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A Comprehensive Method for the Hybridization of a Conventional ICE Vehicle Using a Flexible Design Tool

S.A.A.Shah[1], M.A.Khan[2], H.M.Zubair[3], H.U.Hashmat[4], A.Qadir[5].

[12345] Mechanical Technology Department, The University of Lahore, Pakistan

[1]arslan.shah15@gmail.com

Abstract- The powertrains are having vital role now a days due to fuel economy and less tailpipe emissions. These are helpful in fuel economy and environment friendly architecture development. This research work will provide a helping tool for the elaborated understanding and core mechanical designing of the power transmission device (mechanical coupling) used in hybrid cars, especially in Synergy drives. The device combines the powers, in form of product of angular velocities and torques from various power sources i.e. IC engine and the motors. It is a flexible design tool that can be aided to design any vehicle of any size, power, speed and acceleration. In this paper a small sized vehicle is considered as an example vehicle that can be hybridized using this flexible design tool. The results generated are also verified through software simulations using the advance version of Pro-E i.e. PTC Creo. Then using these results, the motor sizing for the required degree of hybridization is also recommended along with the nature of performance curve required by the motor to provide the required design acceleration, speed and power.

Keywords- Hybrid Engine, the Dual hybrid Architecture, Power split device, Design of Planetary Gear Set (PGS).

I. Introduction

According to 2017's latest global review, proved oil reserves have fallen slightly from 17000 billion barrels to 1696.6 billion barrels which equals 0.5 billion barrels (-0.03%), which might be sufficient only for the next 50.2 years with global production rate of 2017 levels [1]. According to the U.S. Environmental Protection Agency (EPA), the vehicle's factor of “greenness” is based on several specific factors. These scoring categories include: air pollution, fuel economy, and greenhouse gas.

All big automobile manufacturers are very keen and interested in the development of Hybrid Electric Vehicles (HEVs) and Plug in Hybrid Electric Vehicles (PHEVs). The reason behind is the proven success given at both the ends i.e. companies and consumers. Also rules are being made at international and government levels based on different economic, health and environmental, aspects and effects promoting the success of greener cars [2-3]. In Pakistan hybrid technology has been introduced, but still it's not being implemented on smaller vehicles. So there is a gap that needs to be filled by providing a design tool for hybridization of the smaller vehicles in Pakistan. There are many different kinds of green vehicles and one of the most popular and most common types is hybrids. Hybrid Electric Vehicles (HEV's) and Plug in Hybrid Electric Vehicles (PHEV's) use a mixture of gas and electric power to help improve mileage and reduce emissions [4].

Hybrid fuel vehicles named as FCHVs (Fuel-Cell-Hybrid-Vehicle) now a days having a vital role in latest transport sector [5-6]. Fuel cell is having better features than hybrid and electric vehicles. This is more economical source of energy. If the model of FCHV powertrain is generic and specifications of fuel cell, control model and battery sizes are well designed, it can give better results. [7]

The HEV (Hybrid-Electric-Vehicle) technology is also having a vital role in the cleanliness of environment as the fossil fuels are not used and pollution is controlled. This is also used for fuel economy and less tailpipe emissions control. This can affect directly the fuel prices as less fuel is consumed. [8]

Powertrains are used to save the energy and fuel economy. Parallel hybrid vehicle, series hybrid vehicle and power split hybrid vehicles are now converting to pure electric powertrain architecture system [9]. The powertrains are having more complex system which is called electrified powertrains with two or more fundamental types and subtypes.

HEVs are driven by an ICE and a single or couple of electric motor/s. The ICE provides an extended range of driving to the vehicle, while the Motor/Generator (M/G) serves the purpose of increasing the efficiency and fuel economy by regenerative braking and by storing excess energy from the IC engine during the engine is turned on.[10]. This combination of components providing the power to drive the vehicle is the powertrain. Basically, there are two basic types of HEVs, series and parallel hybrid [11]. In series hybrids, the mechanical output of the ICE is first converted to electricity through a generator that is either propels the wheels through an Electric Motor (EM) or is used to charge the battery. While in parallel type, both the ICE and the EM are coupled together and has the advantage that both these power sources can provide power to the wheels at combined mode [12].

