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Design and Analysis of Four Cylinder Diesel Engine Balancer Shaft

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***Abstract*— Noise, Vibration and Harshness (NVH) are the major issues in today's modern diesel engine. The reciprocating movement of the piston and connecting rod, combined with the rotating movement of crankshaft generates inertial forces that act on the engine block and cause it to vibrate in different planes. An important key in the working engine is the first-order and second-order inertial forces which cause high amplitude excitation of vehicle structure and leads to interior noise and riding discomfort to the driver. In our study of four cylinder engine configuration, secondary inertial forces are the main cause of vibration acting in vertical direction. This paper focuses on the elimination of secondary inertial forces by using twin balancer shaft rotating in opposite direction at twice the crankshaft speed. A 100hp four cylinder in-line diesel tractor engine is used in our study where the modeling of balancer shaft is done using *UNIGRAPHICS*. FEA using *ANSYS* is performed to check for its maximum stress developed on the balancer shaft and also modal analysis is carried out to check for its resonant frequency. Finally the multi-body dynamics using *ADAMS* is performed resulting in 95% secondary balancing.**

Keywords*— NVH, secondary inertia forces, balancer shaft, acceleration values, ADAMS

I. INTRODUCTION

Modern diesel engines are widely used in cars which offer its customers several advantages like, performance, safety and ride comfort. These advantages are also required to the tractor where the farmer has to work continuously for hours who requires comfort, noise and vibration free environment transferring to its compartments and it is where the engine plays a vital role in the customer overall satisfaction. As the demands of customer on the performances of a car have become higher, the tractor manufacturers desire to enhance the comfort level as well as the basic driving performances in the off-road application has also become higher. Today the noise and vibration has become a serious issue in the vehicle which is generated by the excitations from the ground and the engine. Although most excitations from the ground are eliminated due to vehicles' stiff structure, the engine itself now remains the source of noise and vibration. The noise and vibration in engine are generated by the combustion impulses produced within the cylinders and the inertial forces due to the reciprocating and rotating motions of the piston and crankshaft exciting the components such as the engine block and cylinder head. This vibration of the engine structure and the dynamic properties of the crankshaft deteriorate the vibration level and sound quality. Therefore, alternatives are required to improve the NVH characteristics of the engine.

Alexandre Augusto Riginik Ferreira, Caetano Calviti, Frederico Barbieri and Celso Argachoy [1], described the balancer shaft development for the In-line four cylinder high speed diesel engine, as well as major design alternatives and development process and issues by using design/decision matrix methodology which can be applied to any design or engineering case helps design engineers to make the right decision amongst different options by using a very simple and objective matrix. Kwon-Hee Suh, Yoon-Ki Lee and Hi-Seak Yoon [2], described the three cylinder engine in order to eliminate the primary out-of balance by using single balancer shaft. A real time analysis was performed with and without balancer shaft which showed the frequency reduction of 26.3%, 6.1% and 14.9% at 1500 rpm-full load condition, 4000 rpm-full load condition and 7000 rpm-no load conditions respectively. David Meek and Martyn Roberts [3] described Balancer shaft conversion for four cylinder engine which shows the 92.5% balancing of secondary forces along with reduction in cabin noise and engine weight. Sagar Sanone and Amit Chaudhari [4] described the four stroke single cylinder engine to eliminate the primary unbalance forces by using single balancer shaft which shows the net vibration reduction on crankcase and further to passenger via chassis. The vibration reduction of 68.77% was achieved at rated speed of 3600 rpm and 64.88% at lower speed of 2200 rpm.

In this paper, study on balancer shaft for four cylinder engine configuration is done to achieve maximum percent balancing of secondary inertial forces according to the packaging constraint inside the engine block. The inertial properties are extracted from the 3D-Model and multi-body dynamics is constructed. The combustion pressure in the current study is neglected. The acceleration values with and without balancer shaft are evaluated mainly at three operating conditions, 1400 rpm-high torque, 2200 rpm-rated speed and 2750 rpm-full speed.

II. LITERATURE SURVEY

Engine vibrations are divided into two modes:

- Vibration due to rotating and reciprocating inertial forces.
- Torsional vibration of the crankshaft due to combustion impulses [5].

