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REVIEW ON INTERNAL COMBUSTION ENGINE VIBRATIONS AND MOUNTINGS

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ABSTRACT
Ride comfort, driving stability and drivability are vital factors in terms of vehicle performance and the customer satisfaction. The power plant (IC engine) is the source for the vibration that reduces the vehicle performance and it need to be controlled to some extent such that the vehicle performance will be improved. The IC engine is made up of reciprocating and rotating parts and they produce unbalanced forces during their operation and produce the vibratory output at the vehicle supporting members. The vibration reduction will be possible by minimizing unbalanced forces and by providing the anti vibration mounts at the engine-vehicle interface. Many researches were made to find the causes for the vibration and to reduce the vibrations at the engine supports. But still there is a research gap on the vibration modeling and vibration isolation of the engine. In this work, an attempt is made to represent the state-of-the- art for the engine vibrations and its isolations and to provide a gate way for the future work on it. It reveals the various work carried out on the engine multibody modeling of the IC engine components and different engine mountsand their orientations. The review is structured as engine multibody modeling, engine vibrations and engine mounting areas and revealed the gaps and untouched parts that requires further research.

KEYWORDS: *Internal combustion(IC) engine, Unbalanced forces, Vibration, Engine mount,*

I. INTRODUCTION

The internal combustion (IC) engine is the concentrated mass in vehicle and if not properly designed it will cause vibrations and transfer to the supporting structures ride comfort, driving stability and drivability are important factors for the performance of a vehicle and are affected by the engine vibrations. Because of the environmental considerations, as well as changes in consumer preferences regarding vibration induced must be reduced. Vibration behavior of an IC engine depends on unbalanced reciprocating and rotating parts, cyclic variation in gas pressure, shaking forces due to the reciprocating parts and structural characteristics of the mounts. Engine vibrations are caused due to the reciprocating and rotating masses of the engine. The variations of inertial forces are due to the combustion and the compression differences of the piston cylinder arrangement during their operation. The engine inertial forces leads to the unbalanced forces of the engine and they are quiet varying with respect to speed, fuel supply and combustion characteristics of the fuel. To predict the vibration output of an engine and to minimize the possible durability and consumer perceived quality problems associated with engine vibration, a robust and accurate design and simulation model is needed. To reduce the engine vibration proper mounting must be provided as dampers at the interface of the engine and chassis.
The vibrations caused at the engine are two types they are torsional and longitudinal vibrations. Engines always have some degree of torsional vibration during operation due to their reciprocating

nature. The rotation of crankshaft of an engine increases the cylinder pressure as the piston approaches top dead centre (TDC) on the compression stroke. Ignition and combustion increases the pressure just after TDC and the pressure starts to decrease when the piston moves down to bottom dead centre (BDC). The pressure on the piston generates the tangential force that does useful work and increases the rotational speed of the crankshaft during this combustion stroke, whereas the compression stroke decreases the engine's angular velocity. The changing rotational speed results in the speed fluctuations of the crankshaft and the torsional vibrations at the crankshaft. The reciprocating and rotating components of engine have subjected to variation in inertial motion and the combustion pressure during the operation and the variation in the inertial motion of the parts during the upward motion and variation in the combustion pressure during the downward motion produce the unbalanced forces at the engine block and the unbalanced forces at the block are measured as longitudinal vibrations in the three orthogonal direction. Both the vibrations can be reduced by minimizing the unbalanced forces and by supporting the engine at proper mounts.

The engine mounts should have characteristics of high stiffness and high damping in the low-frequency range and of low stiffness and low damping in the high-frequency range. Hydraulic mounts do not perfectly satisfy such requirements. Although hydraulic mounts greatly increase damping at low frequencies, they also degrade isolation performance at higher frequencies. Also hydraulic mounts are not cost effective, they had complexity in design and low reliability. Though various types of hydraulic mounts have been developed for the vehicle mount systems, it is still reported that the rubber mounts show significant importance in ride comfort and reduced noise levels. Rubber mounts can be designed for the necessary elastic stiffness rate characteristics in all directions for proper vibration isolation and they are compact, cost-effective, and maintenance free. Also the rubber mounts offer a trade-off between static deflection and vibration isolation. Rubber mounts have been successfully used for vehicle engine mounts for many years.