The design of powertrain is composed of designing the power split device that is composed of an assembly of planetary gears. In the example design opted here, two motors are used along with a conventional small sized engine. . The gear ratios and their number of teeth has a great role to play in the design, along with the selection and sizing of the engine the motors and the batteries. Along with that the different driving modes also depend upon the gear ratios for example the angular speeds of both the motors have their maximum rpm limits and the speeds are dependent on each other on the basis of gear ratios i.e. if the gear ratio of sun(MG 1) and ring(MG 2) is 3:1 respectively, then 2000 rpm of MG2 will give 6000 rpm to Motor Generator 1 (MG1), and in case of gear ratio of 2:1 the same 2000 rpm of Motor Generator 2 (MG2) will give 4000 rpm to Mg1.

The paper composition is as, at first the introduction along with the need of hybridization of vehicles worldwide has been given, secondly, the dual hybrid architecture has been selected, which is a combination of both series and parallel architecture. Thirdly, the vehicle to be hybridized is selected. Fourthly, the vehicle parameters are determined. At sixth, the gearing parameters for the planetary gears are decided. At seventh, the vehicle parameters and the gearing parameters are fed into the software tool and angular velocities, torques, and powers, required on the planetary gears i.e. power train are calculated along with the performance curves, which can be used for the design of motor's and its sizing. In the last, the above parameters are verified using the software simulations. (*) Will represent the owner's own work.

II. Selection of Hybrid Architecture

The dual hybrid architecture has been opted as the target architecture for our example vehicle, as it is mostly being adopted in all the latest hybrid technology because of its better efficiency and advantages. As it has more essence of parallel hybrids i.e. it allows the mechanical combination of both the motor and the engine with the transmission. While the advantage it grabs from series hybrid is the addition of separate generator for charging the batteries through the engine during normal driving condition. [7]

The dual architecture hence uses one power input from the IC engine and another two power inputs from the electric motors which can both work as generators also, at the time of storing energy to the batteries. The Planetary Gear Set (PGS), shown in fig 1, is composed of a ring gear, sun gear, carrier arm, and pinion gears. In normal PGS applications, one of the three main components of the PGS (i.e. ring, sun and carrier arm) is locked, one gets the input and the output comes at the third moveable gear. But in our application, all the gears of the PGS are free to move. The ring gear is coupled with the larger Electric Motor and Generator 1 (EMG1), sun gear with the smaller Electric Motor and Generator 2 (EMG2), while the carrier arm is coupled with the engine.

The final output of this PGS comes out from the ring gear with which the main larger traction motor is attached. At electric only mode, only EMG1 is working and the engine is turned off i.e. the arm is stationary. At more power requirements, the engine kicks in by turning on the smaller motor (works as a starter motor for a momentary time), both the engine and EMG1 works to supply the required power boost.

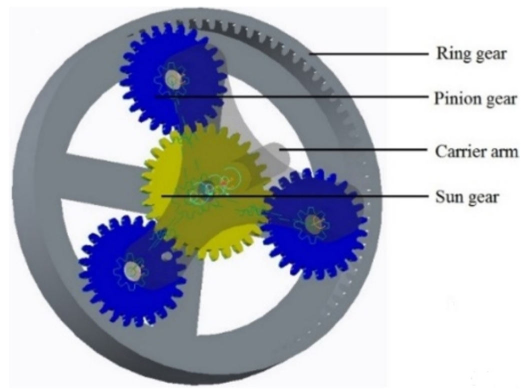

Fig. 1. Planetary Gear Set (PGS)*
*Author's own work

III. Vehicle Opted to Be Hybridized

The vehicle selected is a small IC engine powered 3 wheeler Public Utility Vehicle (PUV) mostly used in Asian countries for the purpose of local public transport. The government of Philippine has also launched a PUV modernization program in 2017 which aims towards making the public transport system efficient and environmental friendly till 2020 [8], as the PUVs needs to run on the road round the clock and hence needs to be focused for better environment and efficiency.

The PUV selected has a 219 cc, single cylinder engine. It uses a manual gearbox for the required five speed and torque variations. The performance curves are also been used for the design purpose (Curtesy of the Saazgar Engineering Works, Pakistan). These curves are used to find the gear ratios for our PGS that are compatible with the designed ratios of the manual transmission, but this automatic transmission would be much smoother and quieter than its manual counterpart.

