OPTIMUM ANALYSIS AND DIAGNOSIS OF THE TRANSMITTE VIBRATIONS IN THE HEAVY STRUCTURES

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ABSTRACT

In this research, the problem of unacceptable high vibration levels of an offshore structures and rock crushers are analysed and diagnosed.

Measurements of vibration, overall levels and frequency spectra are taken at each identified point considering all conditions of the equipments operation.

From the analysis of the measurements, the deck suffers unacceptable high vibration levels varying in severity along the deck. During the normal operation of an offshore structure, both pump backages and their connecting operation are within the acceptable levels of vibration for such machinery. Also, the overflow and test pipelines are suffering turbulant flow condition with cavitation which cause high levels of vibration and there is no correlation between vibrations of the deck and vibrations of the pipes.

The analysis of the results for the two bulldozers under study showed that, engine of Dozer (B) [the problem] is suffering higher vibration levels at no-load. It is obvious that, this higher vibration is due to a faulty gear pump. Covers of both engines are rattling due to resonances from idle engine vibrations. The shocks due to ripping actions vary with the size and locations of rocks.

To absorb high resonance frequencies of valves, specially designed vibration absorber should be welded to the pressure relief in the over flow pipeline of the offshore structure. In order to improve the response of the different sections of the Dozers to the operating conditions, further investigation is recommended by measuring the operating deflection shopes.

1- INTRODUCTION

Vibrations like some diseases, are epidemic. They are vastly transferable from machine to another through their respective foundations. During the process of transfer, vibration may increase in amplitude causing the reciever to be greatly endangered, especially when it happen to be a

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vibration sensitive machine /l and 2/. The vibration measurements must be taken on all units of the system /2 and 3/. From /4/ and /5/, the piping system is excited by internal fluid pulsations which propagate through the piping system and create unbalanced oscillatory at locations where either flow area or direction changes /6/. The vibrations that occure in fluid-filled piping systems are of interest in a variety of industrial, offshore and shipboard applications.

The dynamic behaviour of a piping system includes both flow hammer (a transient phenomenon) and the steady state vibrations caused by unbalanced rotating machinery pumps. Many investigators have proposed various techniques of modeling for piping systems design and analysis purposes /7/. From /8/ and /9/, the pipe line vibration may be caused by internal flow in pipes or by plant machinery such as; engines, pumps, motors ... etc, and it may be also free or forced vibration. Offshore pipe lines are often subjected to wind gust and a knowledge of response of the pipe lines is necessary for their design, analysis and diagnosis the trouble of the system. The predication of response of the offshore units under study is very important to release the reasons of the unacceptable high vibration levels of these units.

The vibration levels of the equipments which are working in rock excavation for a rock crusher is affected by many parameters. These parameters are the loading - unloading operations, vibration induced in the parts from out side /10/ and also sudden application of external load to the bucket /11/. Another important parameter of the dynamic behaviour of the equipment is the shock waves due to the sudden acceleration specially in high speed of the equipment /12/.

The aim of this work is to analyse and diagnosis the un-sufferable vibrations on the offshore units. The vibration measurements are taken at thirty points on the system. The results are analysed to give optimum diagnosis and to give practical recommendations that may help the designers of the offshore units.

The problem which has been investigated also in this work is about high vibration levels during use of bulldozers. This vibration leads to cracks at some places of the equipment.

The analysis and diagnosis of the reasons of this case is the main object of this work. Measurements of the vibration on the bulldozer (B) which is the subject of complaint are analysed, and also to compare these measurements to those taken on the same time and same site on an older similar bulldozer (A) of the same type.

2- EXPERIMENTAL WORK

2.1 - *Offshore Main Specifications*

The configuration of the offshore used in these investigations is consisted of the following units; deck, fire water pumps and pipelines as shown in Fig. (1). The specifications of the units are as follow; Engine; model 3412 (Caterpiller), rated at 800 HP, 1750 r.p.m., and low ideal speed 850 r.p.m., right angle gearbox (Amarillo) S.1.000 G, rating over 1500 HP, agma service factor 2.4. The vertical fire pump 18 HP 600 - 4 (Worthington), four stage turbine pump, total discharge 4276 US gallons/min. and the speed is 1775 r.p.m., steel pipelines of 2 diam; controlled through hand driven valves and consist of; main pipeline for fire fighting, test lines and overflow lines controlled via an automatically operated pressure relief valve on each line.

2.2- *Specifications Of Bulldozers*

The configurations of the two bulldozers used in these investigation are shown in Fig. (2) and specifications of them are listed as follow; The engines of the two Dozers consist of 56D155 - 4,4-cycle, water cooled, turbo charged, diesel engine with 6 cylinder of 19.26 Ltr piston displacement.

Fly wheel horsepower .... 239 KW at 2000 RPM Max.,

Torque .... 144 Kg. m at 1400 PRM,

And direct injection with gear pump - driven force lubrication. Torq-flow transmission cosisting of a water - cooled, 3 - element, single - stage, single-phase torque converter, and a planetary-gear, multiple-disc clutch transmission, that offers single level control of speed (3 forward and 3 reverse) and direction changes.

The final drive of the equipment consists of; Spur - gear, double-reduction final drive, segmented sprocket rims are bolt-on type.

2.2.1 - Under Carriage

- Suspension .... oscillation-type equalizer-bar,
- Track roller fram .... Box section, steel construction,
- Number of track rollers (each side) 7,
- Number of carreier rollers (each side) 2,
- Track shoes .... Reinforced single-grouser shoes,

- Number of shoes (each side) 41,

- Grouser hight 80mm,

- Shoe width ....560 mm,

- Ground contact area ....35280 cm^2,

- Ground pressure (Tractor) ....0.79 kg/cm^2.

2.2.2 - Hydraulic System

All spool-type control valves in hydraulic tank, gear-type hydraulic pump with a capacity of 355 ltr/min and relief valve setting 130 kg/cm^2.

2.2.3 - Rippers

Single shank-variable gian-tripper was used during measurement.

2.3 - INSTRUMENTS

The instruments used in this work are; Vibroport 30 (universal vibration measuring instrument with microprocessor) and Vibro-meter model 2110 (see /3/ for more-specifications of these instruments).

2.4- Measuring Points On Offshore Structure

To survey deck vibrations, 16 measuring points are identified on the deck as shown in Fig. (3), no measurements are taken on the main line. Vibration measuring points on the test line are shown in Fig. (4) while the vibration measuring points on the overflow lines are shown in Fig. (5). Measurements are taken when each pump is working separately at condition of pump operation; (speed 1220 r.m.p. and pressure 8 bar).