The up and down movement of the reciprocating masses constantly accelerate and decelerate which creates the inertial forces within the cylinder. These forces are then transferred to the engine block, cylinder head creating vibration in different planes. The inertial forces are given by the equation:

$$F_{inertial}(N) = m\omega^2 r\cos\theta \ \pm m\omega^2 r\frac{\cos 2\theta}{n}$$

m= mass of reciprocating components; r= crank radius; θ= crank angle measured from top dead center (TDC); ω= angular velocity of crankshaft; n= ratio of length of connecting rod to crank radius.

The primary (1st order) forces is the inertial force produced by the piston mass due to rotating crankpin's projected movement along the line of stroke being relayed to the piston via connecting-rod [5]. It is represented by the first term in the above equation; $F_{prim}=m\omega^2 r\cos\theta$ and its maximum values occur twice per revolution i.e. at 0° and 180°.

The secondary (2nd order) forces is the inertial forces produced by the piston mass due to rotating crankpin's outward and inward projected movement perpendicular to the line of stroke relaying motion to the piston via inclined connecting-rod. It is represented by the second term in the above equation; $F_{secon}=m\omega^2 r\cos 2\theta/n$ and its maximum values occur four times per revolution i.e. at 0°, 90°, 180° and 270°.

"When the crankpin rotates its projected movement for each crank-angle position, both along the line of stroke and perpendicular to the line of stroke occurs simultaneously, this then causes a resultant acceleration and deceleration motion to the piston and hence a corresponding inertia force will be created by the reciprocating piston mass" [5].

Due to our four cylinder engine configuration, the primary forces get cancelled out, remaining with the secondary forces. Secondary inertial forces are caused due to the obliquity of the connecting-rod, because it neither has a pure rotating nor pure reciprocating movement so when the piston goes from TDC to mid-stroke, it travels faster and when it goes from mid-stroke to BDC, it travels slower and the process continues. So due to this faster and slower movement of reciprocating masses, inertial forces are produced. From the above equation, connecting rod to crank-radius ratio (*n*) plays a crucial role as if the length of the connecting rod increases, the inertial forces decreases. So the length of the connecting-rod should be bigger as much as possible in order to minimize the inertial forces. Secondary forces basically act in two directions:

- Secondary forces acting vertically

Up and Down movement of the reciprocating mass gives rise to upward and downward inertial forces which vary periodically twice per crankshaft revolution and cause engine to vibrate in vertical direction.

- Secondary forces acting laterally

When the piston travels from TDC to BDC, it is pressed against the cylinder wall due to pulling action of the connecting rod relative to the cylinder, and at higher speed, it goes much higher and same phenomenon occurs when it travels from BDC to TDC. This sideways forces varies periodically twice per crankshaft revolution and cause the engine to vibrate in lateral direction. But in the present engine configuration, these lateral forces almost cancelled out [7].

So only remaining is the secondary inertial force acting in the vertical direction which is required to be eliminated. One way of eliminating the vibration is by making the vehicles' structure stiffer so that it absorb all frequencies of vibration, but it can leads to the components/structure heavier and expensive. Another way the vibration can be attenuated is by using a two balancer shaft rotating in the opposite direction and at twice the speed of crankshaft which can reduce engine vibration level transferring to the engine mountings.

A balancer shaft is basically a rotating mechanical component having eccentric masses which restrict the vibrations produced due to the reciprocating mass by exciting the counter-forces of the same magnitude in the opposite direction. The balancer shaft was first designed by British automotive engineering master Dr. Frederick Lanchester in the early 20th century. They are specially designed to eliminate vibration caused due to primary and secondary inertial force and couples. It is necessary to use the two balancer shafts instead of one because rotating two shafts in opposite directions will cancel their lateral forces resulting in the remaining vertical force which will be used to balance the secondary inertial forces producing in vertical direction. They are mostly located on both sides of the crankshaft at equal distance having eccentric masses producing the centrifugal force counteracting the inertial forces.

III. DESIGN AND ANALYSIS METHODOLOGY

The specification for the selected four cylinders engine is as follows:-

TABLE I. Engine Specification

Engine type	In-line
Bore x Stroke	98 mm x 122 mm
No. of cylinders	4
Total displacement	3.68 litre
Rated engine speed	2200 rpm
Power	100hp at 2200 rpm
n (L/r)	3.04

A. Calculation for Secondary Inertial Forces

Since primary forces and couples cancel out, it is required to calculate only secondary inertial forces.

