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Design, Development and Validation of Engine Dynamometer for FSAE

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Abstract

The goal of this paper is to develop a low-cost yet accurate dynamometer for testing and validating the powertrain. With ever-advancing technologies, and zeal to develop more and more power and torque, there is no definitive limit for the operating boundary conditions. So, to set the definitive boundary conditions for our project, powertrain for Formula Society of Automotive Engineers (FSAE) applications is selected. The brake torque is calculated based upon the traction limit of FSAE race cars. The brake torque is determined to be 100 N-m. The goal is to develop a low-cost accurate dynamometer, thus hydraulic type, Steady-state dynamometer is considered. A top-down approach is considered for the project. With the determination of brake torque, the gear pump selection is done which is followed by pressure gauge selection, needle valve selection, design of final drive, which is followed by hoses and couplings. The RPM data is tapped from the ECU of the prime mover and the flow rate is computed. A pressure sensor is used to measure the pressure downstream of the gear pump. The data is then computed to obtain power and torque curves. Then the whole setup is tested and validated with FSAE standard powertrain

Keywords: FSAE, Dyno, Dynamometer, Powertrain

1. Introduction

Engine dynamometer is being used for a long time now to measure the power of an engine by applying a load to the tested engine. There are several types of engine dynamometer such as hydraulic, eddy current, water brake, electric, rope brake and dry friction type [1]. The very basic engine dynamometer design is the hydraulic engine dynamometer in which the engine is coupled to the hydraulic pump. The load being given to the engine is then controlled by adjusting the flow control valve that is responsible to change back pressure on the hydraulic pump. The eddy-current type is the most common one of the all in which an electromagnetic load device is used to vary the load on the engine. By using a control system the load that has to be given to the engine is adjusted in order to measure the engine performance [2,4]. This makes eddy-current type engine dynamometer more expensive to be designed and fabricated as compared to the hydraulic and friction types. For hydraulic type engine dynamometer, the hydraulic pump is connected to the crankshaft of the engine [3]. The load to the engine is applied by using a control valve to give a backpressure to the hydraulic system. In this system, the pump rotation will generate a fluid pressure to the system.

For selection of type of dynamometer various dynamometers like Electric, Rope Brake, Water Brake, Eddy current and Hydraulic were compared in terms of low cost, low response time, high power absorption and low inertia. It was observed that only hydraulic type dynamometers fulfilled all the criteria for the dynamometers.

Even though safety concerns caused due to high operating pressure of Hydraulic type dynamometers, **Hydraulic type dynamometer** was selected to be used after engineering solution for safety purposes. Figure 1 represents the layout of the hydraulic type dynamometer. In this, the absorber i.e. Gear Pump in this case is mated to prime mover, engine in this case. Upstream of the gear pump is pressure transducer to measure the pressure in the line. It is followed by pressure relief valve which is followed by flow control valve, which in this case is a needle valve. The orifice of the needle valve is changed to change the flow rate and essentially the pressure is changed. Upstream the gear pump, oil reservoir is present which stored the hydraulic fluid and houses the oil strainer. Oil strainer essentially works to inhibit the entry of foreign unwanted particles in the hydraulic circuit.

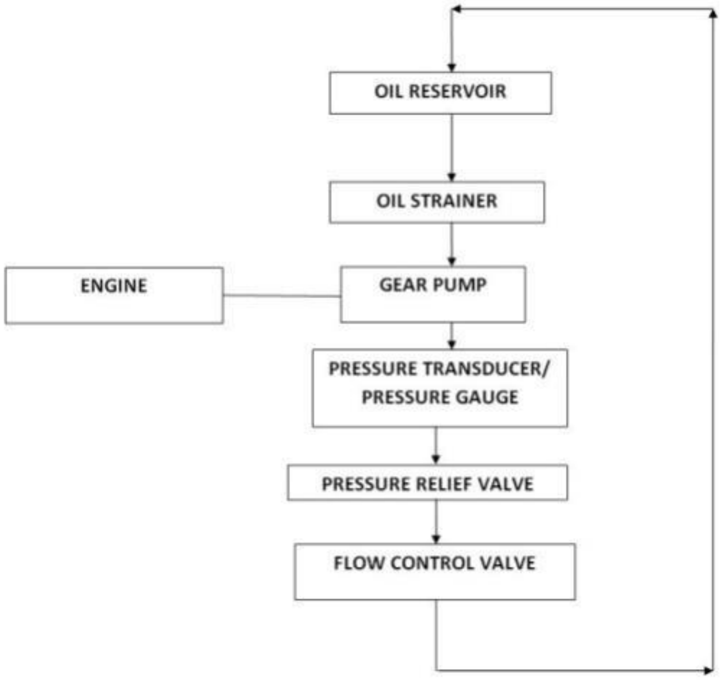

Figure 1 – Layout of the dynamometer.

