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Compressor Vibration Problems Due to Changed Gas Composition - A Case Study

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COMPRESSOR VIBRATION PROBLEMS DUE TO CHANGED GAS COMPOSITION A CASE STUDY

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ABSTRACT

The importance of specifying the exact range of gas composition that a compressor is required to handle is sometimes under-estimated. This situation often leads to severe problems, particularly if the variation happens to be wide enough.

The present paper discusses a case study of vibration problems faced on a multi-stage Reciprocating compressor handling a complex mixture of hydrocarbons. The vibrations were so severe that the compress-or could not be run continuously.

Thorough investigations into the causes of such severe vibrations led to the conclusion that the problems were due to changed gas composition. The methodology of investigations, analysis and further solution of the problem resulting into successful running of these compressors are presented in this paper.

The paper concludes with an appeal to all compressor users and consultants to specify as far as possible a realistic gas composition and in case there is any change later to inform the compressor manufacturer for carrying out an analysis of the changed gas composition so as to enable them incorporate nece-ssary changes to ensure trouble-free running of compressor.

INTRODUCTION

Reciprocating compressors generate flow pulsations which in turn give rise to press-ure pulsations. The reciprocating motion of the compressor piston generates the basic fundamental pulsation frequency corresponding to the compressor speed and passage of the gas through compressor valves gives rise to higher pulsation frequencies or harmonics corresponding to multiples of compressor speed.

The final build-up of amplitudes of individual pulsation components is controlled by the effects of the manifold and the piping system attached to the compressor. These pulsations sometimes get coupled to the mechanical system at points such as closed ends of vessels or pipes, bends, orifices, reducers, etc. to produce acoustic shaking forces. These acoustical shaking forces excite vibrations, the magnitude of which depends upon the location of mechanical natural frequencies relative to the frequencies of acoustical shaking forces and the amount of damping in the mechanical system.

The level of pulsations generated by a compressor, is related to a number of parameters which include compressor operating pressures, compression ratio, cylinder clearance volumes, phasing between cylinders, thermodynamic gas properties and compressor cylinder and valve design. These pulsations are further amplified by the attached piping systems. The consequent vibrations depends upon the acoustical and mechanical characteristics of the vessels and piping systems.

Gas composition is one of the most important parameters that must be exactly specified by the compress-or user. Severe problems may arise in case the actual gas composition is widely different from that specified. The present paper discusses a case study of vibration problems faced on a multi-stage Reciprocating compressor handling a complex mixture of hydrocarbons.

THE COMPRESSOR

The compressor involved was a three-stage Reciprocating Compressor compressing associated gas having composition as per Table 1 (as indicated by client in their order) from a suction pressure of 1.068 Kg/cm^2 absolute to a discharge pressure of 51.533 Kg./cm^2 abs. The scheme of compression is shown in Fig 1.

The compressor was designed to handle specified gas and the design was also checked by Analog study as per design approach 3 of API 618 - 1974, and modifications as required during study were incorporated in the final design. It was expected that this design should result in a trouble-free performance of the subject compressor.

THE VIBRATION PROBLEM

When the subject compressor was installed and commissioned on process gas, it was observed that there were heavy vibrations on 2nd stage cylinders and related suction and discharge dampners. These vibrations resulted into frequent failures of cooling water piping connected to 2nd stage cylinders.

A thorough investigation was carried out to establish the causes of these severe vibrations. Vibration measurements were taken at various locations with the help of B&K and IRD mech-analysis instrumentation.

A typical vibration measurement record is shown in table 2 and typical vibration spectra are shown in figs. 3, 4 and 5. (Fig. 2 shows locations of measurement points).

The vibration readings and also the vibration spectrum suggested a pre-dominance of 6th harmonic (i.e. around 2000 rpm - the compressor speed being 333 rpm). Based on these observations it was concluded that the underlying cause of these vibrations is related to some sort of acoustic resonance and the consequently high shaking forces inside 2nd stage suction and discharge dampners.

THE GAS COMPOSITION

At this stage of investigations the actual gas composition of the gas being handled by the compressor was checked. It was observed that there was a wide variation in the actual gas composition from that specified by client at the time of order, which was used for design.