The secondary forces produced in the four cylinder engine can be calculated using equation:-

$$F_{Secon} = m\omega^2 r\frac{\cos 2\theta}{n}$$

The maximum value of secondary inertial forces calculated at rated rpm is 8036 N.

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B. Balancing of Secondary Inertial Forces

The Secondary inertial forces can be eliminated by equating the above forces with the counter forces (F_B) that will be produced from the balancer shaft having eccentric mass. It is given by the equation [6]:-

$$m\omega^2 r \frac{\cos 2\theta}{n} = 2M(2\omega)^2\gamma$$

M= weight of eccentric mass; γ= distance from centre of rotation of balancer shaft to centroid of eccentric mass.

The Unbalance value coming from Balancer Shaft is 0.0189kg-mm. This Unbalanced value has to be balance in term of eccentric mass on the balancer shaft by varying the mass properties and diameter.

C. Eccentric mass design

It is decided that the total mass of eccentric mass will be divided into eight equal parts so as to make the centre axis of each pair of masses to the centre axis of the each cylinder same. It became mandatory to design eccentric mass first because it helped us in deciding the maximum diameter of eccentric mass that can be achieve according to the packaging constraint. Without disturbing the stiffness of the block, drill is given inside the block to place the balancer shaft. The material properties are given from which weight of single eccentric mass is calculated as 3.668kg with diameter of 48mm. Below in figure 1 shows the design of eccentric mass.

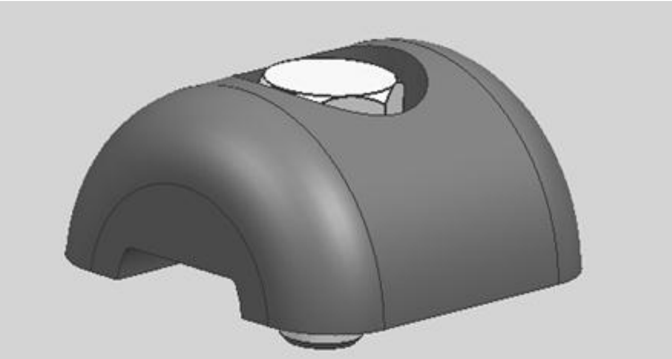

Fig. 1. Design of Eccentric Mass

D. Balancer shaft design

It was decided to place the balancer shaft inside the engine block instead of having a separate unit in order to save purchasing cost. Secondly, the length of the shaft will be along the length of the engine block supported on three bushing so as to match the centre of gravity of engine with balancer shaft to prevent undesirable moments. The following stresses act on the balancer shaft:-

- Stress due to bending and twisting moment.
- Stress due to gear load and self weight of the shaft.

The centrifugal force acting due to eccentric masses is calculated and torque (T) value (obtained from ADAMS) after giving properly meshed gear is applied in the loading diagram. The values for k_m and k_t used for suddenly applied load with shocks in bending and twisting moments is taken as two for each and factor of safety as two.

Hence diameter of balancer shaft calculated using bending and twisting moment equation is 17 mm (higher of above two).

Below in the figure 2 shows the design of balancer shaft which include gears, circlips, thrust plate, bushing and bolted eccentric masses and figure 3 shows the placement of balancer shaft inside engine block.