Therefore a multi-physics approach needed to be used to address all these physical properties in a single design model. To identify the various methods used and assumptions followed to find and reduce the engine vibrations a survey were made on the Engine 1.Rigid body modeling 2. Vibrations and 3. Mountings.

II. ENGINE RIGID BODY MODELING

IC engine consists of different components such as engine block, engine head, piston, connecting rod, crank shaft, flywheel and cam shaft, valves, manifolds, pulleys etc. Some of parts are identified by the researchers and engine manufacturers as the vibration producing parts. The piston connecting rod, crank shaft and engine block are the major components which produces unbalanced forces during the operation. As they are interconnected together the forces are transferred to the engine block and hence to the supporting structures. Many mathematical rigid body models were proposed by the researchers using multibody modeling of the engine structure to calculate the unbalanced forces from the engine.

Sudhir Koul et. al [4], developed a mathematical model based on the structural dynamics that includes frame, power train assembly, swing arm assembly and engine mounting system. The authors developed two models with six degrees of freedom rigid body modeling of the flexible frame. In the first model, it consists of finite element modeled stiffness matrix such that the nodes of the frame connect the other sub-system. The other is the respective dynamic model of the frame and the swing arm. The model was developed such that the mount stiffness, mount locations and mount orientations were the design variables. The model was simulated for two different load models and the results were optimized for the frame stiffness and for the minimum force transmitted to the frame. Hoffman and Dowling [21] conducted an experiment on heavy duty in-line six-cylinder Diesel Engine to measure all the three orthogonal vibration force components at the each of the three engine mounts during standard impact-excitation. Modal identification tests on the quiescent-engine and during engine operation. Deana. M. Winton and Dowling [22] conducted an experimental study to determine the rigid body modal content of engine block vibration on a modern heavy-duty inline six-cylinder Diesel engine. They used three engine mounts fitted with multi-axis force transducers and exploited standard modal analysis to determine rigid body modal characteristics and engine mount forces signatures of the engine vibration modes of engine block. Hoffman and Dowling [23] developed a seven degree-of-freedom model for low frequency engine vibrations that utilizes two way coupling assumption. They compared results of the two way coupling model with the one way coupling model. Also they identified that the new model properly conserves energy and account for gravitational forces.

Hoffman and Dowling [24] used seven degree-of-freedom model that properly conserves energy and predicts the overall features of the engine's vibratory output. They didn't utilize the assumption vibratory state of engine does not influence the loads transmitted from the moving internal components to engine block. They presented a time and frequency domain comparisons of the model and experiment results made on test engine at full load at peak torque and rated speeds. Zheng-Dong Ma and Perkins [25] developed equations of motion for major components of internal combustion engine using recursive formulation. The derivation equation of motion was automated through the computer program by the use of C and FORTRAN sub routines. The entire automated procedure forms the basis for an engine modeling template. Using the template they predicted the engine responses under free and firing conditions were compared with Adam's models. The results obtained by using different bearing models at the crank shaft including linear, non-linear and hydrodynamic models were discussed in detail. Niccolo Baladazini.et.al [29] presented an innovative approach to dynamic design that has the significant advantage of allowing dynamic requirements to be specified from the earliest design stage. They used Genetic algorithm to optimize the dynamic behavior of engine-sub-frame system and its links to chasis. The optimization minimizes complying with the static and dynamic constraints. The GA was applied to a multi body model of Engine-mount system to derive new, improved configurations. Tsuneo Tanaka and Tetsuya [30] Sakai presented a method to effectively reduce a level of idling vibration in heavy-duty trucks. For that they developed a full vehicle vibration model using Finite element method. The flywheel velocity and fluctuation in flywheel speed were the input to this model and the output from the model is engine excitation forces. D. Geoff Rideout, Jeffrey L. Stein and Loucas S. Louca [33], focused on how the application of existing decoupling algorithms can lead to systematic decoupled engine models for specific user defined conditions. They described the decoupling search and model partitioning algorithm, and the bond graph formalism that facilitates the execution. Also they compared the results of partitioning algorithm that applied to a balanced and an unbalanced an in-line six-cylinder engine fully coupled model.