Thus it was concluded that the real reason for the subject vibration problems was the change in the composition of gas being handled, which resulted into generation of high frequency shaking forces with consequent forced vibrations of the compressor cylinders, and vessels, particularly of the second stage. With the initial gas composition the shaking forces on various volume bottles were having very low values (refer to table 3) which were not expected to cause any undue vibrations and as such no internals were anticipated for force balancing in these vessels.

However, due to the changed gas composition and the resulting changes in the acoustic behaviour of the gas mixture, the subject dampners of 2nd stage suction and discharge respectively were subjected to high unbalanced forces of 6th harmonic and this was the reason of vibrations of the second stage cylinders and vessels.

THE SOLUTION

As a first step towards solution of the problems, the clients were convinced about the role of gas composition in this vibration problem. The client as a result made detailed investigations into the actual and possible gas composition at this site. Ultimately, the client specified a wide range of gas composition that could possibly be involved during the running of subject compress-ors over years. (Ref. table 4)

The manufacturers, then endeavoured to make suitable modifications in the design of the subject compressors so as to take care of this newly specified range of gas composition. It was impractical to have a detailed analytical study inclusive of Analog study, both because of the huge variety of possibilities of different combinations of gas mixtures being involved in the wide range of gas composition specified by client and as well because of the prohibitive costs and time required for such an analytical study.

In view of above it was decided to adopt a qualitative solution, that was well proven by manufacturer's experience and as such internals were designed and introduced in suction and discharge vessels of 2nd stage and also suction vessel of 3rd stage, for force balancing of these vessels over a wide range of gas composition as specified by the client. This solution involves shifting pulsation origination point inside a dampner, approximately to the central location so that the pulsations travel path to both ends of the vessels equalize and consequently the shaking forces are reduced to a minimum. This solution is equally useful at all possible gas composition.

The compressors were run successfully after these vibrations and the overall vibration level was reduced to safe limits. (Refer Table 5) These compressors have already registered more than 8000 hrs. of cummulative run without any further problems.

CONCLUSION

The author wishes to make an appeal to all compressors users to appreciate the importance of specifying exact gas composition range for the service involved. This allows the compressor designer to take care of the whole range while maintaining optimum performance of the compress-or. Any inaccurate specification of the range of gas composition, may lead to troubles and costly downtime, as in the case illustrated here.

