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A Method to Optimize Brass Type Single Synchronizer Ring for Manual Gearbox

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Abstract: - Global automotive industry is growing rapidly in recent years. Increased torque range with deeper transmission ratio as well as overall vehicle refinements in modern trucks places higher demands on transmission and its components. Changing market requirement and driver comfort is also a new challenge in commercial market. With this overall effects design and development of transmission and its components plays a major role for performance, comfort and economically driven market.

Synchronizer is one of the critical components which need to look upon. Synchronizer plays a vital role for gear shifting performance in manual gearbox without any shifting assistance. Synchronizers primary function is to reduce the rpm difference between two gears before gear shifting subject to time and more comfort. Increasing demand for performance, enhance life, less shift effort, less synchronization time, optimum size of Synchro pack and low cost has become an interesting area for engineer for improvement. More over existing synchronizer can be economically re-engineered into more efficient designs with the help of new mathematical tools and techniques.

A mathematical optimization model can be developed to establish optimum cone parameter. A model of relationship can built between cone angle and sleeve champers angle, synchronizer size, coefficient of friction, cone torque and index torque. In this study a model was built in Matlab optimization toolbox pattern-search solver to find global minimum value of Cone angle which satisfy set of given and known design constrains. This Model can be used to select first cut synchro parameter with moderate accuracy. These Values can be then finer tune by customized and specialized software's.

It is not easy to study the entire gear-changing process from clutch till wheels, dynamical behavior of manual gearbox is too complicated to simulate, owing to the large number of elements involved in it. This study considers only major parameter involved in synchronization. Also generous experience and data is required to accurately select parameter with desired Shift feel and comfort.

Index Terms: - Synchronizer, Matlab, Blocker ring, optimization Gearbox.

I. INTRODUCTION

Current market scenario shown in Fig 1, Synchro ring with Brass material has 55% market share. Molybdenum coated rings, sinter lining and organic friction linings are some of the other material used for friction lining. Multi-cone synchronizer behave in same manner to single cone, the difference is multi-cone have more than one contact friction lining. So focus is slender on to brass ring synchronizer.

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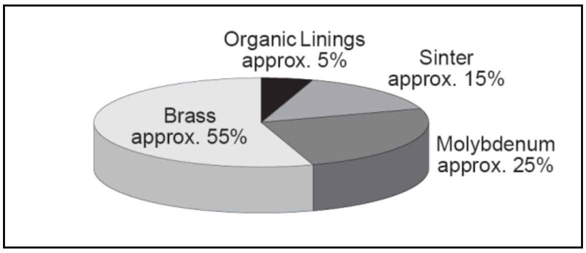

Fig.1 Market Share of today's friction materials in current synchronizers

Synchronizer plays a vital role for gear shifting performance in manual gearbox without any gear shifting assistance. Function of synchronizer is to reduce the rpm different between current and desired gear for ease of gear shifting, with less shift time and more comfort to driver. Constant improvements in engine, clutch performance, increasing torque, load carrying capacity of vehicle and grade-ability place increasing high requirements on manual gearbox and its components. Improvement over gearshift feel, comfort, performance, reducing shifting force and optimizing space between the gears and costs are nowadays has become primary objectives for the new generation of synchronizer systems. A five speed manual Gearbox without any shifting assistance using brass type single synchronizer can be taken to find methods of improving gear shifting performance. A Typical Manual 5 speed Gearbox layout is shown in Fig 2

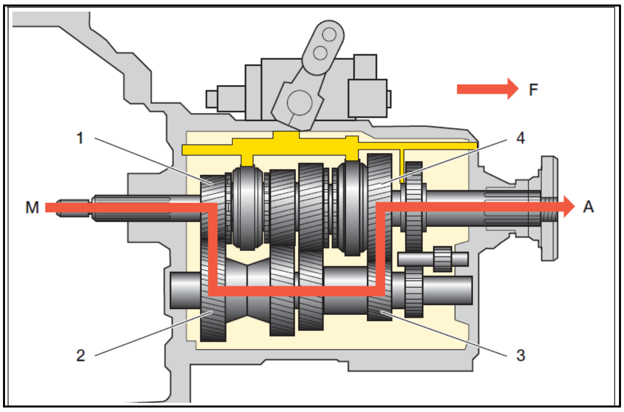

Fig.2 Gearbox Layout

A cut section is shown in Fig.3 for detailed understanding. Ring parameters such as cone angle, mean radius and blocker/ cone friction coefficient can be selected optimum to trade-off between Clash & comfort Zone. These parameters can be optimized with the help of Matlab optimization solution. A Parametric relationship can be built which include shift force, specific power dissipation of ring, limited detent force of spring, max rubbing velocity and stresses in ring.[1][2]

