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Designing isolation system for Engine/Compressor Assembly of GAS Driven Heat Pump

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ABSTRACT

A gas driven heat pump (GHP) core design comprises internal combustion engine, compressors incorporated to a cooling/heating system, rubber mountings and belt transmissions. Main excitation forces are generated by an engine, compressors themselves and belt fluctuation. It leads to high vibration level of the mount that can cause damage of GHP elements. Therefore an appropriate design of the mounting system is crucial in terms of reliability and vibration reduction. In this paper oscillation of the engine mount is explored both experimentally and analytically. Experimental analysis of natural frequencies and operational frequency response of the GHP engine mounting system enables to create simplified model for numerical and analytical investigations. It is worked out criteria for vibration abatement of the isolated structure. Influence of bracket stiffness between engine and compressors, suspension locations and damper performance is investigated. Ways to reduce excitation forces and improve dynamic performance of the engine-compressor mounting system are considered from these analyses. Implementation of the proposed approach permits to choose appropriate rubber mountings and their location as well as joining elements design. A phase matching technique can be employed to control forces from main exciters. It enables to changing vibration response of the structure by control of natural modes contribution. Proposed changes lead to significant vibration reduction and can be easily utilized in engineering practice.

INTRODUCTION

Severe vibration problems have been encountered during development of GHP outdoor unit equipped by an internal combustion engine. The system is supplied by gaseous fuel. The engine mounting consists of engine that is rigidly connected to 2

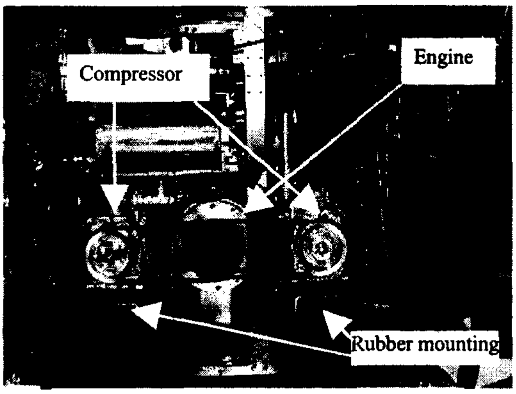

Fig.1 Photograph of the first GHP prototype

compressors, belt transmission, suspension, thermal exchanger and auxiliary elements (Fig.1). The structure generates significant dynamic forces that excite both the engine mounting itself and the rest of the GHP module. It led to destruction of fixation elements that impairs reliability of the outdoor unit. To abate transmission of oscillation from the engine mounting to the structure and decrease vibration of the mounting itself development of bracket and suspension systems has been undertaken.

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IDENTIFICATION of VIBRATION EXCITATION

As it was expected from theoretical considerations main vibration harmonics correspond to

$$f_e = n \cdot R/60 \quad [Hz],$$
$$f_c = b \cdot n \cdot R/60 \quad [Hz],\ n=1,2,3$$

where f_e , f_c are main excitation frequencies of engine and compressor correspondingly, R is a shaft rotation frequency, rpm, b- belt transmission ratio. It is also known that excitation harmonics of a 4-strokes internal combustion engine and 2 pistons reciprocating compressor have maximum at n=2. Experimental measurements confirm this statement. One of the vibration spectrums measured at the compressor body is represented in Fig.2. The GHP is intended to operate at the engine shaft rotation up to 2200 rpm that corresponds to most intensive excitation under 80 Hz. Therefore our investigation is focused