2.5 - Measuring Procedure On Offshore Structure

The procedure of the vibration measurements on the offshore units is as follow:

1- A fixed reflecting tape is attached at the rotating fly wheel at each engine and a photo detector is clamped against to measure the speed of rotation during the whole measuring procedure.

2- Engine (B) (south) is first started and reached 1320 r.p.m.

3- Control valves are adjusted to indicate a pressure of 8 bar in the discharge line.

4- After reaching steady state operation condition, vibrafion overall levels along the deck line infront of the two pump packages are measured at 14 points, at each point the vibration velocity (RMS) is measured in three perpendicular directions using an accelerometer and a velocity probe (in some cases), vibration sensors and vibration analyzers are used.

5- Points behind the pump package are surveyed for maximum vibration level and the point giving this maximum level as (11B).

6- Different measuring points on the engine and pump are identified as shown in Fig. (6) and vibration spectra are taken for each point at running speed.

7- Vibration levels on the test pipeline are taken at point (T) Fig. (7) with pump (B) working at 1733 r.p.m. and pressure of 9.5 bar, the results of the overall levels and the vibration spectrum at the direction giving maximum, overall level are shown in Fig.(12).

8- Pump (B) is stopped and pump (A) started to run at speed of 1320 r.p.m. and pressure of 8 bar. At this condition, the deck vibrations on the 14 points indicated in Fig. (8) are recorded in the three measuring directions. The results are shown on the same figure.

2.6 - Procedure of Measurements on Rock Crusher.

To fully investigate the dynamic status of the bulldozers during action, ten measuring points are identified on the engine, body and ripper of each of bulldozers (A) and (B). These points are identified by both number and direction at each point as shown in Fig.(2). The computer is used for collecting the measurements at each point through the accelerometer. The data thus collected, are downloaded into an IBM personal computer containing a Master Trend software for results organization. Measurements of vibration, Overall levels and frequency spectra are taken at each identified point considering the following variables :

1- Equipment identification : (A) and (B).

2- Loading condition: Unloaded, fully-loaded.

3- Full speed and power for each loaded condition.

4- Change of drivers on bulldozers (A) : Driver (1) [of bulldozer (a)] and Driver (2) of bulldozer (B)].

5- Speeding up on a level track for both (A) and (B).

6- Range of frequency : 1 KHz and 20 KHz.

3- RESULTS AND DISCUSSION

3.1- Offshore Structure

From the vibration measurements shown in Fig. (6 and 7) it is clear that, the deck suffers unacceptable high vibration levels/11/ varying in severity along the deck, during the normal operation(*1) either of the pump packages north (A) or south (B). These vibrations are of the flow induced type causing resonances to the deck structure at (22 Hz, 44 Hz and 173 Hz). Background vibrations resulting from other machines on the deck appear to influence deck vibration levels at 101 Hz.

Deck structure respones to van passing frequencies of the pump specially when it runs at speed of 1320 r.p.m., vane passing frequency at this speed excites structure resonance of deck. At normal operating speed around 1750 r.p.m, high vibration levels are also encountered at a frequency of (173 Hz) which corresponds to vane passing frequency at this speed, but the vibration amplitudes are relatively lower. Background vibrations transmitted to the measured area of the deck show high levels at frequency of the (101 Hz) which is traced to be produced by other machines on the deck.

To investigate the effect of speed and pressure, the engine speed and the flow pressure are raised to reach the normal operating conditions; speed 1730 r.p.m. and pressure 10 bar. Table (1) shows the effect of both, speed and pressure variations on the deck vibrations at point (11B).

Vibration spectra are recorded for points on the pump and gearbox in three perpendicular directions, also for the point on the deck that gave the maximum vibration overall, (point 11B), vibration spectra for points on the pump (A) and its gear-box are taken at the same condition indicated above the results are shown in Table No.(2).

To investigate the vibration levels at the overflow pipeline connected to pump (B), the engine speed is reduced again to 1320 r.p.m; measurements are taken on the overflow pipeline at different points shown in Fig. (5), and at different settings of the pressure reliefe valve, with the orifice present in the down stream of the pipe. The results for these measurements are shown in Table (3).

The overflow line of pump (A) is tested under different pressures with the pump running at speeds of 1730 and pressure of 10 bar and also at speed of 1320 r.p.m. and pressure of 5.5 bar, all with the orifice present. The results for these measurements are shown in Table (4).

From these results, it is clear that, both the overflow pipelines and test pipelines are suffering turbulent flow condition with cavitation at the pressure relief valve when the pump operates at the normal working condition, which causes high levels of vibration /11/ at the

(*1) Normal Operation means: Speed of 1750 r.p.m, Discharge pressure of 10 bar, Orifice Present in Overflow Line.

resonant frequencies of the pipes. It can be noticed that, the pipelines (test & overflow) are excited by turbulent flow and cavitation occurring due to pipe profile and restrictions caused by the presence of orifice downstream and some control valve setting. Standing waves produced in the line due to error in some pipe lengths also contribute to pipe resonances.

The orifice on the overflow discharge line (B) is removed and vibration measurements are taken again on the pipeline and on point (11B) on the deck for different pump operating speeds and flow pressures see Table (3) for pipes, and Table (1) for deck vibrations. From the previous results, it can be seen that, the presence of the orifice downstream of the overflow pipeline increases the vibration level of the pipeline tremendously.

Fig. (8) shows the vibration spectrum at point (11 B) on deck for pump (B) runing at 1320 r.p.m without orifice while Fig. (9) shows the vibration spectra at point (IV) on the overflow pipeline without orifice at 1320 r.p.m. and pressure 5 bar. From these figures, both pump backages (Engine, Gearbox, and pump) and their connecting structures are within the acceptable levels of vibration for such machinery /11/.

From the previous results, an intermediate operating condition for the pump which results in very low levels of vibration on the pipelines is as follows : Speed : 1320 r.p.m; Pressure : 8 bar and no orifice, but this condition produces higher vibration levels on the deck. From the previous discussions, it is clear that; there is no correlation between vibrations of the deck and vibration of the pipes, each has it's own sources and response characteristics.

3.2- Dozer Of Rock Crusher

Fig's (10, 11 and 12) show the effect of loading condition on engine (A) and engine (B). From these figures it is clear that, the deviation of vibration velocity between the two Dozers is very large at point (IV). At Dozer (A), the difference of vibration velocity is about three times that between the cases of load and no-load. At Dozer (B), the vibration velocity is about two times that between the two cases. This means that, the engine of Dozer (B) [the problem] is suffering higher vibration levels at no-load, the source of vibration is tracked to be a faulty gear either in the transmission system or a hydraulic gear pump. These should be checked and fault corrected. From the comparison between the two Dozers in case of unloaded as shown in Figs. (13, 14 and 15). The vibration velocity is measured vertically, horizontally and axially on the two Dozers (A and B). From these figures, it can be seen that, points (1v) and (3v) give higher values of (Vv) for both Dozers as compared with the other points. The axial measurements give also higher values as compared with the horizontal measurements. This results lead to the following; both of Dozers are suffering higher vibration in vertical and axial directions in the case of unloaded engines. It is abvious that, the engine of Dozer (B) is suffering higher vibration levels at unloaded case as compared with the engine of Dozer (A) specially in vertical direction. The source of vibration may be a faulty in the transmission system or hydraulic gear pump.