Fig. 2. 3-Wheeler PUV Model*

IV. CALCULATION OF VEHICLE PARAMETERS

The reverse engineering methodology has been opted, hence the following parameters* are calculated to find the rpm, torque, force and power at the wheels required for particular velocity ranges, and with the help of the differential ratios these parameters required at the output of the PGS are determined. The vehicle selected parameters* found are as follows Table I.

TABLE I: PGS PARAMETERS

Parameter name	Value
Wheel diameter (D_w)	470 mm
Vehicle curb mass (m)	370 kg
Curb weight (W)	3629.7 N
Vehicle velocity (v)	(0-100) Kph
Wheel velocity (N_w)	(0-1128) rpm
Frontal area (A)	1.544m²
Differential ratio (D.R)	2.5:1
Rolling coefficient (C_r)	0.017
Drag Coefficient (μ)	0.9
Angle of slope (Θ)	0-4°

To total traction force F_T is the sum of drag force F_d, rolling resistance force F_r, gradient force F_g, and acceleration force F_a. These forces are calculated as follows,

$$F_T = F_d + F_r + F_d + F_a \ldots\ldots (1)$$

$$F_d = \frac{1}{2}\mu\rho A v^2 \ldots\ldots\ldots\ldots\ldots (2)$$

$$F_r = C_r \times W \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

$$F_g = W \times sin\theta \ldots\ldots\ldots\ldots\ldots (4)$$

$$F_a = ma \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

The acceleration is not a fixed parameter and it is demanded by the driver, but the acceleration ranges can be designed depending on the modes of drive and the driving conditions. The more the acceleration needed more the acceleration force required and hence more is the total tractive force. The various road tests were performed for this targeted PUV, and the acceleration results extracted were used to design the acceleration ranges, while staying confined in the power envelop available. The possible and proposed acceleration curve for our model vehicle is shown in figure 3. This curve may vary depending on the slope of inclination, velocity and power available for the drive i.e. at lower speeds more power is available to be utilized so the acceleration can be much higher than the proposed one.

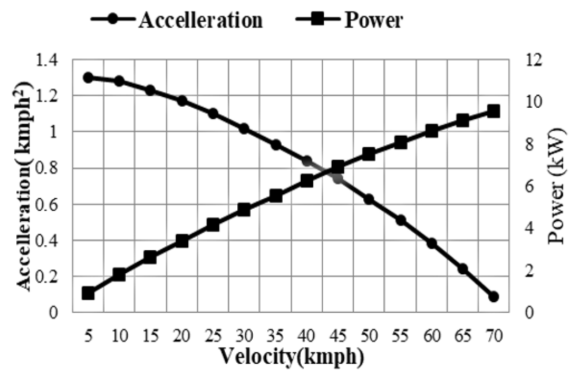

Fig. 3. Proposed Acc. Curve for Electric Mode*

V. CALCULATION OF GEARING PARAMETERS

The gear ratios and parameters are to be defined and designed, then on the basis of required force, torque, power and different driving modes and their conditions the motors and battery banks can be designed. As the design given in this paper can be used as a flexible design hence the values of gear ratios gear teeth and in turn the motor sizes and specs can be varied accordingly. But the gear parameters designed here are designed to meet with the vehicle performance specs. The number of teeth for the PGS are as follows,

$$N_{sun} = 30$$

$$N_{ring} = 78$$

$$N_{arm} = 108$$

$$N_{Pinion} = 24\ldots\ldots\ldots (6)$$

So, the gear ratios for the PGS between sun gear: ring gear and sun gear: carrier arm will be, α and β respectively. Where,

$$\alpha = \frac{N_{ring}}{N_{sun}} = 2.6 \dots\dots\dots\dots (7)$$

$$\beta = 1 + \frac{N_{ring}}{N_{sun}} = 3.6 \dots\dots (8)$$

The gearing parameters including addendum dedendum, diameteral pitch, circular pitch, pitch circle diameter, module, tooth thickness, clearance, are calculated on the basis of number of teeth, pressure angle, and module required, and are shown in table 2.