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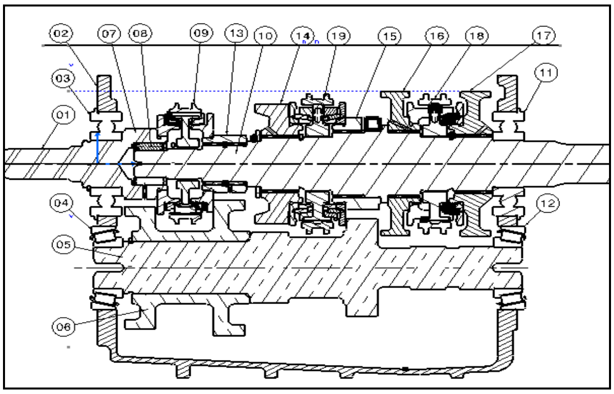

Fig.3 Gearbox Cut Section Layout

II. SYNCHRONIZATION SYSTEMS

A. Components in Single Cone Synchronizer Assembly

The single-cone synchronizer is a conventional blocking synchromesh based on the Borg Warner. Components in Borg Warner are shown in Fig 4. Its consist of Synchro pack which is assembly of Blocker ring, Synchronizer hub, Strut, Ring spring which is use as pack for synchronization between adjacent meshing gears.

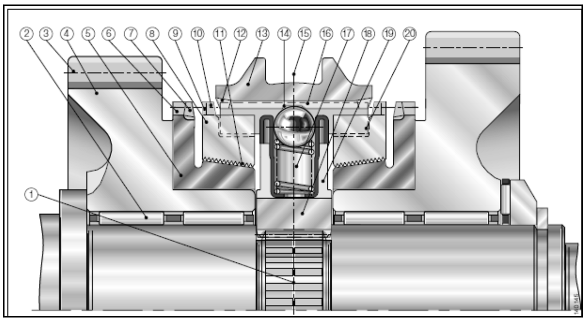

Fig.4 Components in Borg Warner

1. Synchronizer ring:-

The shape of ring is like cup, this behave like cone clutch. Cup surface of ring when forced on gear cone create a frictional torque. This torque is used to reduce rpm difference between selecting gears. The usual angle of this surface is so called as cone angle generally varies between 6 to 7 deg. The cone surfaces are provided with thread and axial grooves which displace the lubricant faster.

A chamfer tooth is cut on the outer periphery on which sleeve teeth chamfer face butts. When the sleeve is force on this chamfer a resistive torque is generated which is called as indexing torque. Till there is rpm difference between gears this torque continues to generate on teeth. As this torque reduces to zero, synchronization is done and ring move to its neutral position.

2. Synchronizer hub:-

Synchronizer hub is positively locked with the gearbox shaft. It contains the components for pre-synchronization in a strut slot and guides the shift sleeve in a notch at the outside diameter. Three notches on the circumference ensure that the blocker ring does not rotate. The shift sleeve has spline teeth on inside diameter. In a circumferential groove has outside diameter in which the shift fork slides and moves the shift sleeve in the axial direction

3 Struts:-

The struts are arranged on the circumference of the synchronizer body and spring-loaded against a notch in the shift sleeve teeth. They retained by means of spring-loaded balls.

4. Gear cone body

The gear cone body is made from steel and splined to the constant mesh gear. It has an outer friction cone which has a high surface hardness with good surface finish. The clutching teeth with roof-shaped chamfers face the blocker ring, in which the sleeve is locked after synchronization.

5. Constant mesh gear

The constant mesh gear has needle roller bearing supports on the shaft and is designed with Involute gear teeth for the gearbox of torques.