In the case of full load for the two Dozers, the vibration velocity is measured at three points as shown in the previous figures. From these figures, it is obvious that, point (9v) gives higher values for the both as compared with the other points. The results of Dozer (A) gives higher values for the both as compared with the other points. The results of Dozer (A) are very large as compared with the results of Dozer (B). From the previous figures, it is clear that, Dozers (A) and (B) are suffering by vertical and axial vibration resonances. This phenomena of higher vibrations may be due to the deflection in ripper due to full load.

Fig. (15) shows the comparison between the two Dozers (A and B) at maximum speed. From these figure, the deviation in acceleration between the two Dozers is very small. This deviation may be due to release ripper of the two Dozers.

Fig. (16) presents the effect of loading condition on the covers of Engines (A and B). From this figure, it is clear that, the vibration velocity of point (3v) on the covers is nearly the same in the case of loaded, the deviation of (Vv) is nearly 3% only. But in the case of no-load, the (Vv) values of cover (A) are larger than the values of cover (B). This deviation is nearly 5%. From these results, it is obvious that, cover of both engines are rattling due to resonances from idle engine vibrations with the following frequencies:

- Vertical vibration resonances at about 160 Hz.
- Horizontal vibration resonances at 34 Hz, 100 Hz, 140 Hz, these covers need to be stiffened both vertically and horizontally to alter their resonances.

Figs. (17, 18, 19 and 20) present the comparison between two drivers with Dozer (A) and Dozer (B). The measurements were taken at point ($9v_2$). From these result, it is clear that, the acceleration value when using driver (2) is very large as compared with driver (1), it is nearly about twice of the acceleration of driver one. It is obvious that, drive (2) is suffered high vibration. From the previous figures, the driver of Dozer (B) has a large effect on the resultant values of acceleration. From the previous figures, it is clear that, driver (2) [driver of Dozer (B)] is more violent than his colleague and possesses bad driving habits.

Figs. (21 and 22) presents comparison between the rippers of Dozers (A and B) at low frequency, and at high frequency. The vibration velocity is measured vertically, horizontally and axially at four points on the rippers. From these figures, it can be noticed that, bulldozer (A) is affected by higher (Vv) than bulldozer (B) at vertical direction. From Fig. (21) at high frequency, Dozer (A) is also affected by higher vibration in vertical and axial direction. From the previous results, it is obvious that, shock due to ripping actions vary with the size and locations of rocks. These shocks affect the body and ripper mechanism of each bulldozer, differently according to their magnitude. During measurements, bulldozer (A) received higher shocks than (B). Repeated high shock values may cause cracks in bodies and ripper mechanisms. Stiffening made by welding thick plates at position (10 H) reduced the response to high shocks and proved to be a good solution.

4- CONCLUSIONS

From the previous analysis it can be concluded that :

1- Deck area around the pump packages should be stiffened in both vertical and horizontal directions to tune the vibrations of the deck away from its resonant frequencies namely (22 Hz, 44 Hz, 102 Hz, and 173 Hz).

2- Piping spans should be detuned by the addition of flexible clamps and braces at or near all valves and bends.

3- Pipe lengths should be accurately checked, pipe lengths between bends should not be equal to 4.3 m or its multiples. The pipe lengths from the last bend to the discharge opening of the pipe end should not be equal to 2.15 m or multiples.

4- Alternatively, specially designed vibration absorbers should be welded to the pressure relief valves in the over flow pipeline to absorb high resonance frequencies of the valves [(13 Hz for valve (A) & 16 Hz for valve (B)].

5- Engine of Bulldozer (B) [The problem] is suffering higher vibration levels at no-load. The source of vibration is tracked to be a faulty gear either in the transmission system or a hydraulic gear-pump. These should be checked and fault corrected.

6- Cover of both engines are rattling due to resonances from idle engine vibrations. These covers need to be stiffened both vertically and horizontally to alter their resonances.

7- Shocks due to ripping actions vary with the size and locations of rocks. These shocks affect the body and ripper mechanism of each bulldozer, differently according to their magnitude.

8- Bulldozer (A) received higher shocks than (B). Repeated high shock values may cause cracks in bodies and ripper mechanisms. Stiffening made by welding thick plates at some positions, reduced the response to high shocks and proved to be a good solution.

9- Driver (2) [the driver of bulldozer B)] is more violent than his colleague and possesses bad driving habits.

10- During speeding up on a level track, both bulldozers exhibit moderate vibration levels.

11- In order to improve the response of different sections of the bulldozers to the operating conditions, further investigation is recommended by measuring the operating deflection shapes (Modal Analysis).

5- REFERENCES

1- H.I. Gaffer and A.M. Easa, " An Experimental study of the Isolation Technique In Air Compressor: 6th Hungarin Seminar and Exhibitation on Noise Control, Budapest, Hungary, 1989.

2- A. Frid, "Fluid Vibration in Piping Systems - A Structural Mechanics Approach,"II: Applications J. of Sound and Vibration, 133 (3), pp 423 - 448, 1989.

3- W. Cristoph Muller, " Simplified Analysis of Linear fluid - Structure Interactions" Inter. J. for Numerical Method in Engg. (17), pp 113 - 121, 1981.

4- Hong - Sun Liu, "Dynamic response of pipes transporting fluids". J. of Engineering for Industry", pp 591 - 595, 1974.

5- D.J. Wood, "A study of the Response of Coupled Liquid Flow - Structural Systems Subjected to Periodic disturbances", J. of Basic Engineering, pp 532 - 540, 1958.

6- Mohamed, A. H., "Vibration in pipe line system", Practical Experiences with Flow - Induced Vibrations" Lahr-Iutam Symposium Karlsruhe, pp 156-159, 1979.

7- M.A. Satter and G. Ahmadi "Mean-square response of a pipe carrying flowing fluid to non-stationary random excitations". J. of sound and vibration, 51(4),pp 577-581,1977.

8- A. Frid, "Fluid Vibration in Piping System-a Structural Mechanics Approach, I: Theory", J. of Sound and Vibration, 133 (3), pp 423-438, 1989.

9- Bruel & Kjaer, "Applicat Nots (Dynamic Design verification of a prototype rapid transit train using modal analysis", B. 0111-11, 1986.