Jae-Yeol Park and Rajendra singh[2], identified the drawback the ignorance on non-proportional viscous damping in the early design of hydraulic mounts. Because of this drawback rigid body vibrations are included as and when the proportional damping is considered in the mounts. To rectify this, they formulated a mathematical model for a non-proportionally damped linear mount and investigated the torque roll axis of passive mounts. X.Zhang and S.D.Yu[12], developed the rigid body and flexible body models to predict the torsional vibrations at various load conditions and different propeller pitch settings of an air craft engine. Here, the rigid body model and Kineto-elasto-dynamic model are coupled together also a stepped crank shaft model is developed with the help of finite element modeling. The aerodynamic torque, developed from the blade element geometry, variations with respect to the speed at the interface of crank shaft and the propeller was obtained. Augmented Lagrange equations were used to obtain non-linear equations of motion then the small scale model was developed by applying the component mode synthesis in the equation of motion without changing the non-linearities. The steady state and unsteady state responses were determined with the help of Runge-Kutta algorithm. From the results the authors identified the significant influence of crank shaft flexibility on the dynamical behavior.

III. ENGINE VIBRATIONS

Engine produces the vibratory forces due to the unbalanced forces from the engine parts during the operation. The vibration caused by the engine at the supports is torsional vibration and the longitudinal vibration. The torsional vibration is caused at the crankshaft due to the fluctuating engine combustion pressures and engine loads. The longitudinal vibrations are caused at the block and the mounts by the reciprocating and rotating parts of the engine.

Snyman et.al [20] concerned with minimization of engine vibration in the mounted 4-cylinder internal combustion engine. They analyzed the mathematical model and the balancing mass and the lead angles were taken as the design variables. The objective function used in their research is the vibratory forces from the engine, transmitted to the engine mounts. The objective function is to be minimized to minimize the vibratory output of the engine and the Leap frog optimization algorithm is employed to minimize the objective function. Conti and Bretl [28] presented a new method for determining an analytical model of rigid body on it mounts and the method is based on data acquired. They used modal experimental data from an artificial excitation of vibration of test of the article suspended to ground through mounts as input to the model and the output is rigid body mass properties and stiffness of the mounts. The extracted modal data for these six modes is input to a least square algorithm, which was used to compute mass and centre of gravity of location, mass moments and principal axes of inertia and tri-axial stiffness of mount. Chung–Ha and Clifford.G.Smith Shu [27], presented simplified method to determine the vibrational amplitude developed by a 4-cylinder engine when supported on viscoelastic mounts. They modeled the engine parts as rigid bodies connected to the rubber mounts which were modeled with spring and damping elements. The location, orientation and stiffness of the mounts can easily be optimized to reduce vibration and noise in the engine design Nader Vahdati and L. Ken Lauderbaugh Saunders [41], described a high frequency test machine was that allowed test engineers to study the high frequency performance of rubber mounts. The mathematical model of the high frequency test machine was presented. Simulation results of the high frequency test machine showed that with the proper design of the test fixture, and appropriate selection of the reaction mass and reaction mass mounts, one can perform a high frequency dynamic stiffness test on rubber mounts at frequencies as high as 5000 Hz. Simulation results indicate that the weight of the test specimen test fixture needs to be kept to a minimum. In the high frequency test machine described that it was not possible but very desirable to directly display the test specimen's dynamic stiffness.