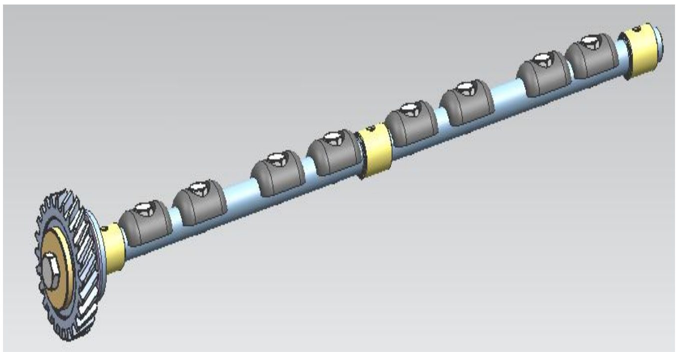

Fig. 2. Balancer Shaft Design

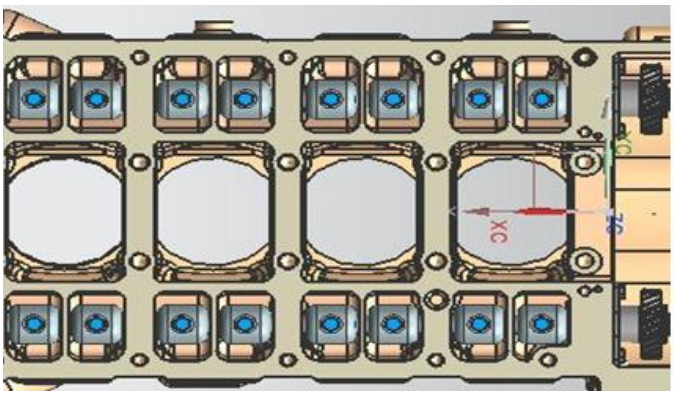

Fig. 3. Balancer Shaft Placement

Below in the figure 4 shows the complete assembly consists of crankshaft, piston assemblies, connecting-rod, transmission gears and balancer shaft.

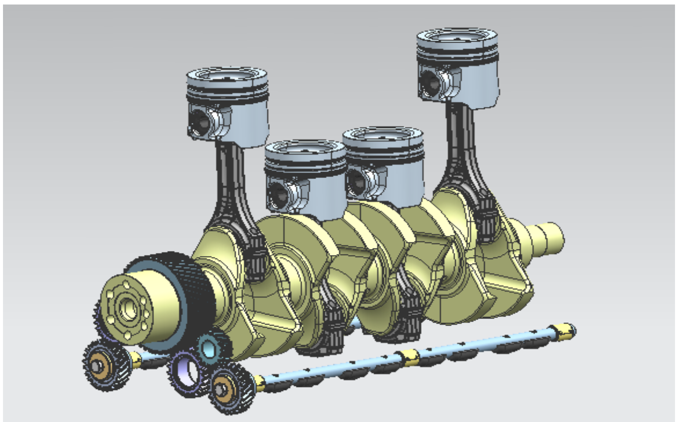

Fig. 4. Balancer Shaft Assembly

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E. Lubrication

Since the shaft is rotating as double the speed of the crankshaft, it is important to supply the right amount of lubrication for the bushing. Below in the figure 5 shows the cut section of assembly with drill holes provided for lubrication. Oil enters from the gallery to all the bearing and will finally drop down in the oil sump. Bush is provided with the holes on the top and groove inside where the oil enters and provide lubrication to the shaft.

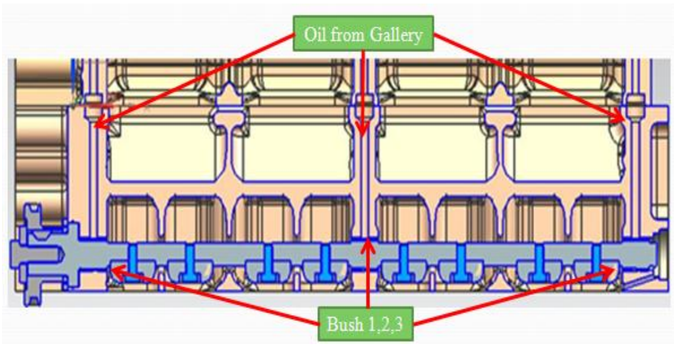

Fig. 5. Cut section of Assembly

IV. RESULTS AND DISCUSSION

A. Finite element analysis

Material properties are applied and finite element analysis using ANSYS is performed to check for maximum stress produced on balancer shaft. Fine triangular mesh is used to obtain better results and boundary conditions are applied to the balancer shaft.

Below in figure 6 shows the balancer shaft where the maximum stress is produced at the center of the shaft whose value is 125.49 MPa which is safe.