10- A. A. Shabana, "Dynamics of Multibody Systems", John Wiley and Sons, New York, 1989.

11- A. El-Katib, "Technical Rep. on the vibration of Komatsu D 155 A Bulldozer," presented to EIM, Alex., Branch, 1991.

12- D.J. Ewins, "Modal Test, Engineering Dynamics Series of Technical Engineering Research Studies" 1986.

13- L. Gray & C. Chang," The analysis of planar linkages using A Modular Approach," Mech & Mach. Theory, Vol. 19,pp 165-172, 1984.

14- Black & H.F. Loch "Effects of Hydraulic forces in annular pressure seals on the vibrations of centrifigual pump rotors" J. of Mech. Engg; Vol. 11, No. 2, 1989.

15- Igusa, T.D. Min, K.W. "Resonance of Connected Sub-system; Theory and simple Configurations. "J. of sound & Vibration, 146, No. 3, pp 407-421, 1991.

16- Berman, A.B., "Determining Structural Parameters from Dynamic Testing" SVD, 7 (1), pp 10-17, 1975.

17- A. M. Easa, "An Experimental study into Vibration and Noise in Cutting process of Milling Machine, "Engg. Ref. Bullt., Vol. 13, Part (1). Faculty of Engg., Menoufia University, 1991.

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SPEED RPM / PRESSURE BAR	1320	1400	1730
5.5	23		
8.0	21 21	18.2*	13 11.16
10.0			16

Table (1) : Effect of speed & pressure on deck Vibration.
[Measured at point 11B vertical (*with Orifice).

SPEED / POSITION	Amp. in mm / sec r.m.s				
1730	3B	G2H	P1A	P2A	P3A
R.PM.	21.68	4.34	6.16	4.41	2.85

Table (2) Overall vib. Level in (mm/s-rms)
on the pump package

SPEED	1320. RPM						1730.RPM					
ORIFICE	YES			NO			YES			NO		
PRESSURE / POSITION	5.5 Bar	8 Bar	10 Bar	5.5 Bar	8 Bar	10 Bar	5.5 Bar	8 Bar	10 Bar	5.5 Bar	8 Bar	10 Bar
3H	14.9			5.67	2.6						15.9	11.5
3V				6.5	2.2						11.8	9.7
6V	19.1			6.9	3.2							8.0
7H	21.1			7.6	2.7						20	18.4

Table (3):Overall Amplitudes in (mm/s-rms) [pipe B]

SPEED	1320. RPM						173/0.RPM					
ORIFICE	YES			NO			YES			NO		
PRESSURE / POSITION	5.5 Bar	8 Bar	10 Bar	5.5 Bar	8 Bar	10 Bar	5.5 Bar	8 Bar	10 Bar	5.5 Bar		
3H	16					17						
3V	20					23						
6V	22					22						
7H	19					22.5						

Table (4): Overall Amplitudes in (mm/s-rms) [pipe A]

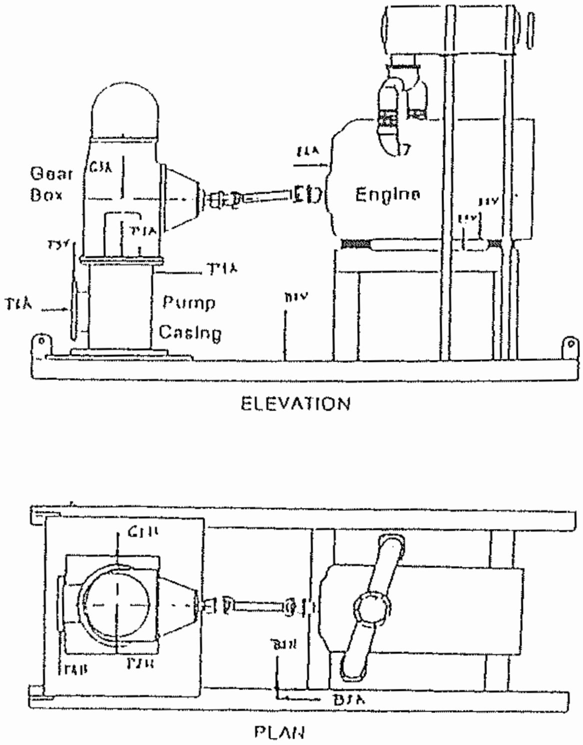
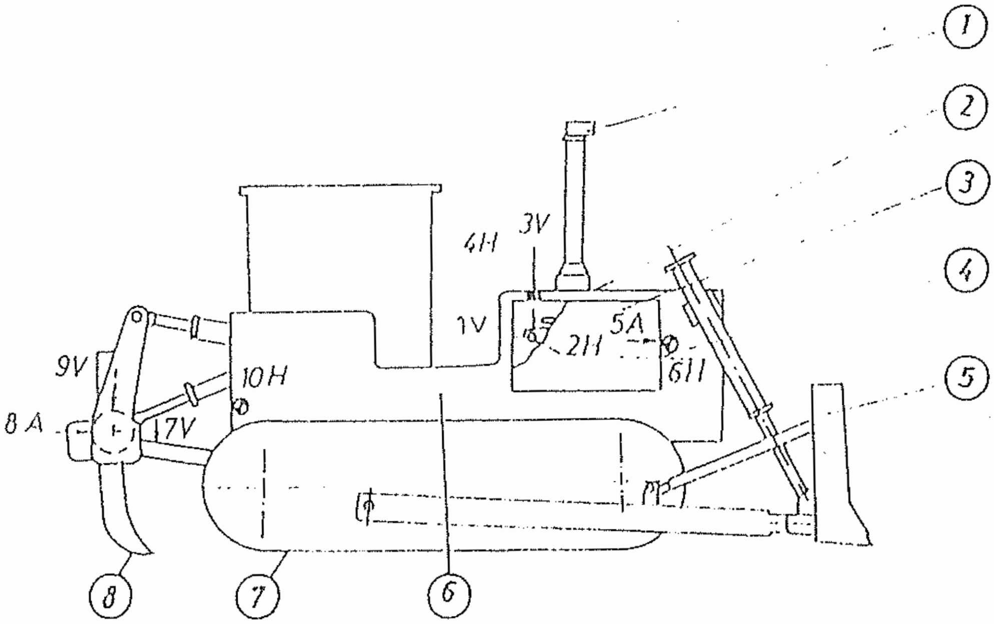


Fig (1) Positions of Vib. Measuring Points of Package.



1_Chimney 2_Engine cover
3_Engine 4_Fan center_line
5_Blade 6_Body
7_Track 8_Ripper

Fig.(2) Measuring points identification of the D 155 A komatsu bulldozer.