P.Charles et. al [13], investigated the fault detection related to diesel engine combustion based on the crank shaft torsional vibration. They used encoder signal, to measure the speed of the shaft, to develop the instantaneous angular speed (IAS) wave form which is the significance of torsional vibration. They used IAS and fast Fourier transform (FFT) to monitor the 16-cylinder engine.. In this investigation enhanced FFT was used by improving signal processing to determine the IAS signal. They also introduced a novel method to present IAS signal through poal coordinates. Fredrik Ostman and Hanna T. Toivonen[14], were presented a method to reduce the torsional vibration in reciprocating common rail diesel engines. They identified that cylinder wise non-uniform torque was the reason for the increased torsional vibration and stresses at the mechanical parts of the engine. The non-uniform torque in each cylinder can be balanced by adjusting and controlling the cylinder wise fuel injections so that the balancing of torque will be obtained. They proposed an active cylinder scheme to reduce the torsional vibration. The model relates the consecutive cylinder firings and the torque and the output of the model were used to adjust the cylinder wise fuel injections to compromise the non-uniformity of the torque. Zhang Juhong and Han Jun[15], investigated the design modifications for a new engine that would reduce low-fequency-radiated noise and vibration below the existing production of engine and optimizing the noise and vibration characteristics. The authors considered the combustion forces, main bearing reaction forces including damper function and flywheel whirling, piston side forces, cam shaft bearing reaction forces, impact of valve opening and closing, valve train forces from gear/chain and drive train forces and moments as relevant excitation forces for noise and vibration harshness(NVH). Considering all the factors the authors developed a complex simulation model to calculate the NVH. The model used in this study is a combination of Finite Element Analysis (FEA) and Multi Body Analysis (MBA). The FEA models were used to simulate and to analyze the vibrational behaviors of the components and the MBA was used to simulate the whole body movement. Rajendran and Narasimhan [26] studied the effect of combined torsional and bending free vibrations in the single cylinder engine crank shaft. For that the developed a all-finite-element model developed and the results obtained indicate that the inertial coupling introduced influences the free vibration characteristics. From the result they shown that, under such condition, modeling the crank shaft as a pure torsional system would involve considerable error.

H. Ashrafiuon [31]et al focused on frequency response of an aircraft engine to determine harmonic forces. The locations, orientations, and types of mounts used are all critical in minimizing the transmitted forces. They also presented the results for a specific aircraft engine. The methodology of this work was applicable to most of the vibration isolation systems. H.R. Karimi and B. Lohmann[33] introduced a computational solution to the finite-time robust optimal control problem of the vehicle engine–body vibration system based on Haar wavelets. The robust optimal trajectory and finite-time robust optimal control of the vehicle engine–body vibration system were obtained approximately by solving the linear algebraic equations instead of solving the differential equations. Hongkun Li, et. al[34] used Hilbert spectrum analysis that was applied to diesel engine pattern recognition. They showed that after pre-processed by the wavelet packet, the HS was more accurate and convenient for real diesel vibration signal analysis. In this paper, experimental data of a diesel fuel injection system of different conditions was used to evaluate the improved methodology for system pattern recognition and fault diagnosis. Claes Olsson [35] considered a general vibration isolation problem covering arbitrarily, structurally complex machines and receivers, whereas 1-DOF isolation and a single feedback sensor were assumed, ie single-input single-output (SISO) vibration isolation problems. The objectives of this paper were investigating the impacts of structure flexibility on two different open loop transfer functions, the consequences of neglecting flexibility on closed loop performance and stability of an automotive vibration isolation application. Borislav Klarin et. al [36] developed a numerical model based on multi body dynamics (MBD) and finite element method (FEM) and the model was simulated at different speeds of the engine to predict the torsional vibrations at the crank shaft and engine mount vibration and structural borne noise. Both the experimental and numerical natural modes analysis was done and results were compared in the form of modal assurance criterion. The finite element model was validated by modal analysis of whole engine in constrained conditions. Konrad Kowalczyk et. al [37] discussed the active vibration control (AVC) system that generates dynamic forces to cancel the effect of incoming excitations. They presented an overview of model-based development of control algorithms, a short description of system components. They also described AVC system installed in the test vehicle and an explicated presentation of tools for the development of control functions. X. Moreau et. al [42], proposed a method analyse the vibration isolation. Initially, they presented the derivative models that were used to model viscoelastic material properties of suspension system. After that by taking into account the vibration sources and the sprung mass uncertainties they described the single-degree-of-freedom model. Then, the design for suspension was transformed into a design for robust controller, independently of whether it is active, semi-active or passive. The use of fractional derivative models not only permits optimisation of just four parameters, showing the ‘compactness’ of the fractional derivative operator, but also leads to robustness of suspension performance to uncertainty of the sprung mass. At the last they presented a called engine mount system.