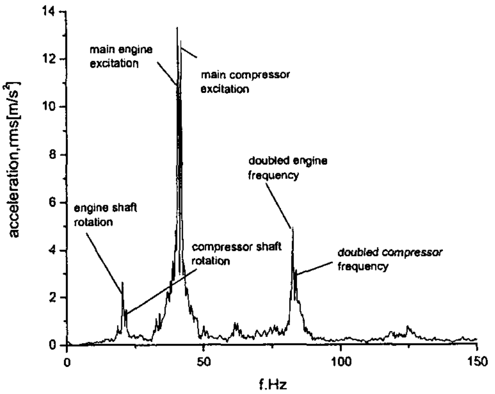

Fig.2 Spectrum of vibration measured at the compressor, regime 1200 rpm

on low frequency band. Modeling of the engine mounting system shows that natural modes are within main excitation frequency band. It was also checked by experimental modal test. The original bracket design has insufficient rigidity of elements and fixation of compressors. It causes significant mutual displacement of engine and compressor shaft even at primarily low excitation that leads to oscillation in belt tension and higher vibration level. This vibration amplification mechanism has been observed at 800-1500 rpm. Contribution of different exciters to the engine mounting oscillation was identified by transfer function method using vibration measurements under operating tests. It is revealed that both engine and compressors have significant impact on the structure vibration. Ratio of their contribution depends on the engine rpm and loading conditions. Oscillations are amplified by dynamical interaction in the engine mounting through the belt transmission and mutual displacement of the engine and compressors shafts at low regimes.

VIBRATION DEVELOPMENT GOALS

To secure robustness of the structure to vibration impact it is necessary to eliminate excitation amplification mechanism that is usually achieved by an appropriate designing of mounting brackets. Stiffness of the elements should be enough to restrict dynamic distortion of engine and compressor shafts. Natural frequencies of an entire engine mounting, joining elements are to be out of main excitation frequency band. It is needed to abate vibration of the engine mounting in one hand and to restrict transition of oscillation to the structure of outdoor unit in other hand. Since intense vibration is detected at low frequencies it is a conventional way to specify vibro-displacement limits:

$$D_{i\,\max} \leq [D_i], \tag{1}$$

where $D_{i\max}$ is a maximal vibro displacement, i is a number of check up point, $[D_i]$ is a permitted vibro- displacement that is chosen to meet regulation requirements or engineering demands, for example to secure reliability of an attached pipe line. Transmitted to the GHP base plate vibration should not exceed a permitted level that is equivalent to requirement by rubber mounting vibration insulation for this case. Abatement of excitation force that is transmitted to the base plate is also very important in terms of reliability of the outdoor unit and low structural noise emission. Excitation force reduction can be expressed by transmissibility function of rubber mountings T_n, n is a number of rubber mounting [1, 2]. In simple case required fraction of force transmission reduction [R] is specified:

$$1 - T_n(\omega) \geq [R], \omega \in [\omega_1, \omega_2], \tag{2}$$

$[\omega_1, \omega_2]$- circular frequency span of interest. If transmitted force or vibro- displacement must not exceed a permitted value, vibration insulation criteria takes form similar to (1):

$$\begin{aligned} D_{n\max} &\le [D], \\ F_{n\max} &\le [F], \end{aligned} \tag{3}$$

where D_{nmax} , F_{nmax} are maximal vibro displacement and transmitted force, [D] is a permitted vibro- displacement, [F] is a permitted force that is transmitted to a base plate.

THEORETICAL APPROACH for DESIGNING of ENGINE MOUNTING SUSPENSION

Reduction of the engine mounting excitation urges to create a rigid bracket system for eliminating undesired dynamic effects. Since the vibration development is required for low frequencies it is possible to utilize equations of a rigid body motion to designing GHP suspension. In this case an engine mounting motion can be represented as a superposition of center of rotation O translation and rotation of mounting around O. Putting into effect conventional assumptions of small oscillation dynamics it is possible to define position of center O by vector $\mathbf{l_0}$ in the coordinate system based in the mass center from torque balance condition for small deviation from ambient GHP state:

$$l_0 = f(k_n, l_n), \tag{4}$$

where k_n is a rubber mounting stiffness, $\mathbf{l_n}$ is a vector defining location of a rubber mounting, i=1,2…- number of mounting. It is also assumed that their reaction is translational and rotational deformations are insignificant. Conventional GHP designs have three or four rubber mountings located at the same vertical level. Thus center of rotation is determined by stiffness of rubbers for different directions and their location in horizontal plane (XOY). Let us consider dynamics of mounting in new coordinate system with center in point O and axes are parallel to principal ones. It is better to employ motion equations for a dissipative system in the Lagrange form [3]:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_r}\right) - \frac{\partial L}{\partial q_r} = F_{q_r}, r = 1,2...6 \tag{5}$$