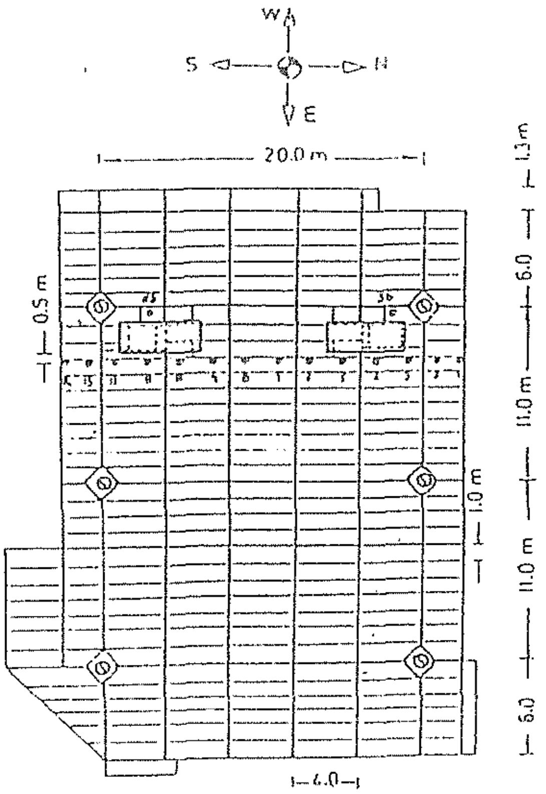


Fig (3) Locations of Measuring Points on Deck.

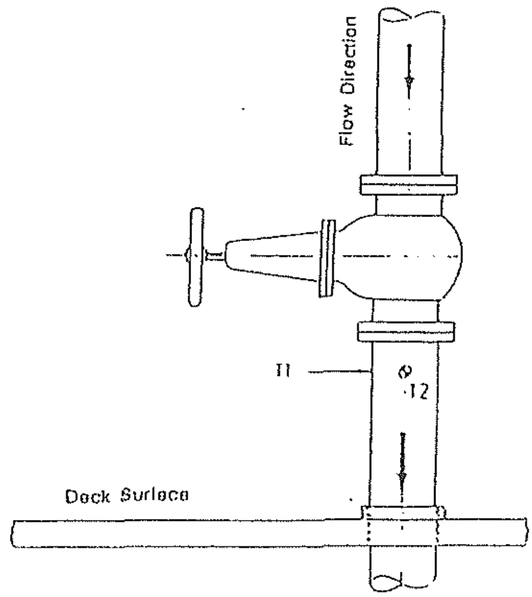


Fig (4) Locations of Measuring Points on Test Line

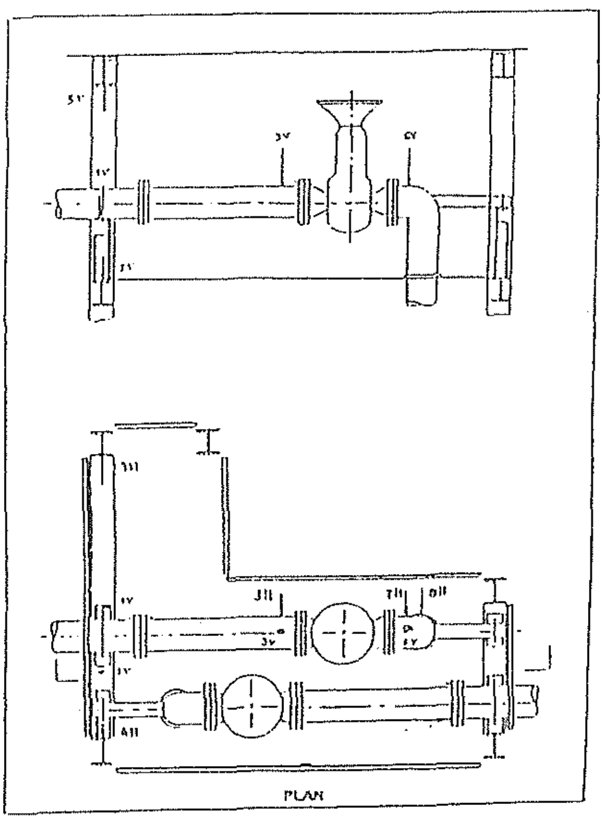


Fig (5) Positions of Vib. Measuring Points on Overflow Pipe Lines.