2. Material Selection

Gear pump selection: If power produced by the engine is more than the power rating of the absorber, then the engine speed will increase uncontrollably. It was decided that power rating will be the first criteria which will be checked when selecting an absorber. We referred to Mark Baldisserotto and George J. Delagrammatikas' research paper and found out the amount of power produced by a formula student team's engine. It was found out to be 100 HP. With selection criteria of brake power and brake torque set, Various gear pumps were compared on the basis of specifications provided.(Table 1) Even though gear pump OP-330 satisfied the needs of brake power of 100 HP, OP-380 gear pump was used considering its lower peak operating pressure.

Transmission Design- The major problem in the experimental setup was to transmit the power of the engine to the Gear Pump efficiently and cost effectively. For that we selected a chain drive over a Belt or Gear Drive, chain drive being more efficient than Belt drive and less expensive than Gear Drive in terms of cost.

Table 1: absorber selection criteria

Gear pump selection									
S.NO	Minimum cc/rev of pump i.e. at maximum pressure	Maximum RPM	Flow rate (cc/min)	Minimum flowrate of pump i.e. at max pressure (m^3/sec)	Maximum pump pressure (KPa)	Power dissipated by pump (w)	Power dissipated by pump (HP)	Name of the pump	remarks (min HP100)
1	20	3500	70000	0.0011666	28000	32666.666	43.806	GHP 3 SERIES GROUP3 GEAR PUMP	Not Sufficient
2	87	2800	243600	0.00406	24000	97440	130.669	ALP 4 SERIES GROUP4 GEAR PUMP	Okay
3	45.33	2500	113325	0.00188875	22000	41552.5	55.722	OP-150	Not Sufficient
4	54.33	2500	135825	0.00226375	22000	49802.5	66.786	OP-180	Not Sufficient
5	63.67	2500	159175	0.002652917	22000	58364.16	78.267	OP-210	Not Sufficient
6	75.67	2500	189175	0.003152917	22000	69364.16	93.018	OP-250	Not Sufficient
7	90.67	2500	226675	0.003777917	19000	71780.41	96.259	OP-300	Not Sufficient
8	100	2500	250000	0.004166667	18000	75000	100.576	OP-330	Not Sufficient
9	115.33	2500	288325	0.004805417	16000	76886.66	103.106	OP-380	Okay
10	151.33	2500	378325	0.0063054	16000	100886.6	135.291	OP-500	Okay

Sprocket Design:

Below given data were taken in to consideration from the manufacturers' as input for the reduction ratio calculation (Table 2):

Table 2 – Input Data

Gear Pump Max. Operating RPM	2,500
Gear Pump Min. Operating RPM	700
Engine Max. Running RPM	10,000
Engine Min. Running RPM	2,400

Primary reduction is the first gear reduction in the engine at the crank shaft, which is followed by secondary reduction that are the gearing ratios of the sample KTM 390 engine used. The final drive ratio (FDR) of 1.9 was calculated. The driver sprocket and the driven would be made out of EN24. The following analysis was carried on the driven sprocket. Since the calculated design load on for the chain was 9124.89N, the same force when multiplied with the driven sprocket's radius (.084m) gave a moment of 1313.708Nm. This was used for the analysis. Then selecting 23 teeth profiles (angle of contact for the sprocket is 209.73°), the moment is then distributed. Sprocket design (Table 3) was completed and the CAD model was created.

Table 3 - Reduction ratio selection criteria

Required reduction ratio for gear pump											
FDR	Pump Max RPM	Pump Min RPM	Engine minimum RPM	Engine maximum RPM	Primary reduction	Secondary reduction	Minimum Engine output shaft rpm	Maximum engine output shaft rpm	Minimum Gear pum p shaft RP M	Maximum Gear pum p shaft RP M	Status (OK/ Not OK)
1.9	2500	700	1500	10000	0.375	0.372	209.25	1395	110.1315	734.2105	Oka y
			1500	10000	0.375	0.5384	302.8846	2019.23	159.4129	1062.753	Oka y
			1500	10000	0.375	0.7037	395.8333	2638.888	208.3333	1388.888	Oka y
			1500	10000	0.375	0.875	492.1875	3281.25	259.0460	1726.973	Oka y
			1500	10000	0.375	1.0454	588.0681	3920.454	309.5095	2063.397	Oka y
			1500	10000	0.375	1.1904	669.64286	4464.285	352.4436	2349.624	Oka y

Connecting shaft:

The drive-shafts are the link between the Pillow blocks which will connect the Driver and Driven Sprockets forming the reduction ratio of 1.9. They also act as a torsion spring that filters torque peaks out of the drive-line. The diameter of driveshaft was chosen by analyzing the torque it will transfer. The material taken was EN24. The length of connecting shaft was taking approximately depending upon the jig that was designed to hold the gear pump and attach the Pillow block.