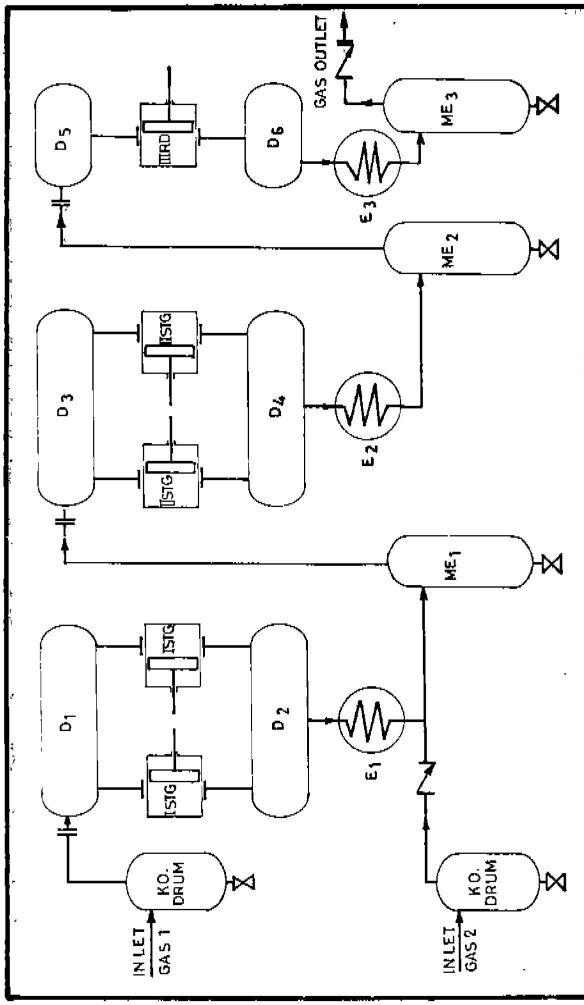


FIG. 1. SCHEME OF COMPRESSION

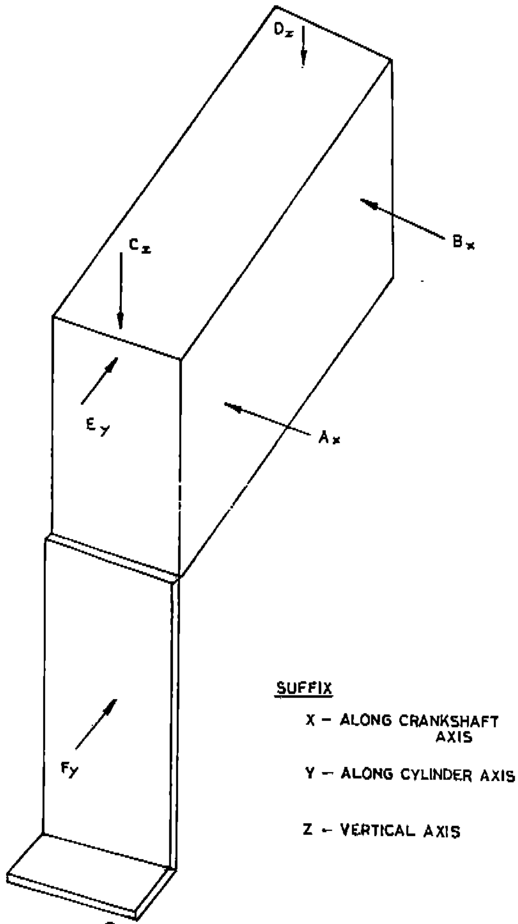


FIG. 2 LOCATIONS OF VIBRATION MEASURMENT POINTS

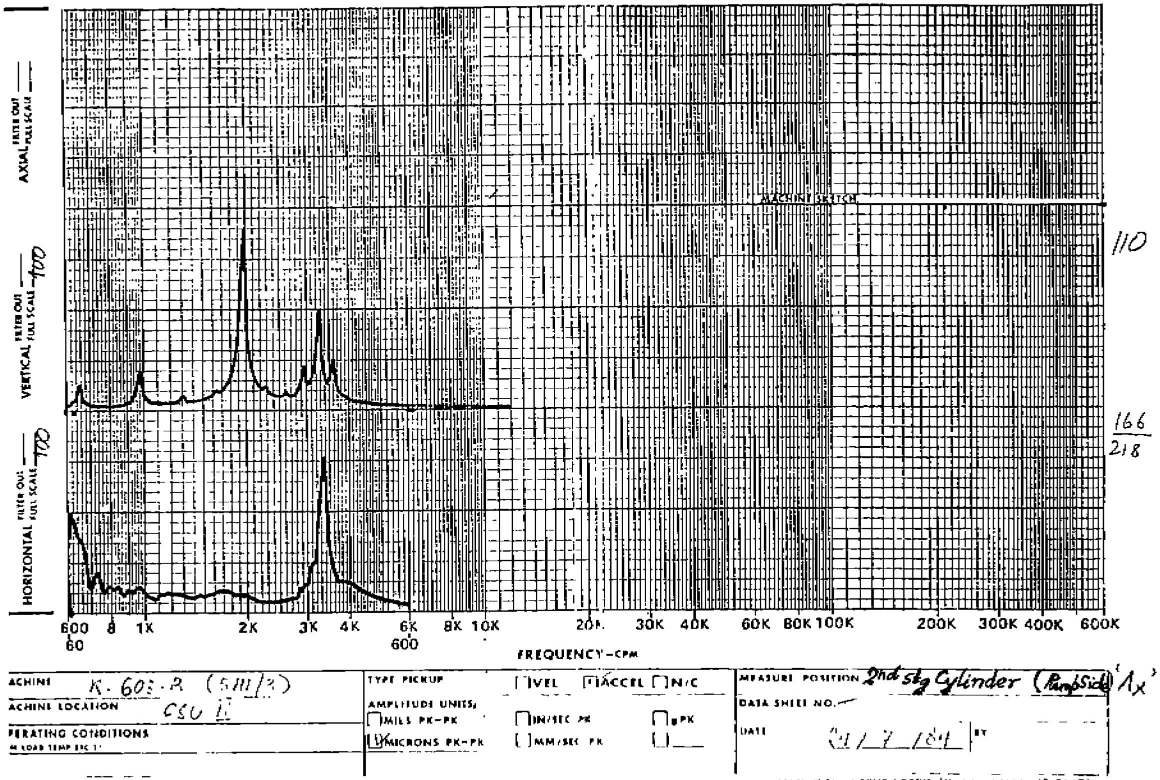

AXIAL FILTER OUT FULL SCALE
VERTICAL FILTER OUT FULL SCALE 400
HORIZONTAL FILTER OUT FULL SCALE 100
MACHINE SKETCH
110
166
218
600
60
8
1K
2K
3K
4K
6K