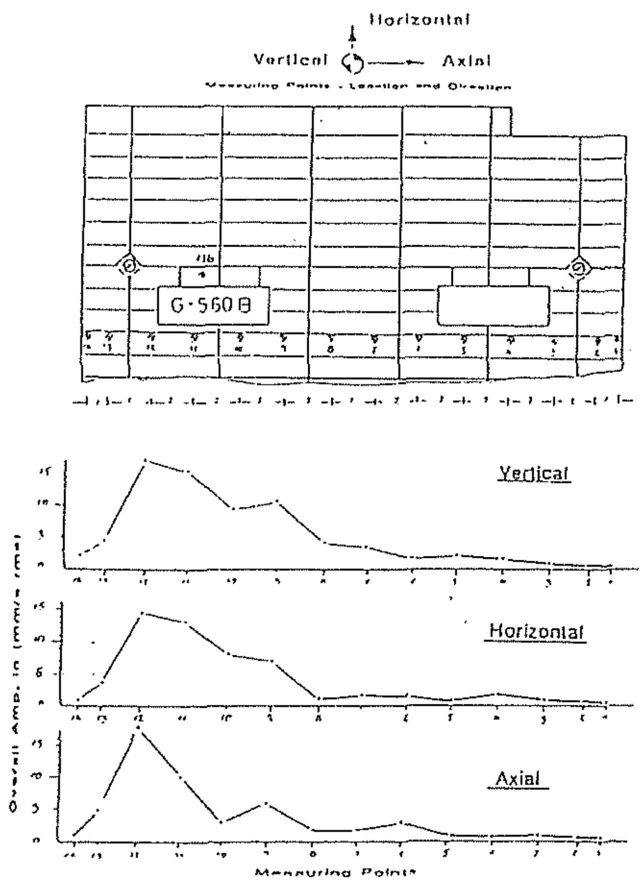


Fig. (6) Vib. Level Distribution Along the Deck.
Pump (B) at 1320 rpm. & 8 bar

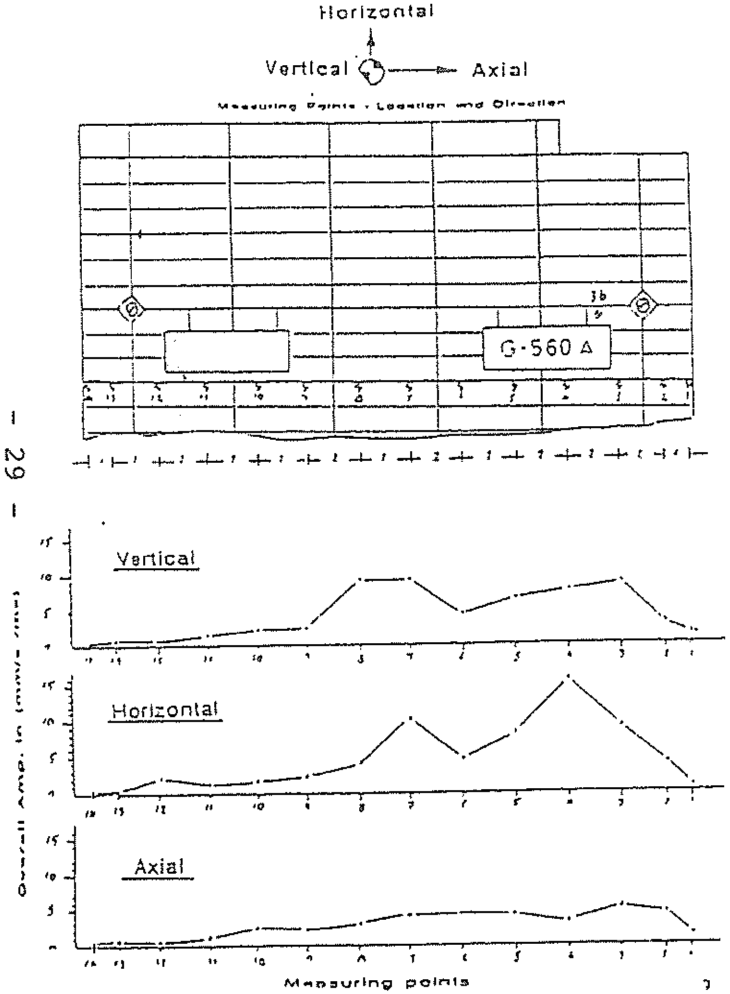
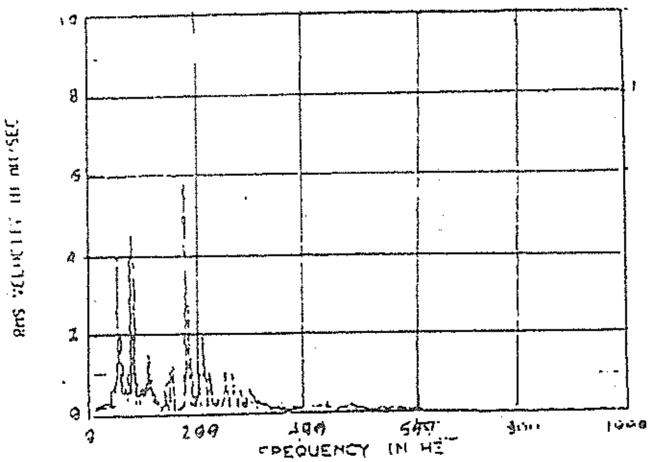


Fig. (7) Vib. Level Distribution Along the Deck.
Pump (A) at 1320 rpm. & 8 bar



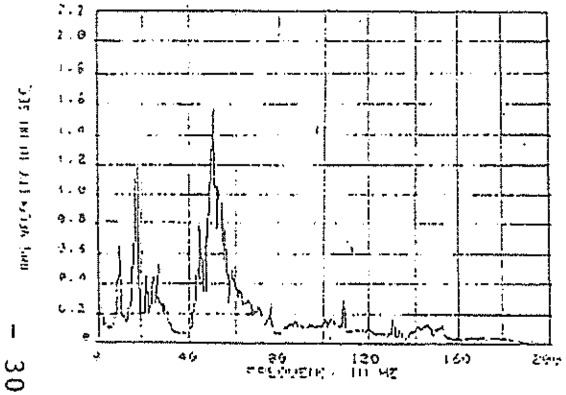
List of Spectral Peaks

Subject: (PFRM) VERTICAL FIRE PUMP G-560 B
Meas Point: PUMP-B -11B --> PUMP-B/DECK 1320RPM -10/APR.
Date/Time: 09-APR 10:47:48 Amplitude Units: MM/SEC RMS
Data Label: NO-ORIFICE · 10 BAR/ 1320 RPM

PEAK NO	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	43.34	4.7754	1.50
2	72.23	5.8999	2.50
3	100.93	1.9211	3.50
4	144.40	1.8681	5.01
5	173.27	7.0123	6.01
6	202.18	3.0617	7.01
7	216.54	1.1169	7.51
8	245.75	1.2519	8.51
9	260.04	1.3675	9.02
10	289.06	.9312	10.03

TOTAL MAG	SUBSYNCHRONOUS		SYNCHRONOUS		NONSYNCHRONOUS	
11.5576	.4278	0%	9.1581	49%	9.3154	51%

Fig. (8)



List of Spectral Peaks

Subject: (PFRM) PIPE/B/NOORIF/1320rpm/5.5bar
Meas. Point: B/NO ORIFC-1V --> STRUCTURE BEFORE VALVE
Date/Time: 09-APR 09:28:38 Amplitude Units: MM/SEC RMS
Data Label: 5.5 BAR/1320 RPM WITHOUT ORIFICE

PEAK NO.	FREQUENCY (Hz)	PEAK VALUE	ORDER VALUE
1	10.12	.6723	.46
2	17.77	1.2915	.81
3	26.90	.5432	1.22
4	45.22	.8968	2.06
5	48.66	.9300	2.21
6	50.24	1.3161	2.28
7	51.27	1.8221	2.33
8	53.73	1.2193	2.44
9	56.22	.8668	2.56
10	59.21	.5404	2.69

TOTAL MAG	SUBSYNCHRONOUS	SYNCHRONOUS	NONSYNCHRONOUS
4.8077	2.2052 / 21%	.9861 / 4%	4.1567 / 75%

Fig. (9)

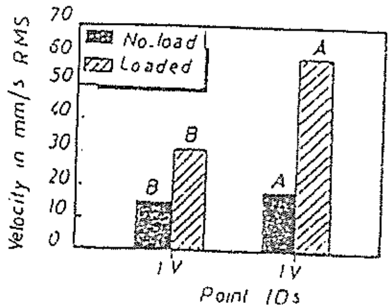


Fig.(10) Effect of loading condition on engine A & engine B. [Comparison of point 1V on dozer (A) & dozer (B) at load/no-load].