Jae-Yeol Park and Rajendra Singh [38], identified ignorance of non-proportional theories or design methods in viscous damping formulations. In the earlier studies, though the torque roll axis decoupling is still theoretically possible with proportional damping assumption rigid-body vibrations are needed to be coupled whenever non-proportional damping is introduced to the mounting system. They rectified the difficulty by re-formulating the problem for a non-proportionally damped linear system and recognized that significant damping will be possible with passive mounts. Also they examined the torque roll axis decoupling paradigm by keeping given mount rate ratios, mount locations and orientation angles as important design parameters. Based on the above a derived two eigen-value problems and necessary axiom for a mode in the torque roll axis direction in terms of stiffness and damping matrices that are on currently satisfied and both steady state and unsteady state responses and the extent of coupling or decoupling is quantified. Results shown that, the torque roll axis for a mounting system with non-proportional damping was decoupled, when one of the damped modes lies in the torque roll axis direction. Z.K. Peng and Z.Q. Lang [3], invented a new concept of output frequency response function (OFRF) through an effective algorithm for the purpose of design on non-linear passive engine mounts for the effective vibration isolation. They developed an analytical relationship between output frequency response and the non-linear parameters by the polynomial type non-linear differential equation. The objective of this analysis is made to analytically reveal the effect of non-linear characteristics of passive engine mount on the frequency response and how it facilitates in the design of engine mounts. The model was simulated to validate the output frequency response of the system. The author strongly suggests that the OFRF results shows significant importance in the analysis, design and selection of non-linear engine mounts. Also this OFRF model can be extended for the multi-degree of freedom.

IV. MOUNTINGS

The unbalanced forces produced from the engine are transferred to the engine supporting members and causes the structure borne vibrations. To reduce the vibratory forces from the engine to the structures, the engine is supported by the damping members called vibration isolators (engine mountings). The mountings are the final most sources to reduce the vibratory forces by its damping property. Mounts are designed to satisfy two important criteria the first is the support function, reduction of the large amplitude vibration, at lower resonance bands. It requires the mountings to have higher stiffness and damping. The other is noise control, the mountings have to reduce the noise in the supporting structures induced by small amplitude vibration of the engine, at higher bands. It requires the mountings to posses lower stiffness and damping. These two requirements are contradictory, and the main aim in the design of engine mounts is to stabilize these two different conflicting requirements.