TABLE II: GEAR PARAMETERS OF PGS (mm)*

Parameter Name	Value
Diameteral pitch	0.1843 mm
Pitch circle diameteral (ring)	188.05 mm
Pitch circle diameteral (sun)	72.33 mm
Pitch circle diameteral (arm)	130.19 mm
Module	2.411 mm
Circular pitch	17.042 mm
Addendum	5.4247 mm
Dedendum	6.7809 mm
Tooth thickness	8.5211 mm
Clearance	1.3561 mm

VI. USE OF THE SOFTWARE TOOL FOR CALCULATION OF DESIGNED PARAMETERS

Finally, the main parameters which are power, angular velocity and torque required at the final transmission are needed to be determined at various speeds and modes. As in this method all these parameters are first calculated on the wheels, then on the powertrain.

A. Calculation of Power at Wheels P_w

It is the product of wheel velocity v_w and total traction force F_t on the wheels is calculated by,

$$P_w = F_t \times v_w \dots\dots\dots (9)$$

The velocity range is the selection of the designer, any range can be opted, and here it is (0-100 kmph), while the total traction force is calculated corresponding to this velocity range, and hence the power is calculated. All these values are calculated using Microsoft excel software.

B. Calculation of Wheel rpm N_w

Wheel rpm would be calculated with the help of above parameters i.e. wheel diameter (D_w), circumference and the vehicle velocity, v = (0-100 kmph). The circumference (1.477 m) measured in meters would directly give the distance in meters covered in one revolution of the wheel, or in other words it can be said that the circumference of the wheel gives MPR (Meters Per Revolution). Hence dividing velocity with this MPR will give us the value of wheel revolutions per second, further multiplying it with 60 will give the wheel RPM as follows. Hence by the use of equation 10 the rpm at wheel at any velocity can be measured.

$$N_W = \frac{v \times 60}{MPR} \dots\dots\dots\dots (10)$$

C. Calculation of torque at wheels τ_W

After the calculation of power and angular velocity at the wheels, it is now possible to calculate the torque required at wheels at various speeds and power requirements. The general formula used for torque calculation is as follows,

$$\tau_W = 30\,\frac{P_w}{\pi N_W} \dots\dots\dots (11)$$

The torque power and velocity curve determined is shown in fig. 4 below

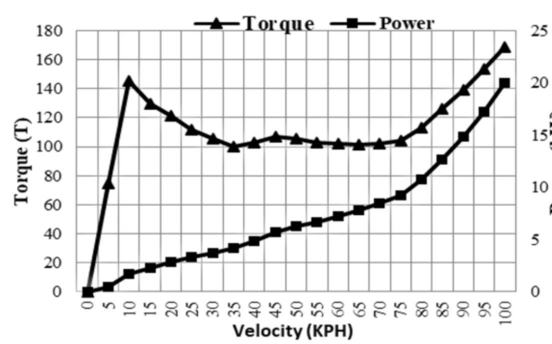

Fig. 4. Torque, Power and Velocity Curve of Wheels*

D. Calculation of the parameters at PGS

The power, torque and rpm going at the wheels are provided by the PGS components through the means of differential or may be any gear reduction depending upon the application. In the case of our PUV, it is through the differential. The differential gear ratio (D.G.R) is 2.47. So, the parameters calculated at the wheel are the multiples or dividends of this ratio. i.e. rpm would be the multiple and torque would be the dividends, while the power transmission is the same (a part from the losses depending upon transmission efficiencies η). The general representations would be,

$$N_{PGS,Out} = D.G.R \times N_W \ldots\ldots\ldots (12)$$

$$N_{PGS,Out} = \frac{\tau_W}{D.G.R} \ldots\ldots\ldots\ldots\ldots (13)$$

$$\tau_{PGS,Out} = \frac{P_w}{\eta} \ldots\ldots\ldots\ldots\ldots\ldots (14)$$

Once the required parameters at the PGS output are known, then the PGS gear ratios are needed to be known in order to have a set of equations defining the dependency of the angular velocities of the respective components of PGS (ring, sun and arm carrier) on the final output of the PGS. Hence a control system can be designed to match the required values at the PGS power output adjoining from the multiple power sources. The gear ratios are as follows,

$$G.R = -\frac{N_{driver}}{N_{driven}} \ldots\ldots\ldots (15)$$

This is for the externally meshing gears and for internal mesh it would be,

$$G.R = \frac{N_{driver}}{N_{driven}} \ldots\ldots\ldots (16)$$

In our case of the compound train (PGS) it would be the product of the gear ratios of the multiple gears in mesh as an example a case of power transmission from the engine to the ring gear power transmission would acquire the following path,