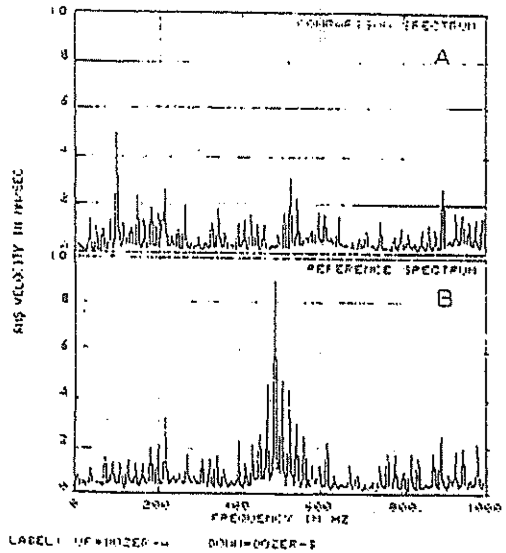


Fig.(11): Comparison of Engine A and B at No Load (Vertical)

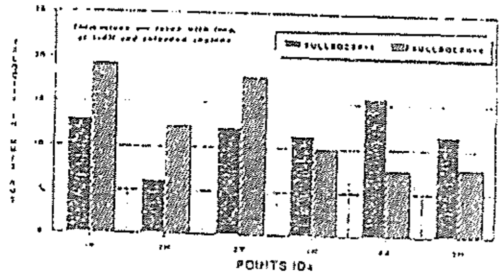


Fig.(12) Comparison between engines of bulldozers A & B.

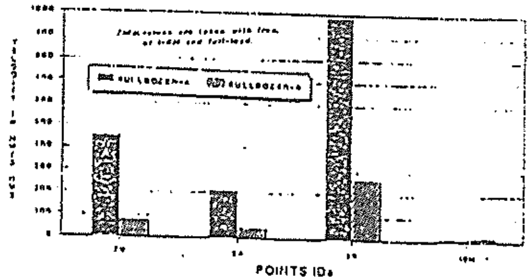


Fig.(13) Comparison between bulldozers at full load.

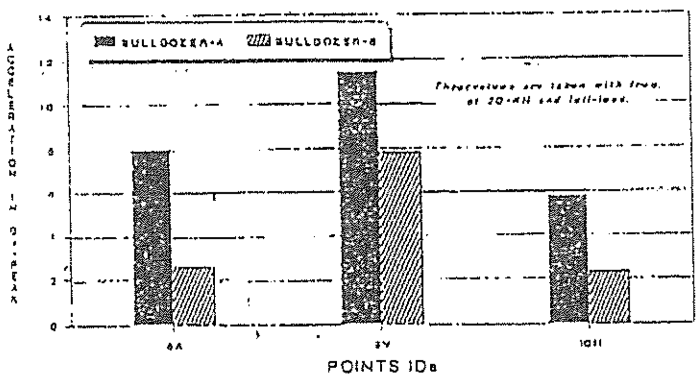


Fig.(14) Comparison between bulldozers at full load.

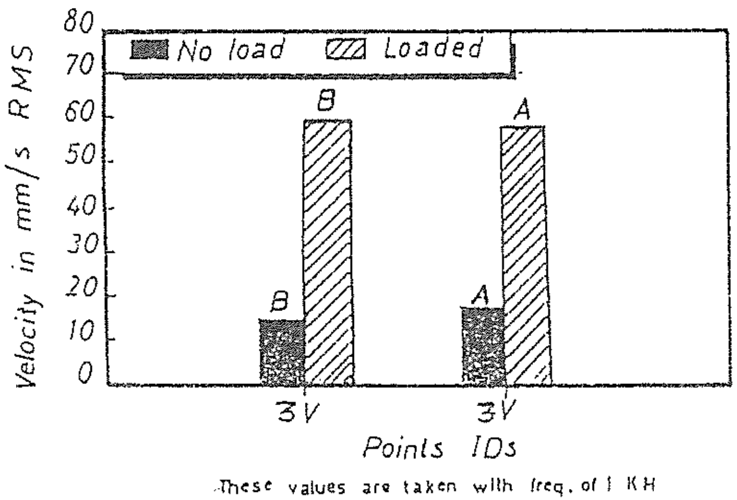


Fig.(16) Effect of loading condition on the cover of engines.[Comparison of point 3V on dozers at load/no_load].

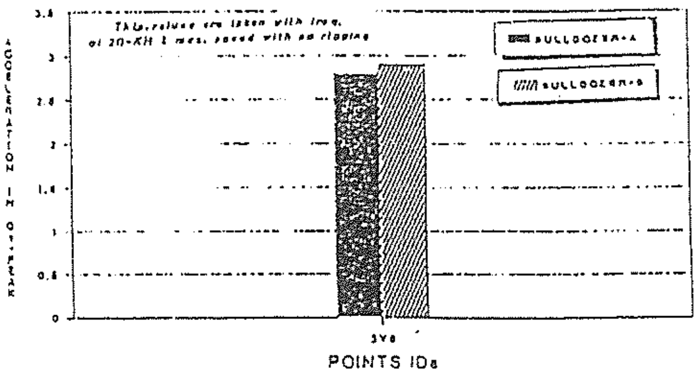


Fig.(15) Comparison between bulldozers A & B at max. speed.

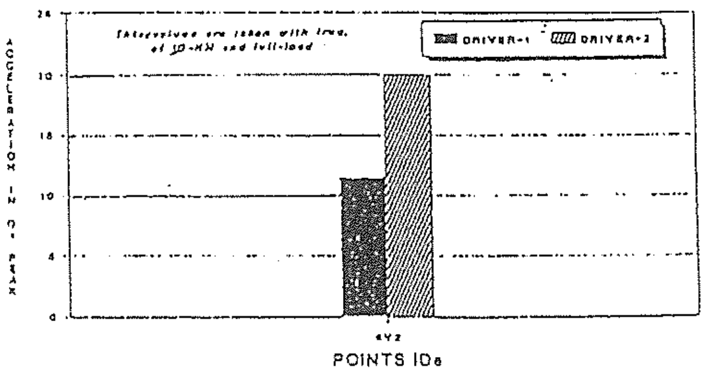


Fig.(17) Comparison between two drivers with dozer_A

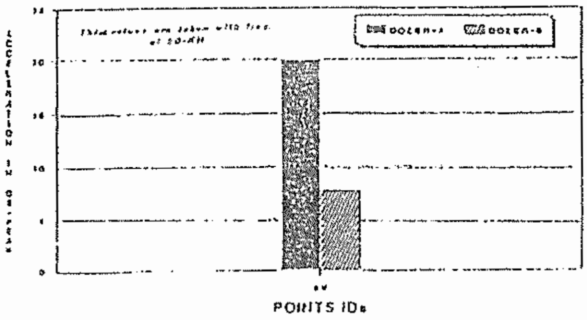


Fig.(18) Comparison between dozers A & B with driver (B).