where L is Lagrangian, q_r is a generalized coordinate, dot upper script denotes corresponding generalized velocity, F_q is a generalized non conservative force. It enables us to gather equation of motion without consideration of reaction forces and acceleration calculations. In the new coordinate system generalized coordinates correspond to translation and rotation around axes. It is also known that gravity force can be disregarded for suspended systems because it is balanced by static deflection of rubber mountings [1]. Dissipation can be considered either by Rayleigh's dissipation function or by adding dissipation forces to non conservative generalized forces that is applied to our calculations. It should be noted that information about mass and inertia properties of the engine/compressor assembly is known from a CAD process or gathered experimentally [4]. Calculation of F_q involves detection of active forces from exciters and dissipation forces that depend on rubber mountings damping factors C_i only (consider viscous damping) for translation movement and on damping factors and location of rubber mountings for rotation. Active forces can be calculated either from modeling or experimental data and referred to the center of rotation. It simplifies solution of equations (4) since generalized forces are equal to projections of resultant torque and force to corresponding axes. For calculation we gather resultant applied force and torque as a function of frequency f, location of exciters that is defined by vector $\mathbf{l_0}$ subtraction from position vector in OXYZ coordinates. It is needed to be defined in the frequency span of interest. Since we consider forced oscillation it is advantageous to find out location of the center O to providing lowest excitation that means minimization of resultant torque from active forces:

$$\sum_{j=1}^{J} r_j \times F_j(\omega) \rightarrow min, \omega \in [\omega_1, \omega_2], \quad (6)$$

where $\mathbf{r}_j$ is a vector defining location of j^{th} exciter, F_j is an excitation force from exciter, j is a number of excitation source that is considered for calculation. Thus we can get new generalized coordinate system where mutual position of center of mass and center of rotation is fixed. It imposes additional geometric- stiffness constraints on rubber mountings which position and properties are to be determined utilizing equations (4). Solution of equations (5) is well known for oscillating systems and can be represented in form:

$$M\ddot{q} + C\dot{q} + kq = F_q, \quad (7)$$

where $\mathbf{M},\mathbf{C},\mathbf{k}$ are matrices of mass (inertia), damping and stiffness, $\ddot{q}, \dot{q}, q$ are matrices of generalized acceleration, velocity and coordinates respectively, $\mathbf{F_q}$ is a generalized force matrix. Inertia components of $\mathbf{M}$ matrix depend on mutual location centers of mass and rotation:

$$I_o = I_M + mf(l_o),$$

where $\mathbf{I}_M$ is an inertia dyadic in reference to mass center M, m- mass of engine mounting, $f(\mathbf{l}_0)$- function performing transformation of inertia dyadic for another coordinate system with parallel axes. $\mathbf{I}_o$ is considered as a fixed parameter for specified center of rotation. Finding out solution of equations (7) for harmonic excitation it is possible to calculate amplitude frequency response function for every generalized coordinate that has a generic form:

$$|Q_r| = \frac{|q_r(\omega)|}{|F_r(\omega)|} = \frac{1/k_r}{\sqrt{(1-(\frac{\omega}{\omega_{0r}})^2)^2 + \left(2\xi_r \frac{\omega}{\omega_{0r}}\right)^2}}, \quad (8)$$