B Synchronization Working

As there is demand for torque or speed at wheel end driver shift gears as required. The gear shifting knob is connected to gear shifting mechanism on the top cover of gearbox. Through the shifter linkage the force is transfer to the fork on gear sleeve. The shift sleeve is moved out of the neutral position and displaced axially toward the constant mesh gear. Because of chamfered teeth on the shift sleeve, the struts are also moved. They press the blocker ring against the friction cone at the clutch body of the constant mesh gear. This allows a frictional torque to build up and the gear is pre-synchronized.

Cone torque is primarily a function of the axial force applied to sleeve, cone angle, coefficient of friction, and cone diameter is shown in Fig 5.

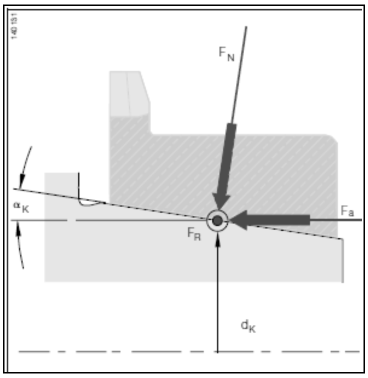

Fig.5 Cone Torque

The cone torque is countered by the index torque shown is Fig.6 .Cone torque must be greater in magnitude to overcome the torque due to ring indexing force and complete synchronization successfully, Tc ≥ Ti. [2][3]

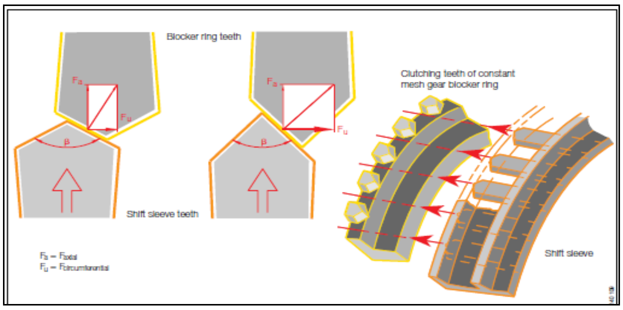

Fig.6 Locking Torque /Indexing Torque

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For the uniform dynamic friction between the mated cones during the entire sliding phase, the locking torque Ti must be sufficiently high.
Solving and Rearranging terms, we get an in-equality; in this left hand term must be always higher than right hand term to avoid Clashing.

$$\tan \beta \geq \frac{\frac{r}{R} - up\,\frac{uc}{\sin a}}{\frac{uc}{\sin a} + up\,\frac{r}{R}}$$

Due to the cone frictional torque the blocker ring immediately rotates with the available clearance of the notches in the sleeve support. The chamfered teeth on the shift sleeve contact the blocker ring teeth, thereby preventing a premature, axial shifting of the shift sleeve. As the axial displacement force increases, fully effective frictional torque now aligns the differing speeds between the constant mesh gear and the hub and the gear is synchronized.[4][5]

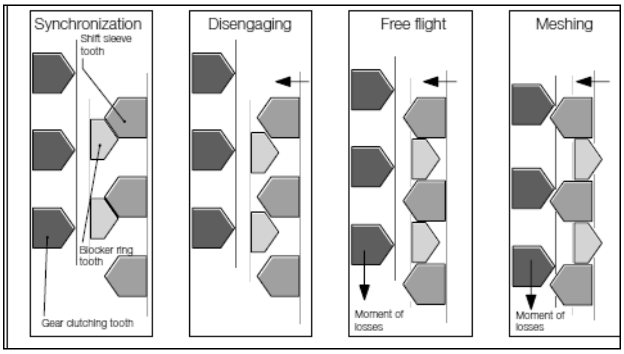

Fig.7 Various phases of Synchronization.