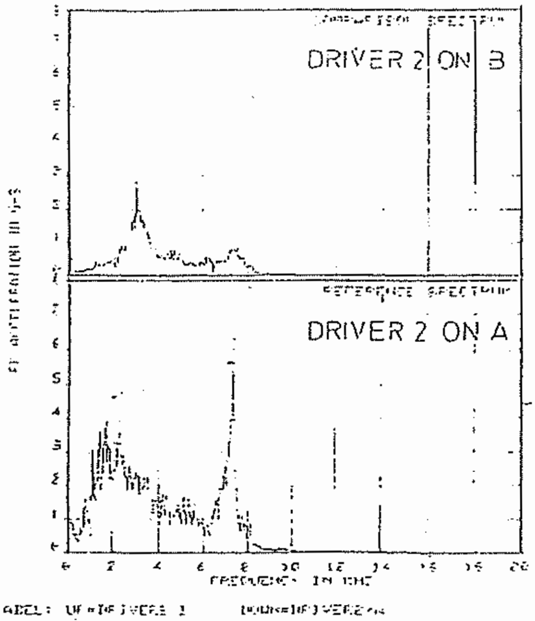


Fig.(20): Comparison Between Bulldozer A and Bulldozer B With the Same Driver (Driver of B)

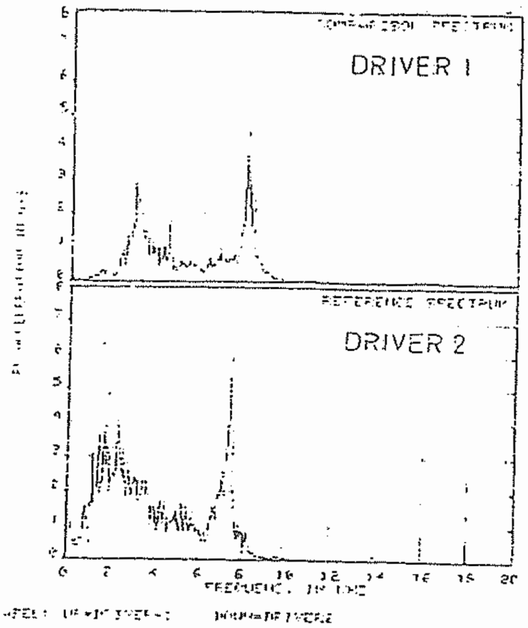


Fig.(19): Effect of Driver Habits
Driver 1 (of A) - Driver 2 (of B)
Both Driving Bulldozer A

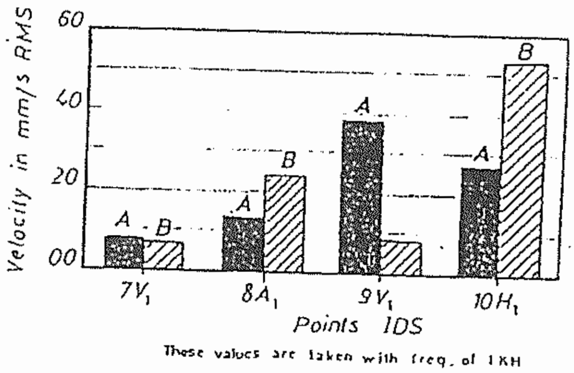


Fig.(21) Comparison between rippers of bulldozers A & B (Low frequency)

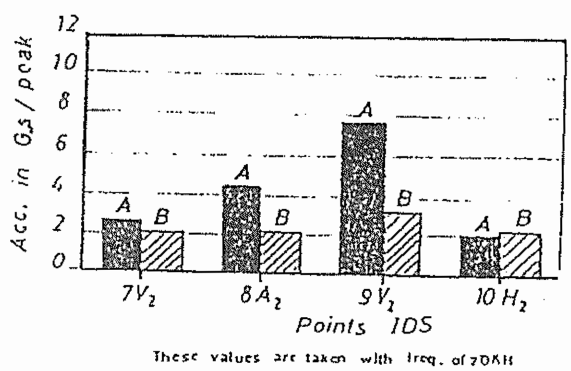


Fig.(22) Comparison between rippers of bulldozers A & B at full load (High frequency).

العنوان باللغه العربيه :

التحليل والتشخيص الأمثل للإهتزازات المنقوله في المنشآت الثقيله

المشتركون : مفرد

ملخص البحث :

لقد تم في هذا البحث تحليل وتشخيص مشكلة الاهتزازات العاليه والغير متوقعه في منصة الحفار وكسارة الحجارة. ولقد تم قياس الإهتزازات على نقاط مختاره بدقه واضعين في الاعتبار جميع ظروف التشغيل.

ومن تحليل النتائج وتشخيصها إتضح مايلي :

- أن هناك إهتزازات عالية على المنصه تختلف في شدتها من مكان لآخر بطول المنصة.
- كلا من مجموعة المضخه وتوصيلاتها بها إهتزازات عالية غير متوقعة بالنسبة لهذا النوع من المعدات .
- كلا من خطي السريان سواء Overflow أو Test pipeline بهما سريان مضطرب وتكهف يؤديان بالتالي إلى مستويات إهتزازات عالية .
- ليس هناك إرتباط بين الاهتزازات الموجوده على المنصه وإهتزازات الأنابيب .

ومن تحليل نتائج كساره الحجاره :

١- أن محرك البلدوزر (B) يعاني من مستويات إهتزازات عالية أثناء تشغيلة بدون حمل – ووضح من التحليل أن سبب هذه الاهتزازات هو عطب في مضخة التروس .

٢- إن الصدمات نتيجة لعمل الـ Ripper تختلف بإختلاف حجم ووضع الحجارة التي يتم تكسيرها.

٣- أظهرت القياسات أثناء تحرك كلا من البلدوزر (B, A) على الطريق أن قيم الاهتزازات الناتجة معتدلة.

وعلى هذا يمكننا أن نوصي بما يلي :-

- منطقة المنصه حول مجموعة المضخة تحتاج لتقويه في كلا الاتجاهين الرأسي والأفقي لتقليل إهتزازات المنصه.
- بالنسبة Piping Spans يجب إعادة ربطها بإضافة مواسك مرنه ووضع دعامات عند جميع الصمامات وأماكن الانحناءات في الأنابيب واختبار أطوال الأنابيب بكل دقه .
- يجب تصميم عازل خاص للإهتزازات يتم لحامه (Pressure Relief Valve) على خط السريان وذلك لامتصاص الاهتزازات العالية عند الصمامات .
- يوصي بعمل أبحاث أخرى لتحديد أنماط الإزاحه الناتجة من أجزاء كسارة الحجاره.