Coupling:

The coupling was designed to attach the connecting shaft and Gear Pump input shaft with keyways. The input shaft keyway was given by the gear pump supplier and the connecting shaft key-way was designed according to the acting torsional load.

2.3 Peripherial Design/ Selection.

A Printz pressure transducer with an operating pressure range of 0-250 BAR and temperature range of 0-125°C was selected for pressure sensing purposes.

For data acquisition an open source microcontroller Arduino Micro was used to collect and process the data. The calibration data of the pressure sensor was fed to the microcontroller using Arduino studio.

Table 4: Plumbing Selection:

Hydraulic oil	ISO grade68 (recommended by the catalogue)
Oil strainer	2 inch
Direct acting pressure relief valve	½ Inch 250 bar rated
Flow control valve/ Needle valve	1 inch 500 bar rated
Inlet hose	2 inch 40 bar rated
Outlet hose	1 inch 50 bar rated
Return hose	½ inch 250 bar rated

3. Testing

A customized testing procedure was developed to test our in-house made hydraulic dynamometer which would prove that the dynamometer is working as expected and generating the Power and Torque value's of the running KTM 390 engine. It was planned to test the Dynamometer by part throttles, i.e.at different engine throttle angles which will give Power and Torque at different RPMs for the same fixed Engine throttle angle. These throttle angles included 20%, 50% and 100%throttle. The throttle angles were set by observing Engine Control Unit's data and the engine was loaded with the help of the needle valve. Different graphical data was collected and the final graphical data, that is at 100% throttle was taken and compared with the graphical data that the same engine gave when it was taken to a commercial dynamometer and was tested at 100% throttle, i.e. at wide open throttle. Test setup shown in figure 2.

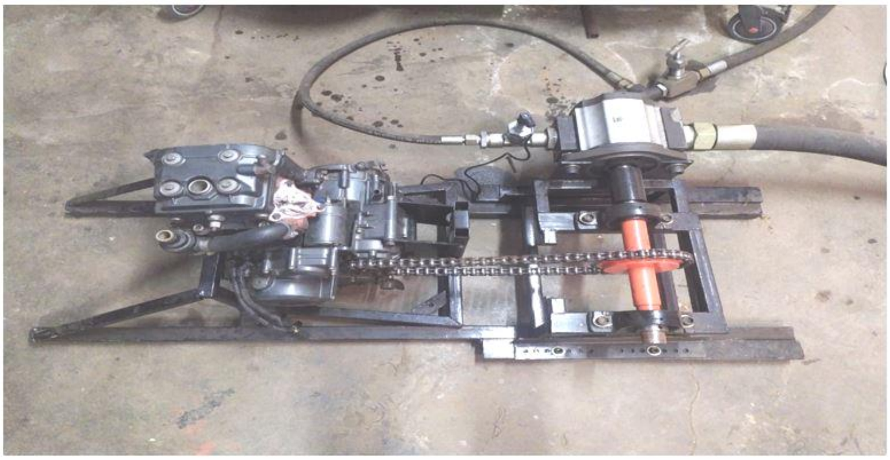

Figure 2 – In-house manufactured hydraulic type engine Dynamometer.

4. Results and Validation

The testing carried was tabulated as: (Table 5)

Table 5 - Test results on 100% throttle (full load)

100 % THROTTLE (FULL LOAD)								
S.No.	Engine RPM	Pump RPM	Pressure (Bar)	Theoretical flow rate = (115.33 * 10 ^ (-6) * pump RPM /60)	Corrected flow rate = (theoreticalflow rate * correction factor)	Power produced (W) = (pressure in Pascal * corrected flow rate)	Power produced (H.P.)	Torque produced(Nm) = (power in KW / enginerpm / 2 / 3.14 * 60)
1	4000	297.538	24	0.0005719	0.0004861	11667.137	15.639	27.867
2	5000	371.923	23	0.0007148	0.0006076	13976.258	18.734	26.706
3	6000	446.307	21	0.0008578	0.0007291	15313.118	20.526	24.383
4	7000	520.692	27	0.0010008	0.0008507	22969.677	30.790	31.350
5	8000	595.076	30	0.0011438	0.0009722	29167.844	39.098	34.834
6	9000	669.461	31	0.0012868	0.0010937	33907.619	45.452	35.995
7	10000	743.846	29	0.0014297	0.0012153	35244.478	47.244	33.673