Engine (arm) → pinion gear → EMG2 (ring)

Hence the gear ratio would be as follows,

$$G.R_{E \overset{\Delta}{\rightarrow} R} = \frac{N_{Engine}}{N_{Pinion}} \times \frac{N_{Pinion}}{N_{Ring}} \ldots\ldots (17)$$

This same way the other gearing ratios for all combos are also calculated. The generalized equations excluded from the above equations, shown below (eqn. 18), becomes the governing equation for the rotational motion of gears.

$$\beta N_A - \alpha N_R - N_S = 0 \ldots\ldots\ldots (18)$$

Whereas β and α are calculated as 3.6 and 2.6 respectively in section V. The torque values are also needed to be calculated by the use of following equations.

$$\tau_{PSD,Out} = \tau_{Ring} \ldots\ldots\ldots\ldots (19)$$

This equation gives the total torque output at PGS in electric only mode. While the total output torque at PGS in combined mode would be governed by equation 20.

$$\tau_{PGS,Out} = \tau_{Ring} + \tau_{Arm \rightarrow Ring} \ldots (20)$$

The engine assists the vehicle in driving at high power by providing supplementary torque to the ring gear, and the part of the torque sent to ring gear is governed by eqn. 21

$$\tau_{Arm \rightarrow Ring} = \gamma\, \tau_{Arm} \ldots\ldots (21)$$

The torque from the engine splits up into two values governed by equations 21-24. A part of which is sent to the ring gear as the assisting torque while the remaining goes to the sun gear for charging the battery/s though EMG1.

$$\tau_{Arm} = \tau_{Arm \rightarrow Sun} + \tau_{Arm \rightarrow Ring} \ldots\ldots (22)$$

$$\tau_{Arm} = (1-\gamma)\tau_{Arm} + \gamma\, \tau_{Arm} \ldots\ldots (23)$$

$$\tau_{Arm \rightarrow Sun} = (1-\gamma)\tau_{Arm} \ldots\ldots\ldots (24)$$

E. Graphical representations of parameters

The graphs for the power, torque and rpm corresponding to different velocities and accelerations are given in figure 5 and figure 6 below.

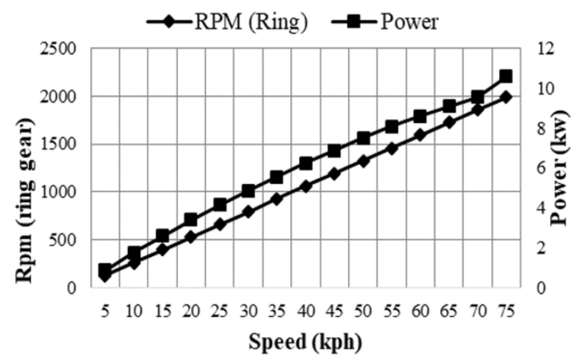

Fig. 5. Speed Power and RPM Curve*

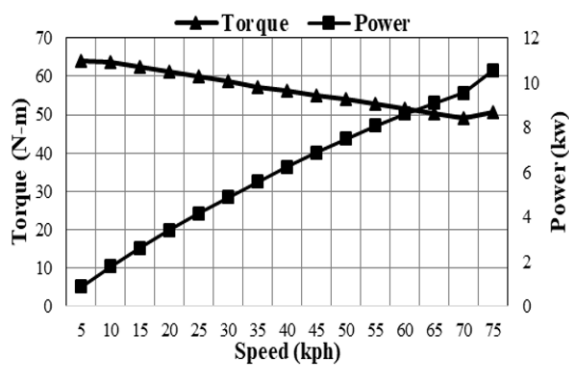

Fig. 6. Speed Power and Torque Curve*

These are the curves when the vehicle is not climbing any slope, but at the slope the acceleration is reduced with sudden increase in power and torque demands at reduced speeds, so the curves deduced from the calculations with a slope of 4 degrees are shown below in figure 7,