C. Other contributing factors

1. Break Through Load/ push through- load effectively set the blocker ring into block position.
2. Proximity-Axial distance from sleeve tooth pointing to the blocker ring tooth pointing contact.
3. Impulse should be limited below acceptable limit of synchronizer mechanism.
4. Coefficient of friction depends upon cup and cone material, number of contact surface and oil temperature.
5. Power losses due to the clutch drag, gearbox oil churning and friction of rotating components.
6. Losses due to bearing, seals and other mechanical components. [6]

III. LITERATURE REVIEW

Richard J.Socin, L. Kirk Walters, 680008, presents general design parameters for Borg warner type synchronizer, he presents design parameters, method of evaluation and variable which is important in synchronization process.
Syed T. Razzacki, develop mathematical algorithm which facilitates establishing the sleeve and blocker ring pointing angle relationship with the synchronizer size, coefficient of friction, cone torque, and index torque. He presented graphically the clash and hard shift zones. Also it allows sizing ring and selection of the parameters for a given application subject to comfortable shift ability between the two extremes of clash and hard shift.
The Nomograms assist in viewing significant physical parameters relationship instantly.
LászlóLovas, Daniel Play, JánosMárialigeti and Jean-François Rigal, develops modeling of the gear changing behavior. Based on this a numerical simulation software of the synchronizer behavior was realized. Phenomena of angular velocity synchronization and second bump in gear changing force are modeled in details. Author considers mechanical behavior with large number of input parameters. Effects of gear changing force, synchronized inertia and initial angular velocity difference variations are shown.
Andreas Gustavsson, show Dual Clutch Gearbox are more efficiency and comfort. Its construction was much like two parallel manual gearboxes, where the gear shifts are controlled automatically. The gear-shift of a manual gearbox involves a synchronization process, which synchronizes and locks the input shaft to the output shaft via the desired gear ratio. This process, which means transportation of a synchronizer sleeve, was performed by moving the gear shift lever which was connected to the sleeve
T. M. Manoz, Kumar, Sandeep D'mello and V. Pattabiraman, present a new methodology evolved for the optimized design of synchronizers of automotive synchromesh gearbox using statistical simulation techniques. The design methodology involves developing a transfer function relating the design parameters of synchronizer with that of the performance requirements. The performance requirements i.e., shift quality, synchronization efficiency and synchronizer life are predicted based on gear shift force, clash ratio and specific power dissipation & Hoop stresses respectively.
The statistical method of Design of Experiments' was used with the aid of simulation tool for the optimization of synchronizer performance
Daniel Angstrom and Mikael Nordlander develop MATLAB program for calculating synchronization of gearboxes, with focus on versatility, accuracy and user-friendliness and verify it by physical testing.
The tri-biological contact is represented by using experimentally determined friction properties and material limits and comparing calculated values with the material limits.
David Kelly Christopher Kent, made research on Ricardo who has many years' of experience in investigating gearshift quality problems and has created software and hardware for the measurement and analysis of gearshift quality. Gearshift quality is made up of several different areas including gate definition, shift effort, second load and vehicle response.
Shift effort requires careful sizing of the synchronizer cones, cone angle, friction material and balancing this with the gearbox reflected inertia, clutch inertia and gearbox drag. Vehicle response requires a full gearbox and driveline model coupled with a vehicle model which includes engine and gearbox mounts and suspension components. This analysis may be best achieved by using a co-simulation approach where different analysis tools aroused for gear shift and suspension modeling. Ricardo has developed a series of dynamic models to investigate the dynamic effects of gearshift quality at the design stage and for specific gearshift quality development.

ANA PASTOR BEDMAR, identified quality factors such as the shift comfort and shifting time, as well as the problems

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that could appear such as double bump and stick-slip phenomenon.
In addition, the first three stages of the process have been simulated using Matlab and the equations of motion have been established for the remaining five stages.

IV. MATHEMATICAL MODEL SET UP

A. Quality and Qualifying factors

1. For general LCV application, Gear Shifting Time, 0.2 s to 1 s, can be considered as good assumption.
2. Knob effort should remain, between 60-100 N
3. Size Ratio, r/R should lie between 1-1.4, for synchronizer hub-pack packaging.
4. For brass type synchronizer ring Cone friction, μc, lie between 0.1-0.11
5. For brass type synchronizer ring Cone static friction, μc, lie between, 0.09-0.11
6. Practically Index angle β lie between, 105-115 deg for better result.
7. Brass Material has Energy Capacity and Power Capacity up to 1.5 J/mm2 and 1.5 W/mm2 respectively so this value should not be exceed
8. Shift Impulse is another limiting criterion should be kept below, 100 N-s
9. For Brass material Rubbing velocity limit is 10 m/s