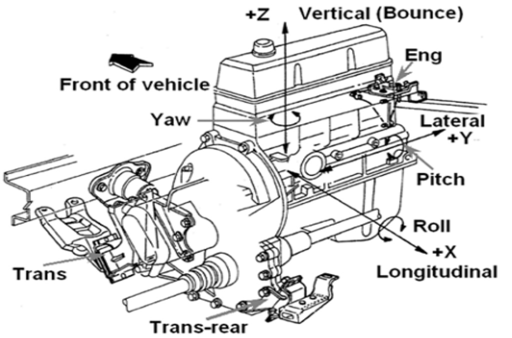

Fig.1 Engine model

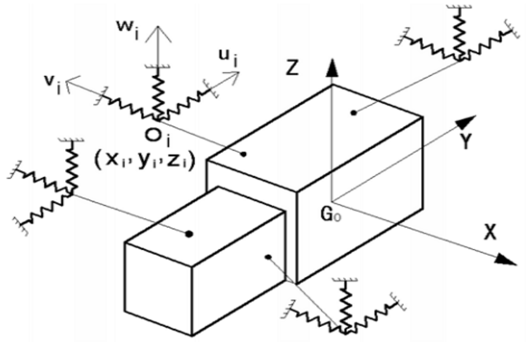

Fig.2 Typical engine mounting system

Yunhe Yu, Saravanan M.Peelamedu, Nagi G.Naganathan, Rao V.Dukkipati[19], made a survey on automotive engine mounts. The survey was on the basis of overview and development of different engine mounts and optimization of the engine mount systems. They made a study about the ideal engine mounting system which would isolate the vibration excitations from the engine and shown the concentration requirement of improvement of frequency and amplitude dependent properties. They explored how the rubber mounts trade-off the static deflection and provides the vibration isolation and what way the hydraulic mounts provides the better performance than the rubber mounts. At the final they reviewed about the different methods of optimization of engine mounting systems and suggested that active mounting systems would be considered for future trend. A.R. Ohadi and G. Magsoodi[1],Developed a mathematical model by assuming the hydraulic mounts as lumped parameter model to investigate the non-linear parameters, such as inertia and decoupler resistances and the effect of non-linearity in hydraulic engine mounts. The model was developed such that the model includes the forces and moments due to engine mounts, pistons, balancing masses. The model was simulated at different speeds of the engine. The results obtained were in terms of base forces transmitted to the mounts for both hydraulic and rubber mounts. The comparison were made and found that the efficiency of the hydraulic mount was more at the low frequency regions. J J Kim and H Y Kim[10], suggested the optimum design using parametric design process with the help of the parameter optimization method. The method was developed using finite element based computer program and the program determines the shape of engine mount that fulfils the requirement of the engine mount stiffness. They used a bush type engine rubber mount model for a passenger car. The developed model was considered as basic model and modification were made for the large deformation and endurance analysis of the rubber mounts. The model was simulated in the bush type rubber mount for optimization of the shape of the mount and subsequently for the desired stiffness values. Qian Li et. al[40], predicted the fatigue life of a rubber mount by combining test of material properties and finite element analysis (FEA). They arrived the fatigue life equation for the natural rubber material based on uniaxial tensile test and fatigue life tests of the natural rubber and the strain distribution contours and the maximum total principal strains at different loads in the x and y directions were obtained using FEA method. The author first obtained and analyzed were the critical region cracks prone to arise. The maximum total principal strain, fatigue parameter, was used to predict the rubber mount fatigue life from the fatigue life equation and the prediction method was validated with help of the results of the fatigue life test rig.

Thanh Quoc and Kyoung Kwan Ahn[5], developed a theoretical model to isolate the vibrations from engine and to control the area of inertia track under shocks and multi-signal excitation in the passive hydraulic engine mounts. The authors introduced a controllable inertia track and made considerable changes in the dynamic properties with respect to the area of inertia track. They identified that the dynamic properties were varied as the inertia track was changed. The results of the numerical simulations showed that the vibration isolation of the passive engine mount was affected by changing the inertia track area and the optimization, developed based on the frequency band and magnitude, of dynamic properties was made. The authors were also identified that many of the researcher were not considered the displacement of the mount in the reduction of dynamic stiffness to reduce the force transmitted to chassis. Bo-Ha Lee and Chong-Won Lee[6], studied and developed a new type of active control engine mount (ACM), feed forward control based electromagnetic engine mount that illustrates both the active and passive characteristics of ACM such that the model includes the vibration isolation estimation algorithm, current shaping controller and an enhanced model for ACM. The authors used two sensors, one is to measure the force transmitted to the ACM and the other to monitor the position and hence acting as position feedback control. The results obtained from the prototype were depicted that the ACM behavior at the dynamic conditions at the desired range of frequency were found to be accuracy. They also showed that the vibration estimation algorithm efficiently explored the anti-vibration signals for the vibration isolation and the proto type ACM effectively isolated the vibration forces from the engine. Lee Jun Hwa and Rajendra Singh[7], states that hydraulic mounts has at least one inertia element even at low frequencies. Thus the constraints at the input and output of the hydraulic mount are not identical and like a conventional spring. So the mechanical model analogous to the hydraulic mounts that has spring, dashpot and mass elements will lead to the poor results in the system analysis. The author critically examined the dynamic response of the hydraulic mounts using mechanical models. They clearly depicted that the models must not be employed where both the ends of hydraulic mount move and the models will work only if one of the end is fixed.