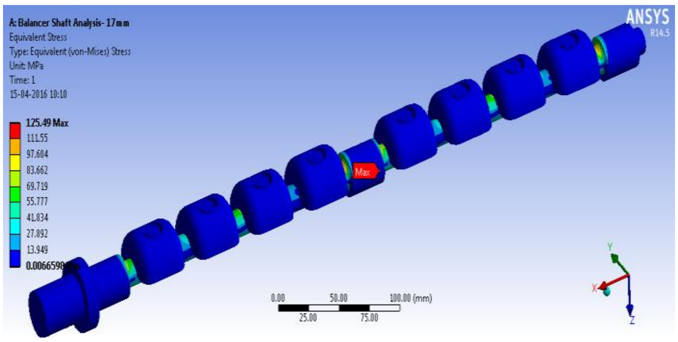

Fig. 6. Von Mises Stress

B. Modal analysis

Modal analysis is used to determine the vibrations characteristic of the structure during excitation i.e. its natural frequencies and mode shapes. Modes of any structure depend upon its material properties which includes masses and stiffness. In our study, six modes are taken into consideration to make sure that the component is reliable. The frequency obtained from the crankshaft is evaluated as 275 Hz.

In the Table II given below, the vibrations frequencies of balancer shaft obtained through ANSYS, which shows that the frequency of the balancer shaft is higher as compared to the frequency generated through crankshaft thus prevents the resonance in the system.

TABLE II. Vibrational Frequencies of Balancer Shaft

MODES	FREQUENCY (Hz)
1	1002.3
2	1010.5
3	1033.5
4	1044.0
5	2155.6
6	2311.7

C. Multi-Body Dynamics

MBD is used to check the dynamic behavior of the interconnected rigid bodies which goes under the rotational and translational displacement. In our study, MBD is important to determine inertial forces which are transmitted to the engine and with the implementation of a balancer shaft, how much the forces can be controlled.

The inertial properties are assigned to all the components and constraints are defined between them. The multi-body dynamics of the four-cylinder engine is performed using ADAMS which consists of an engine block, cylinder head, crankshaft, connecting rod, piston assemblies, flywheel, balance shaft, and an adapter plate.

Below in the figure 7 shows the graph obtained between secondary forces and balancer shaft forces which conclude the 95% of secondary balancing in vertical/z-direction.

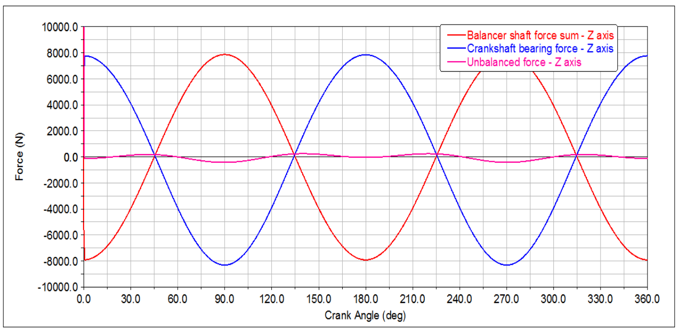

Fig. 7. Secondary Forces vs. Balancer Shaft Forces

D. Acceleration values

Rigid mode analysis of the engine is performed. In this study, the combustion pressure is neglected and crankshaft speed is ramped up to get the inertial forces. Two sets of mechanism are analyzed; one without the balancer shaft and the other with the incorporation of the balancer shaft.

Accelerometer, a flexible bushing element having a stiffness value of $1xe^5$ N/mm is placed at two locations on the crankcase to measure the acceleration values as shown in figure 8. The root mean square (RMS) acceleration values are obtained which shows the drastic change in the vibration in z-direction, whereas in y-direction, it is almost negligible. On the other hand, acceleration values are also been found on x-direction. This happened because of the combined displacement action in y and z-direction leading to displacement in x-directions also.

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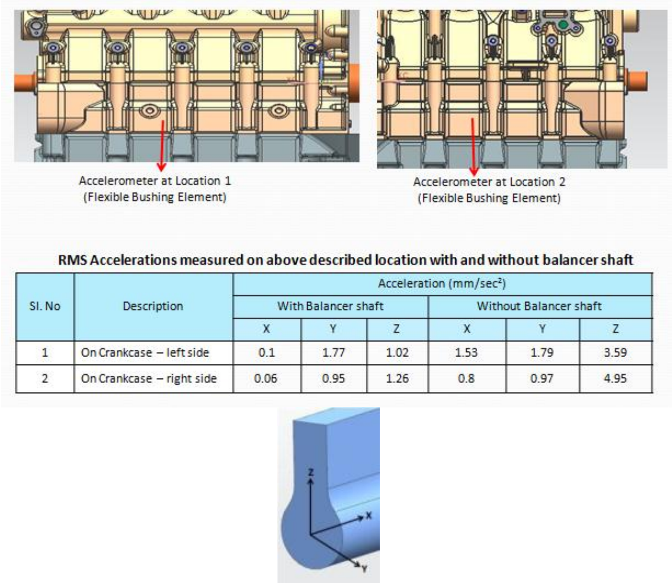