ω_{0r} is a circular natural frequency of conservative system corresponding to generalized coordinate q_r, ξ_r is a viscous damping factor for generalized coordinate q_r. The frequency response for translation movement of a rigid body is affected by properties of rubber mountings only and does not depend on their location. However $Q_r(\omega)$ is influenced by rubber mounting position through stiffness and dissipation coefficients with respect to rotation. Conditions (1) can be represented in form:

$$[\sum_{r=1}^{3}(|Q_r||F_r|)^2]^{0.5} + [\sum_{r=4}^{6}(r_i|Q_r||F_r|)^2]^{0.5} \le [D], \quad (9)$$

where r_i is a distance between check up point and axis of rotation. First term in the left part of expression (9) represents contribution of translation motion to vibro- displacement, the second term has a meaning of rotation contribution to a check-up point vibration. It is known from engineering practice that ξ_r is to be small and natural frequency should be:

$$\omega_{0r} < (0.4 \sim 0.5)\omega_1. \quad (10)$$

Under this circumstance expression (8) is a decreasing function in $[\omega_1, \infty]$ frequency span that can be used to specifying additional requirements with respect to rubber mounting location and properties. Form of restrictions depends on a base plate vibro- insulation specification. To meet conditions (2) stiffness coefficients of rubber mountings k_n must satisfy inequality that can be derived from wide used technique to choosing deflection of rubber mounts [2]:

$$k_n \le m_n(m, N, l_n)\frac{1-R}{2-R}\omega_1^2, \quad (11)$$

where m_n is a mass static load on a rubber mounting, N- number of mountings. It should be noted that the relation above is valid for translation motion and rotation movement is not important under adopted assumptions. If vibration insulation requirements are specified in form (3) it is needed to perform analysis to satisfying conditions that are similar to (8):

$$\Big(\Big[\sum_{r=1}^{3}(k_r|Q_r||F_{rn}|)^2\Big]^{0.5} + \\ +\Big[\sum_{r=4}^{6}(r_n^{-1}k_r|Q_r||F_{rn}|)^2\Big]^{0.5}\Big)T_n \le [F], \qquad (12)$$

where T_n is transmissibility function of a rubber mounting, r_n is a distance between mounting fixation point and axis of rotation. It depends on dissipation properties of mounting and static mass load i.e. on total number of suspensions and their location. Thus expressions (9), (11) or (12) enable to compile an array of rubber mountings physical properties and locations in generalized coordinate system. Massive of probable solutions is subject to geometric and engineering feasibility restrictions. In some cases it is possible to solve optimization task if set of parameters that meet vibration development requirements is extensive. For example conventional task about minimization of force that is transmitted to a base plate in frequency domain [ω_1, ω_2] can be considered as a part of design process.

RESULTS OF THE GHP VIBRATION DEVELOPMENT

The approach described above is applied to development of the GHP engine mounting. Old bracket system has been replaced to create rigid fixation of the parts and eliminate amplification of excitation forces. Modeling results indicated lower stiffness limit for front brackets in vertical direction (Fig.3). If the bracket stiffness is higher than 10 MN/m the vibration amplification effect is not observed.

Structural response of the engine mounting to forced excitation can be controlled by phase matching of exciters. The compressors are equipped by electromagnetic clutches that can be used for phase tuning of compressors and engine. Thus it is possible to change contribution of different natural modes to forced frequency response (Fig.4).It looks like promising technique for vibration control problem. To design effective vibration insulation suspension the mounting was considered as a four degree of freedom body (translations along vertical axis and in horizontal direction normal to the engine shaft axis, rolling and pitching motion). Number of rubber mountings was taken from static load considerations. It equals four. Multi parametric

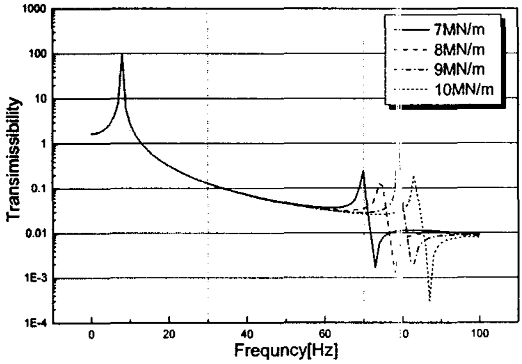

Fig.3 Transmissibility of the engine mounting brackets for different stiffness magnitudes