Taehym Shim and Donald Margolis[8], developed controlled equilibrium mounts (CEM) for an aircraft engine which is much smoother than usual mounts to isolate the engine vibration. The CEM uses the additional control effect of equilibrium position of mount by the by-pas air from the engine. The Equilibrium position of mounts can be obtained by pressurizing and exhausting of air in to the mounts by the control valves. They developed a CEM simulation model considering the thermodynamic effects and heat transfer characteristics of air, valve control logic and equation of motion. The model consists of parts like conventional mount, expandable volume, control-valve and supply system. The electrometric part of the mount provides the basic isolation and the expandable volume provides the additional isolation by pressurized/exhausted air, supplied from the engine, with help of control valves. The simulations were made to find mount displacements during taxi, climp, cruise and decent conditions of the air craft. The authors stated that the heat transfer rate between inlet and outlet air was excessive and will be considered in future study. J.Christopherson and G.Nakhgie Jazar[9], investigated the linear and non-linear aspects of two distinct passive hydraulic engine mounts with floating decoupler and with direct decoupler. Both the decoupler mechanisms were used to control the amplitude dependent behavior of the mount. In the first type, the fluid is forced through the inertia track due to relative motion between the engine and the chassis and this design relies upon the appropriate combination of inertia track and decoupler gap size. In the second type, the decoupler directly connected to the engine mount and its motion is directly controlled by the engine motion of the engine and the chassis. The linear and non-linear mathematical models of the both type were simulated and from the results it has been identified that the direct decoupler mount exhibits the lowest transmissibility in low frequency domains whereas the floating decoupler shown better performance as the excitation frequency increases. The non-linear response solutions of the both mounts were validated by direct comparison with the linear counter parts and it has been identified that the similarity between the solution regions sufficiently removed the resonance in the non-linear modeling They also experienced difficulties in the mathematical modeling of the direct decoupler when using the new method, energy-rate method, for the analysis of stability of the parametric system.

J.S.Lee and S.C Kim[11], focused on the performance enhancement of engine mount rubber(EMR) by adopting a form a design optimization approach. The optimal design was arrived by considering the material stiffness and fatigue strength of a rubber. The objective of the optimal design was made to minimize both the weight and the maximum stress of the EMR and to maximize the fatigue life cycle subjected to constraints on static stiffness of rubber. The number of life cycles associated with the fatigue strength should be increased as much as possible under the acceptable material stiffness behavior. Such design requirements made possible through multi objective optimization method. In the context of approximate optimization a back propagation neural networks was used to construct global response surface between input design variables and output responses of objective functions. A micro-genetic algorithm was adopted as global optimizer in order to consider the inherent non-linearity analysis of the model. Jun-Hwa Lee and Kwang-Joon Kim[16], developed viscous damping model for hydraulic engine mount and represented the model in terms of design variables. The design variables are geometry of inertia track, resultant stiffness and damping characteristics. The parametric studies were presented the relation between the equivalent viscous damping coefficient and the design variables. The authors discussed the lumped parameter and dynamic performance characteristics of the mounts. Based on these two combinations the efficient design technique for the hydraulic mount was made. The authors state that relation between the design variables and the dynamic characteristics will be helpful in design modification and initial design of hydraulic mounts so that the desirable performance of the mount will be obtained.

Li-Rong Wang et.al [17], developed a non-linear parameter model to determine the dynamic characteristics of hydraulically damped mount (HDM). The results of dynamic characteristics obtained from the mathematical model with fixed-decoupler were compared with the experimental values of the typical HDM and verifies the effectiveness of the modeling. The HDM's working methodology was clarified by super positioning the performance with the rubber mount, inertia track and. The authors also made a analysis of parametric effect to bring out the influence of structural design parameters on the vibration isolation performance of the HDM. The authors identified the following : modeling technology of rubber viscoelasticity, fluid-structure interaction between rubber parts and fluid in chambers and frequency and amplitude dependent characteristics of volumetric stiffness of upper chamber for future study. Lirng Wang et. al [18], developed a mixed finite element formulation to analyze hydrostatic fluid-structure characteristics of hydraulically damped mount (HDM). They developed a set of equations for the fluid transfer analysis of mass flow rate and pressure difference between the lower and upper chambers. The fluid structure interaction finite element (FSI FE) modeling of lower and upper chamber was done they were bridged and finally the HDM integrates the FSI FE and fluid transfer characteristics models. The author states that such type of integration of hydrostatic FSI approach with lumped parameter modeling of fluid track can reduce the computational cost of FSI in HDM.