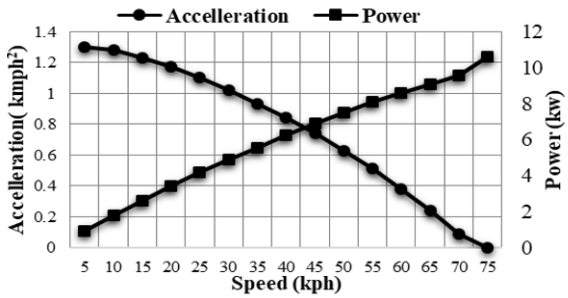

Fig. 7. Speed Power and Acceleration Curve at Slope*

The motor curve is also proposed in figure 8, after knowing the power and torque requirements at various angular velocities and modes.

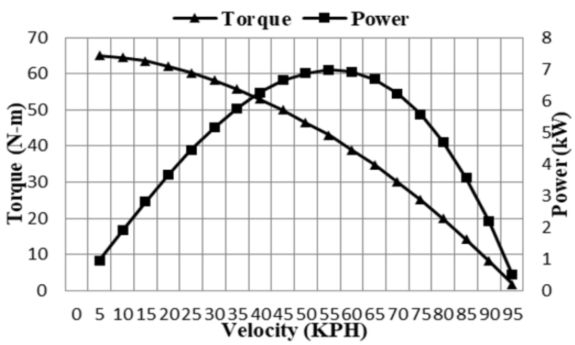

Fig. 8. Proposed Motor Performance Curves*

VII. Verification of Results Through Software Simulations

The CAD model for the PGS is made in PTC CREO as shown in fig. 9, according to the gear parameters calculated.

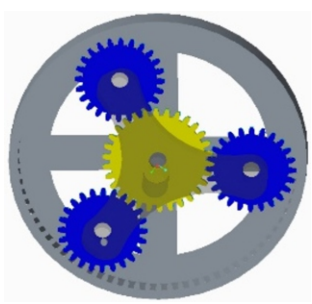

Fig. 9. PGS Creo Model*

A. Verification at electric mode

At first the angular velocities were checked to be in accordance with the calculated values once after given the different angular velocity inputs by motors and engine to check the resultant angular velocities. At electric mode the output velocity at the ring gear would be the same as the velocity of EMG2, as it is the only power source at electric only mode which (for testing purpose) is taken as 846 deg/sec i.e. 141 rpm as shown in figure 10.

when only motor (MG2) is supplying power					
Inputs		Outputs			
Gear	Deg/Sec	MG1(Sun)	Pinnion	Engine(Arm)	MG2(Ring)
MG1(Sun)	0	0	0	0	0
MG 2(Ring)	846	-2199.6	2749.5	0	846
Engine(Arm)	0	0	0	0	0
		-2199.6	2749.5	0	846

Fig. 10. Ang. Velocity Calculation (Electric Mode)*

It can be seen that only MG2 is providing input power shown in yellow and MG1 rotates in opposite direction, hence both the angular velocities are verified through simulations as shown in figure 11.

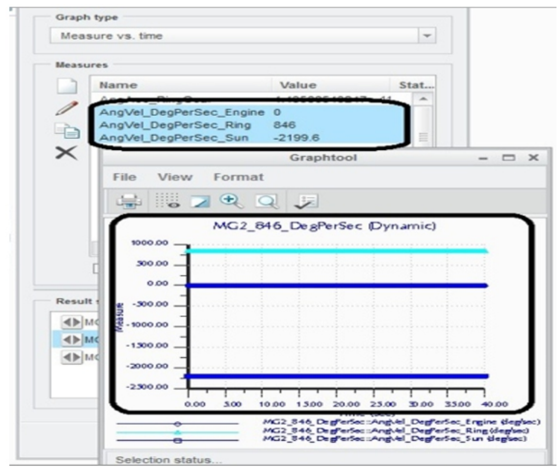

Fig. 11. Verification of Results at Electric Mode*

B. Verification at combined mode

EMGI works as a starter motor for starting the engine. Once the engine is kicked in then the power supply to EMG1 from the battery is stopped, and then engine runs EMG1 to make it work as a generator for charging the battery/s. Normally the engine rotates a bit faster than EMG2. For testing purpose and to have a good visualization in the test run in simulations, the engine is given the speed of 1000deg/sec (166.667 rpm), though it rotates much faster than this even at idle speed. While EMG2 again is given the same angular velocity of 846 deg/sec (141 rpm). So the values are calculated as shown in figure 12. While the simulations results for verification of velocities and ratios is also shown in figure 13.