B. Vehicle Configuration,

LCV Vehicle with 9T GVW, 3.3 Lit Engine
Max Power 115 Hp@ 2800, Application-School Bus

Engine Model
1. Torque-RPM characteristics, 325 Nm @ 1800 rpm
2. Engine Min, 1000 rpm
3. Engine Governing, 3000 rpm
4. Engine Max, 5000 rpm

Clutch Model
1. Clutch Inertia, 0.034 km2
2. Clutch Drag, 1 Nm
3. Clutch Capacity, 400 Nm

Driver Model
1. Gear Shifting Time, 0.2-1s
2. Gear Lever Force, 60-100 N

Gearbox Model
1. Gear Ratio's, 5.26; 2.85; 1.59; 1; 0.75
2. Rotating Components Reflected Inertia, 0.046 kgm2
3. Synchronizer Parameters

Gear shifting Model
1. Lever Ratio, 6.75
2. Detent force, 3x10 N

Synchronizer Limiting Range,
1. Index Radius/Cone Radius, 1-1.4
2. Index friction, 0.09-0.11
3. Cone friction, 0.1-0.11
4. Index angle, 105-115 deg

Synchronizer,
1. Width, 10 mm
2. Contact Ratio, 0.6
3. Efficiency, 0.85%

Synchronizer Limiting Parameter (Brass)
1. Energy Capacity, 1-1.5 Jmm2
2. Power Capacity, 1-1.5 W/mm2
3. Impulse < 100 Ns
4. Max Frictional torque ~ 22 Nm

5. Rubbing velocity, <10 m/s

Optimum Parameters,
1. Cone Radius/Indexing Radius
2. Cone Angle for selected Index Angle

With this input value, using mathematical model and reference equations [6] [7] [8]

B. Problem Setup.

1. Defining Solver: - "patternsearch-PatternSearch"

This is a global optimization solver, which gives global minimum value. It includes Objective function, Constrains and Non-Linear constrains function.

2. Setting up Objective Function which is,

1. $$\alpha = \sin^{-1}\left(\frac{R*\mu c*\mu p + R*\mu c*\tan(\beta)}{r - mp*r*\tan(\beta)}\right)$$

3. Setting up Non-Linear constrains function

Limiting Non-Linear equation parameters

3.1 Limiting Size Ratio

1. $c(1) = \frac{r}{R} - 1.4$
2. $c(1) = -\frac{r}{R} + 1$

3.2 Limiting Cone and Index Torque in Nm

1. $Tc = \frac{fa*R*\mu c}{\sin(\alpha)}$
2. $Ti = r*fa*\left(\frac{\cos\beta - \mu p*\sin\beta}{\sin\beta + \mu p*\cos\beta}\right)$
3. $Tr = \frac{Tc}{Ti}$
4. $c(3) = \frac{Tc}{Ti} - 1.6$
5. $c(4) = -\frac{Tc}{Ti} + 1$

3.3 Limiting energy capacity/area

1. $KE = \frac{Igb * \Delta rpm^2}{2}$
2. $\frac{KE}{mm2} = \frac{KE}{cA}$
3. $c(6) = KEmm2 - 1.5$

3.4 Limiting Synchronizing time

1. $ts = \frac{Igb*\Delta rpm}{Tc}$
2. $c(5) = ts - 0.4$

3.5 Limiting Power capacity/area

1. $Pmm2 = \frac{KEmm2}{2*ts}$
2. $c(7) = Pmm2 - 1.5$

3.6 Limiting rubbing velocity

1. $V = R*\Delta rpm;$
2. $c(9) = V - 10$

3.7 Limiting Impulse

1. $Ip = fa*ts;$

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2. $c(8) = Ip - 400$;