For validation the same test as 100% Throttle (Full Load) conditions were carried out with a commercial dynamometer which is an Inertial Dynamometer with has a huge drum with a measured mass that is being rotated by the Engine and the Moment of Inertia that is being generated by the drum is then multiplied by the angular acceleration of the drum to give the torque at different engine RPM values (Table6).

Table 6: test results on 100% throttle (full load) for inertial dynamometer

100 % THROTTLE (FULL LOAD) INERTIALDYNAMOMETER			
S.No.	Engine RPM	Power produced(H.P.)	Torque produced (Nm)
1	4000	16.3	29
2	5000	20.2	28.8
3	6000	22.6	26.8
4	7000	28.6	29.1
5	8000	37.4	33.3
6	9000	42.1	33.3
7	10000	47	33.5

Now data comparison was carried out between the readings of the two tests to validate the engine dynamometer test results and to find out the Mean Deviation from the true value (if any). The results were again tabulated and plotted as (Table 7and Figure 3, 4)

Table 7: Inertial dynamometer comparison with hydraulic dynamometer

COMPARISION BETWEEN INERTIAL AND HYDRAULLICDYNAMOMETER AT FULL LOAD					
S. No	RPM	Power value from inertial dynamometer(HP)	Power value from hydraulic dynamometer (HP)	DEVIATION PERCENTAGE(%)	DEVIATION PERCENTAGE(%)
1	4000	16.3	15.639	4.051	4.051
2	5000	20.2	18.734	7.252	7.252
3	6000	22.6	24	-6.194	6.194
4	7000	28.6	30	-4.895	4.895
5	8000	37.4	39.098	-4.542	4.542
6	9000	42.1	45.452	-7.963	7.963
7	10000	47	47.244	-0.520	0.520
MEAN DEVIATION					5.060

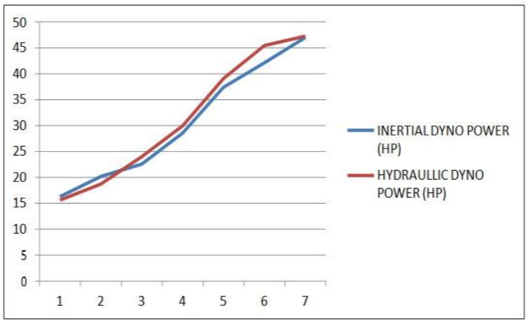

Figure 3 – Comparative power curves of inertial and hydraulic dynamometers.

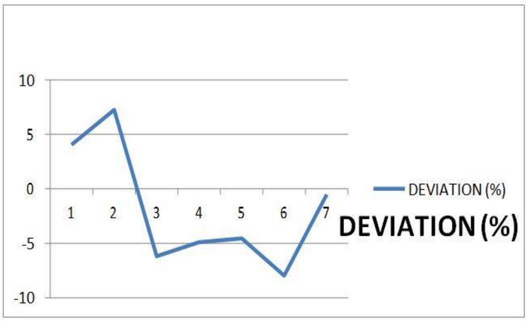

Figure 4 - Percentage deviation between commercial dynamometer and hydraulic dynamometer.

On comparison of data from a commercial dynamometer, mean deviation of power was found to be 5 percent. The test results were conclusive, definitive and repeatable with the mean deviation of 5.060% which is well in the acceptable limits. Moreover it was observed that the trend followed in the graph from the commercial dynamometer was replicated by our Hydraulic type dynamometer.

5. Conclusion

In this project we fulfilled our objective and developed a dynamometer which was tested and results were plotted. Results of this dynamometer with 100% throttle were compared with a commercial inertial dynamometer with 100% throttle and the mean deviation was found out to be 5.060%, which is within the accepted range. The overall cost of this dynamometer was only INR 45,500 which when compared with price of imported dynamometer which costs INR 3,75,000, is almost 10 times cheaper to manufacture. This reduction in cost was achieved by using a hydraulic absorber, which is cheaper yet accurate, and by eliminating the expensive features which the imported dynamometer offers and are not necessary for formula student application. Hence, we believe that we succeeded in our objective to offer the motorsport community especially formula student with economically affordable solution powertrain development.

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