Torques						
Inputs			Outputs			
Sr. #	Gear	Torque	MG1(Sun)	Pinnion	Engine(Arm)	MG2(Ring)
1	MG1(Sun)	5.5	0	-4.4	20	-14.4
2	MG 2(Ring)	35.0	0	10.7	-48.4	35.0
3	Engine(Arm)	20	5.5	-4.4	20	14.4
			5.5	1.88	20	49.4

Fig.12. Ang. Velocity Calculations at Combined Mode*

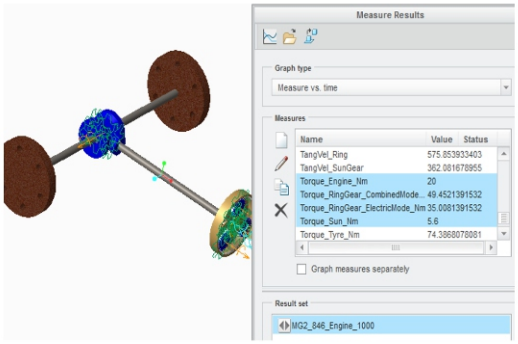

Fig. 13. Verification of Results at Combined Mode*

C. Verification of torque values

Once after the verification of angular velocities the gear ratios are confirmed and torque values are then defined, calculated and verified again through simulations. After knowing the total torque and traction force required at the PGS output i.e. ring gear, the ring gear parameter is also calculated and then by the use of gear ratios the module and the other gearing diameters are calculated. The torque values calculated are as follows in figure 14,

Torques						
Inputs			Outputs			
Sr. #	Gear	Torque	MG1(Sun)	Pinnion	Engine(Arm)	MG2(Ring)
1	MG1(Sun)	5.5	0	-4.4	20	-14.4
2	MG 2(Ring)	35.0	0	10.7	-48.4	35.0
3	Engine(Arm)	20	5.5	-4.4	20	14.4
			5.5	1.88	20	49.4

Fig. 14. Calculated Torques at Combined Mode.*

These values are verified through simulations as shown in the figure 15.

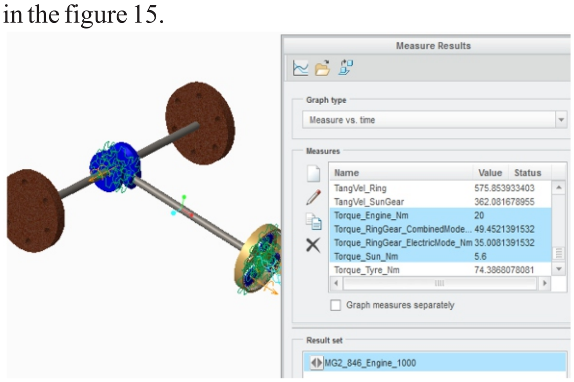

Fig. 15. Simulation Results For Torques*

VII. CONCLUSION

The motive of the paper is to design the powertrain for a small hybrid vehicle, as in Pakistan hybrid technology is not being implemented on smaller vehicles. So as a result of this effort a software tool has been made for the purpose, and another feature is that it's a flexible design that can be applied to design hybrids of any size, at any rate of hybridization.). The other important purpose served by this PGS is that it replaces the manual transmission and works as electronically Controlled Variable Drive (e-CVT). The engine performance curves of the vehicle under study have been taken from the manufacturer and are permitted to be used for this research work. So, using these curves and the curves (graphs) generated in the software are used to propose the motor curves for required torque, RPM, and power for the required vehicle and acceleration, along with the threshold speed and acceleration at which the engine needs to be kicked in. i.e. the engine will kick in at the time when the torque required to drive the vehicle will exceed the limit torque of the motor.

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