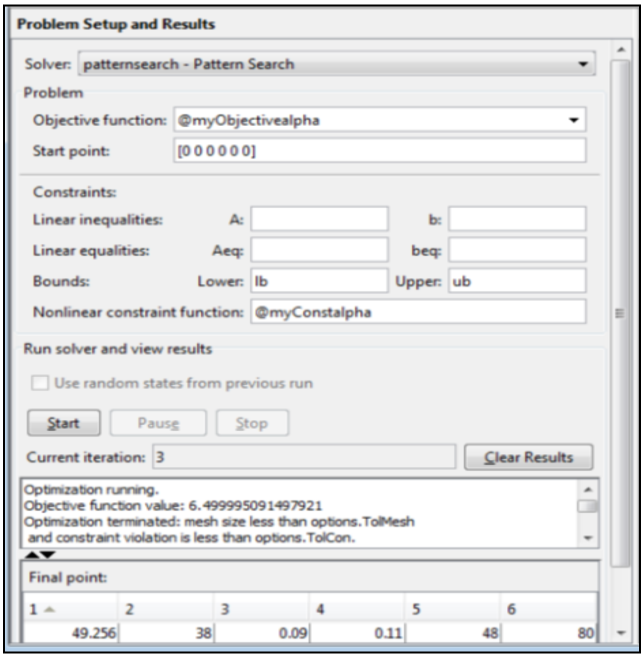


Fig.8 Problem Setup and Results

4. Setting Constrain bounds (Permissible Limit)

Bounds	R	r	uc	up	β	Fa
Lower	46	38	0.09	0.10	48	60
Upper	60	38	0.11	0.11	62	100

Table No:-I Setting Constrain bounds

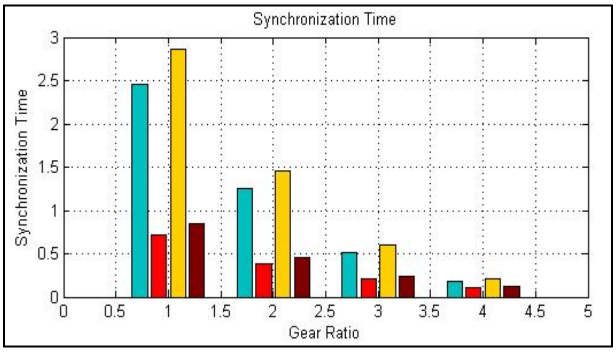
V.RESULTS AND OBSERVATIONS,

A. RESULTS:-

Single Synchronizer Parameter 4th-5th Gear.

Single Cone Brass Ring	Synchro 1	Synchro 2
	Existing	Proposed
Cone Radius (mm)	38	38
Indexing Radius (mm)	46	49
Indexing Angle (deg)	105	112
Cone Angle (deg)	7	6.5

