	97.57 Kn	104 Kn

There were no bearings with ID 25,30,32,35 & 38 which could suffice the requirement of Co and thus I have selected a bearing with ID 40 mm

XIII. MOTOR SPECIFICATION

For the specifications mentioned in VI, after going through various 3 phase AC induction motors and found a motor for the same

1) 10 KW 96V

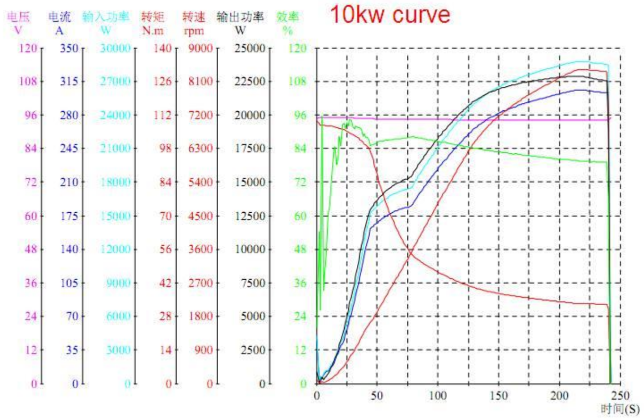

Figure 5 10Kw motor torque curve

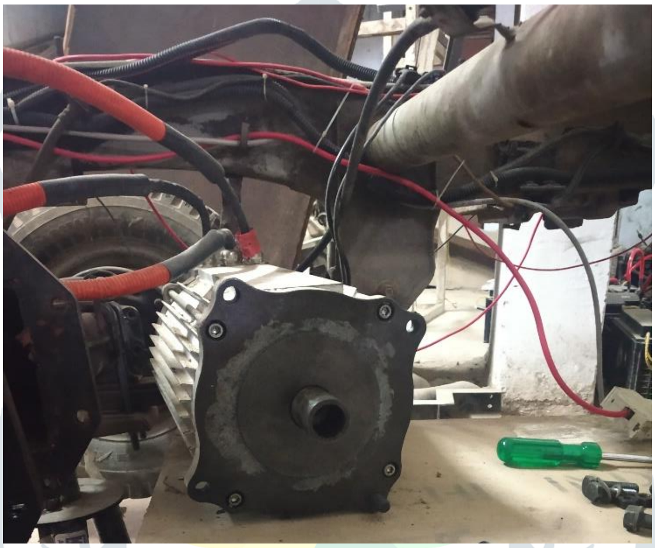

Figure 6 The motor

XIV. GEARBOX CASING DESIGN

Basically the casing has been designed in two parts from the vertical axis. Consideration has been taken to mount the motor on the casing along with housing the gear assembly. It will be manufactured from casting.

The basic consideration for casing design are 1) strength of casing 2) light weight material 3) higher heat dissipation 4) enough volume for lubrication

MATERIAL

Material used is LM 24 alloy as a material for casing. Following are the properties of LM 24 material

Table 8 casing material specifications

Parameter	Value
Tensile strength	180 MPA
Hardness	85 BHN
Density	2.79 g/cm3

The basic property of LM 24 is good casting, it is suitable for leak proof casings, good heat dissipation

Figure 7 Wooden Pattern

Figure 8 Casing

XV. FINAL ASSEMBLY

Figure 9 Final Assembly

XVI. LEARNING DURING MACHINING OF GEARS & GEARBOX

1) During manufacturing of gears the manufacturer didn't provide webs in the gear blank as designed. Which resulted into a huge play as both the pinions are built on shafts and gears are mounted on these shafts. So 2 spacers had to be manufactured but this increased the noise generated which functioning of gearbox
2) Pinion-1 was manufactured with faulty pitch circle diameter. This resulted into wobbling of the pinion during operation. This also generated a lot of noise which is way beyond standards. Also it would generate heat which reduced the performance of gearbox.
3) The oil pouring and drain plug points were not provided during casting. This was then done by machining by welding a 1 inch block on the spot.
4) Grinding was not done on the first pair of gears. This resulted into rough surface finish and resulted into noisy operation.
5) The bearing slots in gearbox casings were bored 2 mm more than the bearing width. Thus 2 spacers had to be used.
6) As there was no provision for bearing removal, there were small holes provided on the gearbox casing. This was done once the gearbox was manufactured.
7) One of the major problem leading to higher noise is the difference in tip diameter of the pinion and gears. This was suggested by the engineers at Hi-tech engineers.

XVII. CONCLUSION

- It's better to use helical gears for automobile powertrain than use spur gears as they produce very less noise at high rpm.
- The powertrain developed has a wide application in light commercial vehicle segment due to it's universal design.

XVIII. ACKNOWLEDGMENT

I have taken efforts in this research project. However, it would not have been possible without the kind support and help of many individuals and organizations. I would like to extend my sincere thanks to all of them. I am highly indebted to Mr Hitesh Raiyani, Assistant Professor, L.J.I.E.T. Ahmedabad, for their guidance and constant supervision as well as for providing necessary information regarding the project. I take this opportunity to thank all my friends and colleagues who started me out on the topic and provided extremely useful review feedback and for their all time support and help in each and every aspect of the course of my project preparation. I am grateful to my college L.J institute of engineering and technology, Ahmedabad for providing me all required resources and good working environment. I would like to express my gratitude towards Head of Department, Prof. Gayatri Pandi Jain for their kind co-operation and encouragement which help me in this project. Finally, we thank our families without whose support this project would have had no meaning.

REFERENCES

[1] a. Miranda1, n. A.-b. (2013). Study of mechanical properties of an lm24 composite alloy reinforced with cu-cnt nanofillers, processed using ultrasonic cavitation. Trans tech publication, 245-249.

[2] bearing, s. R. (n.d.). Skf bearing catalogue.

[3] bhandari, v. (2014). Machine design. Ahmedabad: mcgrawhill education.

[4] bhandari, v. B. (2013). Design of machine elements. New delhi: mcgraw hill education (india) pvt ltd.

[5] g. Thendral1, s. C. (2014). Effect of phosphorus in the heat treatment of en353 steel gears. Journal of basic and applied engineering research, 17-20.

[6] giri, d. N. (n.d.). Automobile mechanics. Khanna publication.

[7] gwangmin park a, s. L. (2013). Integrated modeling and analysis of dynamics for electric vehicle powertrains. Elseveir ltd.

[8] jinglai wu a, b. ,. (2018). Efficiency comparison of electric vehicles powertrains with dual motor and single motor input. Elseveir ltd, 569-585.

[9] juan de santiago, h. B. (2012). Electrical motor drivelines in commercial all electric vehicles: a review. Ieee transactions on vehicular technology, 475-484.

[10] k. J. Kim1, y.-c. L.-h. (2015). Fundamental research on power train systems for electric vehicles. Wiley-vch verlag gmbh & co. Kgaa, weinheim, 414-419.

[11] mech4study. (2014, april thursday). Retrieved from mech4study: http://www.mech4study.com/2014/04/types-of-gear-box.html

[12] mehdi mehrgou, i. G. (2018). Nvh aspects of electric drives-integration of electric machine, gearbox and inverter. Sae international, 1-6.

[13] pan zhang, y. C. (2015). Optimum matching of electric vehicle powertrain. Elseveir ltd, 894-900.

[14] s sathish kumar1*, b. S. (2014). Heat treatment on en 8 & en 353 for heavy duty gears. International journal of mechanical engineering and robotics.