optimization gave a set of preferable location,

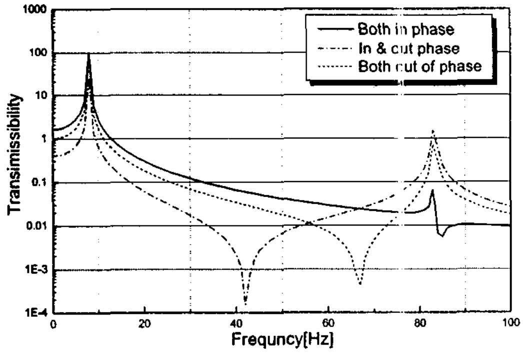

Fig.4 Force transmission depending on relations of compressor excitation with respect to the engine excitation force

stiffness and damping ratio data. Suspension system with rectangle like location of rubber mountings with equal properties (Fig.5) was chosen from available data with additional manufacturing issue is taken into account. These measures brought significant reduction in vibration level of the engine mounting (up to 10 times) and GHP outdoor unit.

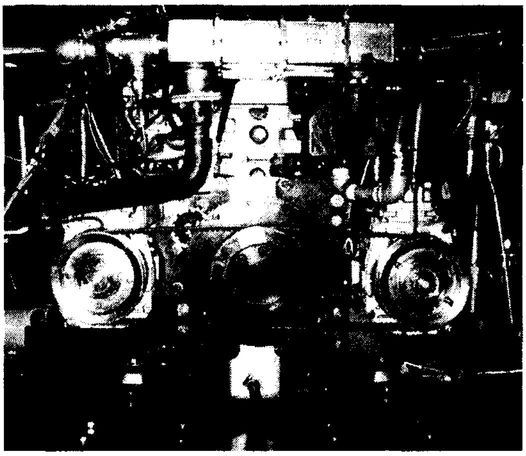

Fig.5 Photograph of the developed GHP engine mounting

Transmissibility functions of the engine mounting for original and developed design are shown in Fig.6 (excitation in vertical direction). The new suspension provides effectiveness of vibro- insulation higher than 95% within required frequency band (30-100 Hz) while the old design has effective vibration reduction at frequencies higher 80 Hz that was a reason for intensive vibration of GHP structure at low operating regimes.

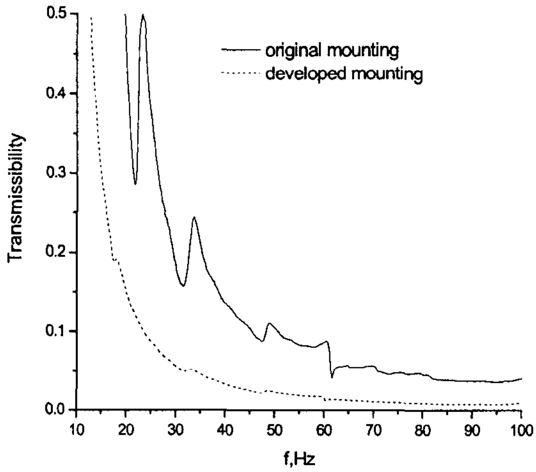

Fig.6 Transmissibility function for original and developed GHP engine mounting

SUMMARY

Modeling and experimental investigations of GHP dynamics enable to working out main requirements for engine mounting design. The bracket system should have natural frequencies higher than frequency band of intense excitation and stiffness of elements that is enough to eliminate undesired dynamic effects. Since vibration development is performed for low frequency band vibro-displacement and transmissibility function are taken as criteria. Under this circumstance optimization of the engine mounting suspension should be based on the proposed theoretical approach that leads to natural separation of stages for choosing parameters of interest. It is calculated position of center of rotation in terms of minimization of resulting generalized force and torque system first. Compilation of inequalities to meeting vibration requirements enables to identify set of probable combinations for rubber mountings properties and their location. Control of phase relations of main exciters (engine and compressors) brings additional opportunities for vibration abatement. Phase match of excitation forces changes contribution of the natural modes to vibration level that can be employed for reduction of oscillations. Implementation of the proposed technique does not require employment of complex design solutions and leads to significant vibration reduction of GHP outdoor unit.

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