Table No:-II Single Cone Brass Ring



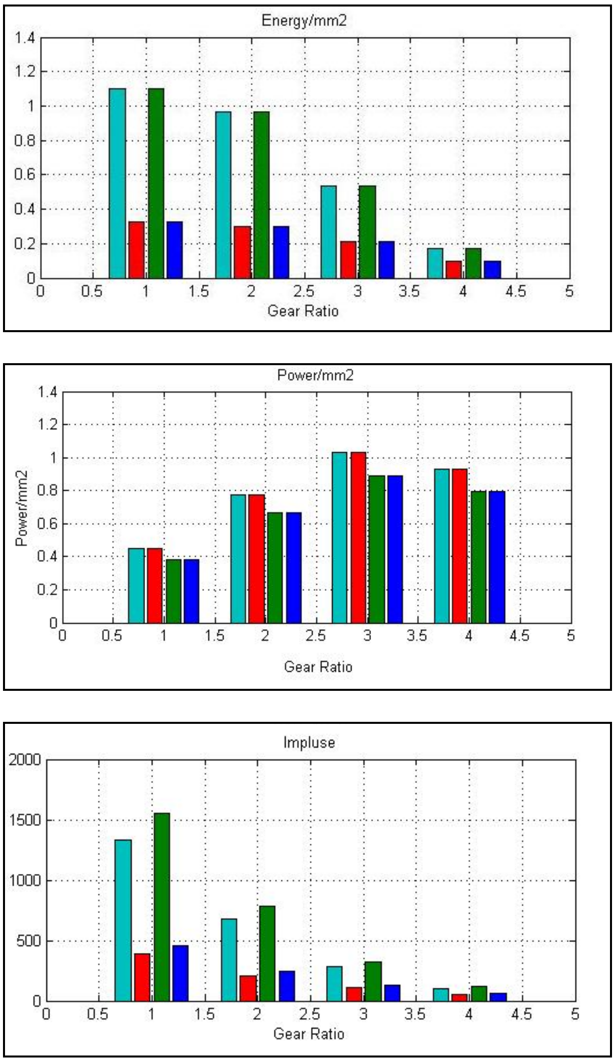


Fig.9 Results

B. OBSERVATIONS:-

Shift feel for LCV (Light Commercial Vehicle) is applicable

1. Inertias: Total gearbox inertia higher than the general trend for a gearbox with given torque Capacity.
2. Single Cone Synchronizer is not suitable for 1 gear ratio, double cone synchronizer need to use. Single Cone Synchronizer can be use between 2-3, but with negotiation on Handball force. Can comfortably use with between 4-5, for usual shifting time of synchronizer in LCV application
3. Handball force & impulse:

a. For Cone angle of 7 deg, with handball force of 60N which is comfortable limit for very ease, for 4-5 gear synchronizer, shifting time (ts) of 0.12s for upshift and 0.214s for downshift

b. For Cone angle of 7 deg, With same upshift time of 0.12s for downshift handball force has to increase from 80N to 140N

c. Compare to cone angle of 6.5 deg, with handball force of 60N which is comfortable limit for very ease, for 4-5 gear synchronizer, shifting time (ts) of 0.104s for upshift and 0.2s for downshift

d. With Cone angle of 6.5 deg, With same upshift time of 0.11s for downshift handball force has to increase from 80N to 145N

e. For 2-3, gear ratio synchronizer time of 0.6s in downshift is though less than 1s, but improvement and

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shift feel may not be comfortable as the speed difference to be synchronized is high.

4. Clash avoidance geometry: Cone torque over index torque (Tc/Ti):
   a. Single cone 4/5 while Cone torque to Index torque ratio with cone angle of 7 is marginally, more value is required for shifting comfort. So it is recommended to modifying cone angle to 6.5 deg for improved ratio and handball impulse.
   b. With single cone, for comfortable shifting trade-of between clash and hard shifting usually indexing angle is kept between 105 to 125 deg, for Tc/Ti of 1.3 which is recommended for LCV, Index angle of 112 deg need to use instead of 105 deg to improve Index torque and Tc/Ti overall.
5. Synchronizer energy capacity: is a function of cone geometry and shift energy. Maximum permissible levels for brass synchro rings are 1-1.5 J/mm2. Even under extreme shift conditions, maximum shift energy is 1.1J/mm2, which is within acceptable range. For 4-5 ring, maximum shift energy is 0.0962 J/mm2 in upshift and 0.1711 J/mm2 in Down shift which are within permissible limit
6. Synchronizer powers capacity: For the synchronization time of 0.2s the synchronizer powers are within the acceptable levels even for skip shift as below
   a. 3-4 the synchro power is 1.33 W/mm2
   b. 4-5 the synchro power is 0.9285 W/mm2

VI. CONCLUSION

A mathematical model was developed in Matlab optimization tool box to select optimum synchronizer parameter. LCV 9 ton, 5 speed manual gearbox with brass type single synchronizer vehicle was considered. Vehicle specification and required parameter like Torque-rpm characteristics, clutch drag, rotating components inertia till synchro was noted. Available synchronizer was measured for improvement. Objective and Constraints function were built and related function was correlated to parameter which has to optimize. Objective function was formed which has to be satisfy in-equality and size & angle parameters, material and comfort limits. For optimization Pattern-Search minimization solver was used to in Matlab, which gives global minimum value.

Practically this angle varies in the range of 6 to 7.5 deg for Brass type synchro ring and that of Index angle varies between 105 to 115 deg, these values together with size effects Cone and index torque. This cone and index torque should lies in comfort zone, bound between clash and hard shift zone.

Mathematical model show existing synchronizer parameter, was not refine, results show cone angle of 6.5 deg with size ratio of 1.3 is more appropriate for such application.

Future scope is to model each and every component in gearbox from driver knob till tire, including driver and road model including dynamic parameter to optimize synchro pack.

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