Yougn Kong Ahn et.al [14], developed a methodology to improve the non-linear hydraulic engine mount by an optimal design process. The hydraulic mount with inertia track and decoupler has variation in property based on the disturbance frequency range and is small at higher excitation. For that they developed two linear models, one is low high frequency range and the other for high frequency range. The combination of two models also was used in optimization of the hydraulic mount. For the optimal design of the non-linear fluid engine mount with inertia track and decoupler they used sequential quadratic programming (SQP) technique to minimize the transmissibilities of the fluid mount. The frequencies used in the SQP for low frequency range model and high frequency range model were calculated from the simulation of the mathematical model. To obtain effective vibration for the high and low frequency range models the SQP was combined with low and high frequency range models. The design parameters that greatly affect vibration isolation effectiveness were considered as design variables such as the effective piston area, inertia track, inertia of decoupler, inertia track resistance, decoupler resistance, rubber stiffness and compliance in top and bottom chambers. A. Geisberger et. al [39], developed the non-linear model such that the model is able to capture both the low- and high-frequency behavior of a hydraulic engine mount and the model was validated with the help of unique experimental set up.The results of the model here provided a significant improvement over existing models by considering all non-linear aspects of a hydraulic engine mount. They made Enhancements on the existing non-linear models to include a continuous function that follows a simple and effective approach to capture the switching effect and leakage through the decoupler, and upper chamber bulge damping. The developed model also showed appropriate system response over the full range of loading conditions. From the experimental set up initially the individual components of the mount model was verified and then the behavior of the whole assembly was verified. The data obtained from the results gave the relative importance of several damping, inertia and stiffness terms and the measured responses of the mounts at various frequencies and amplitudes are compared with the results of the mathematical model. Daniela Siano et al [43], used Multi-Body and FEM-BEM methodology used to predict the noise radiated by a turbocharged 4-cylinder diesel .A Multi-Body Dynamic Simulation (MBDS) of the engine was carried out by simulating an engine to estimate the forces acting on the cylinder block. The dynamics of the engine is described taking into account the effects of the gas pressure and the inertia forces of the moving parts. In this work to identify the real engine operating behaviour, both the crank and the block have been considered as flexible bodies. The cylinder block excitations were used to evaluate the engine radiated noise with the MATV methodology.

V. CONCLUSIONS

Based on the literatures from various research articles for the engine various multibody modeling of engines, engine vibration testing and measurement techniques used and the various mountings used to reduce the vibration and recent technologies invented and adopted. The following observations are made on the above literature survey:

- Many of the researchers considered the engine as rigid body model not the flexible one.
- Meager number of articles only took the forces from the engine to the mount for the mathematical modeling as two way coupling.
- Less research were made by combining the both the torsional and longitudinal vibrations.
- For the determination of the engine vibration calculations most of them were not considered the engine combustion force variations.
- In the vibration modeling of the vehicles the engine vibration modeling and road vibration modeling were considered separately not as single model.
- The recent developments of the mounts on the hydraulic and electromagnetic mounts were used as the isolator not the conventional rubber mounts.
- A simple and versatile model required to be developed which includes the multibody dynamics of engine and dynamics characteristics of the rubber or hydraulic mounts to analyse the vibratory characteristics of the power plant.

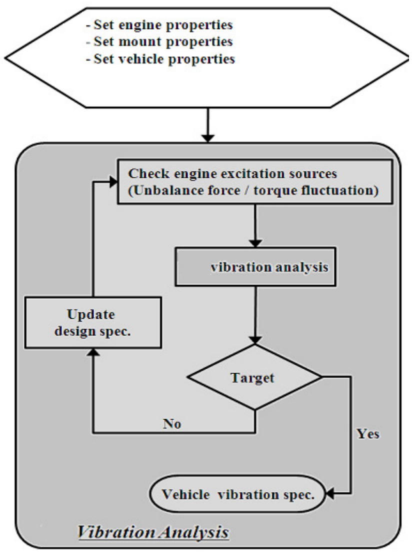

Fig.3 Vibration analysis flow chart

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