Sl. No	Description	Acceleration (mm/sec²)					
		With Balancer shaft			Without Balancer shaft		
		X	Y	Z	X	Y	Z
1	On Crankcase – left side	0.1	1.77	1.02	1.53	1.79	3.59
2	On Crankcase – right side	0.06	0.95	1.26	0.8	0.97	4.95

Fig.8. Acceleration Values

The values of acceleration on the crankcase at different speeds are simulated and the maximum acceleration value is taken into consideration. Firstly, one set of vibrations simulation check is performed, including no balancer shaft. Secondly, one more vibration check is performed on the engine crankcase including balancer shaft.

The two different values are compared for different speed as given in figure 9 which concludes that with the implementation of the balancer shafts, vibration can effectively be reduced thus providing the riding comfort to the driver.

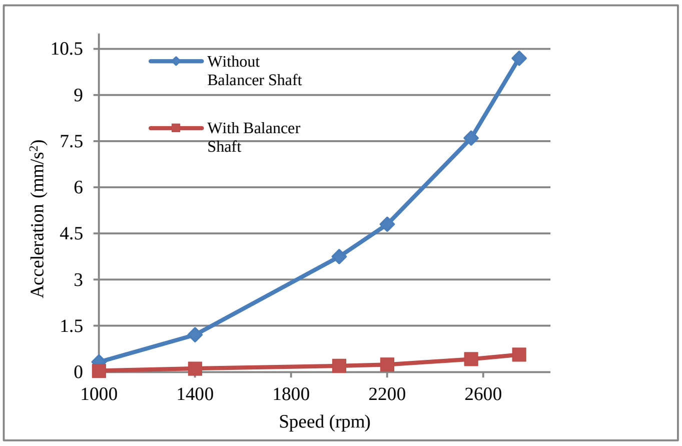

Fig. 9. Comparison of acceleration values with and without balancer shaft

The vibration reduction at rated speed of 2200 rpm is calculated as 95%, whereas at lower speed/high torque and high speed/low torque condition at 1400 rpm and 2750 rpm respectively is calculated as 90.82% and 94.4% as given in table III.

TABLE III. Vibration values with and without balancer shaft.

Speed (rpm)	Without balancer shaft	With balancer shaft
1000	0.325	0.040
1400	1.210	0.111
2000	3.750	0.203
2200	4.800	0.240
2550	7.600	0.420
2750	10.200	0.565

V. CONCLUSIONS

Internal combustion engine on dynamics generates first-order and second-order inertial forces depending upon the configuration. These forces are transmitted through the engine to vehicles' structure and finally to the driver compartment thus providing discomfort. Secondary inertial forces from in-line four cylinder engine are the cause of the vibration and it can be attenuated by installing the twin secondary balancer shaft rotating in opposite directions with twice the speed of the crankshaft. In this study, design and analysis of balancer shaft was performed and vibration level was simulated with and without balancer shaft to check the acceleration values. The following conclusions are made:-

1. Secondary inertial forces are calculated and analytical calculation is done for required balancing.
2. Finite element analysis is performed on the balancer shaft under given load conditions which show that the shaft is safe.
3. Modal analysis is performed for balancer shaft to check for the resonance conditions. Thus, it is concluded that frequency of balancer shaft is much higher than the frequency from engine and shows the shaft is safe.
4. Multi-body dynamics is performed which shows 95% secondary balancing.
5. Simulation of the engine is performed with and without balancer shaft to check the acceleration values at the crankcase. The results show the drastic change in the vibration level by implementation of the balancer shaft.
6. Comparable results are achieved against benchmarking (80.5%).

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