600
8K
10K
20K
30K
40K
60K
80K
100K
200K
300K
400K
600K
FREQUENCY-CPM
MACHINE K-603-R (5111/3)
MACHINE LOCATION CSU II
OPERATING CONDITIONS
TYPE PICKUP
VEL
ACCEL
N/C
AMPLITUDE UNITS:
MILS PK-PK
MICRONS PK-PK
IN/SEC PK
MM/SEC PK
g PK
MEASURE POSITION 2nd stg Cylinder (Pump Side) 'Ax'
DATA SHEET NO.
DATE
BY

FIG. 3

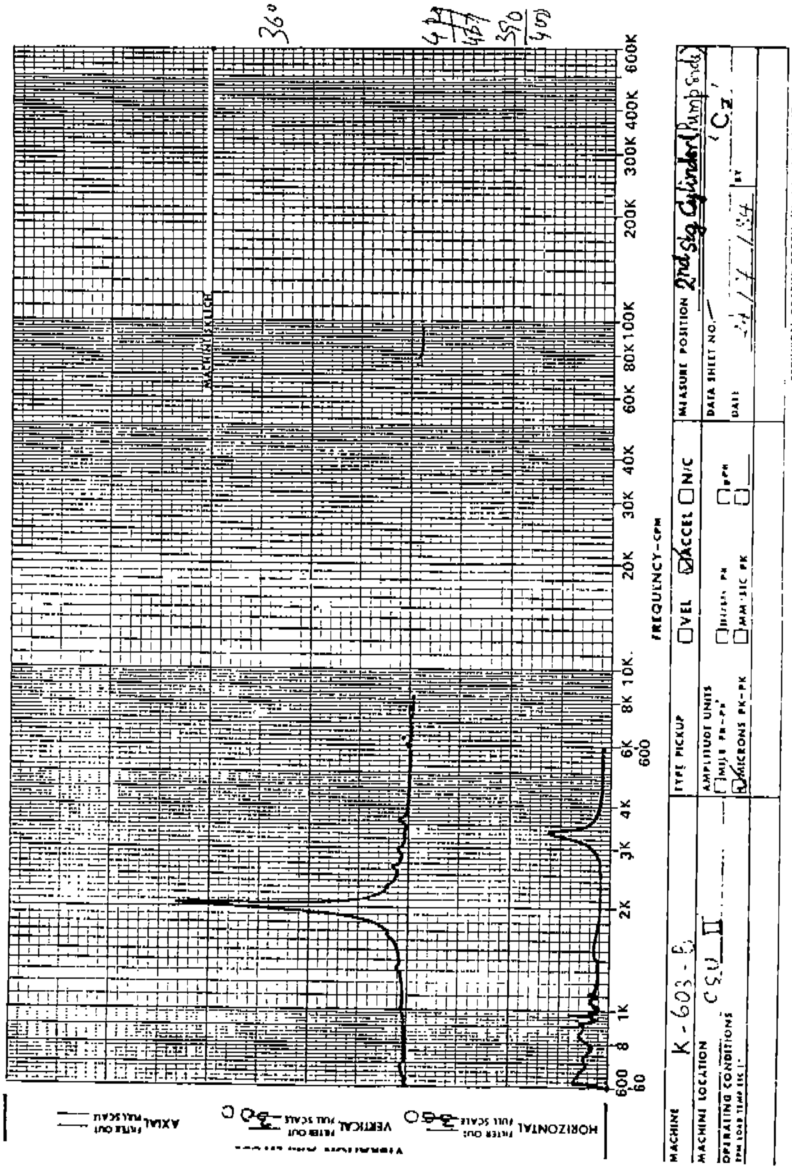
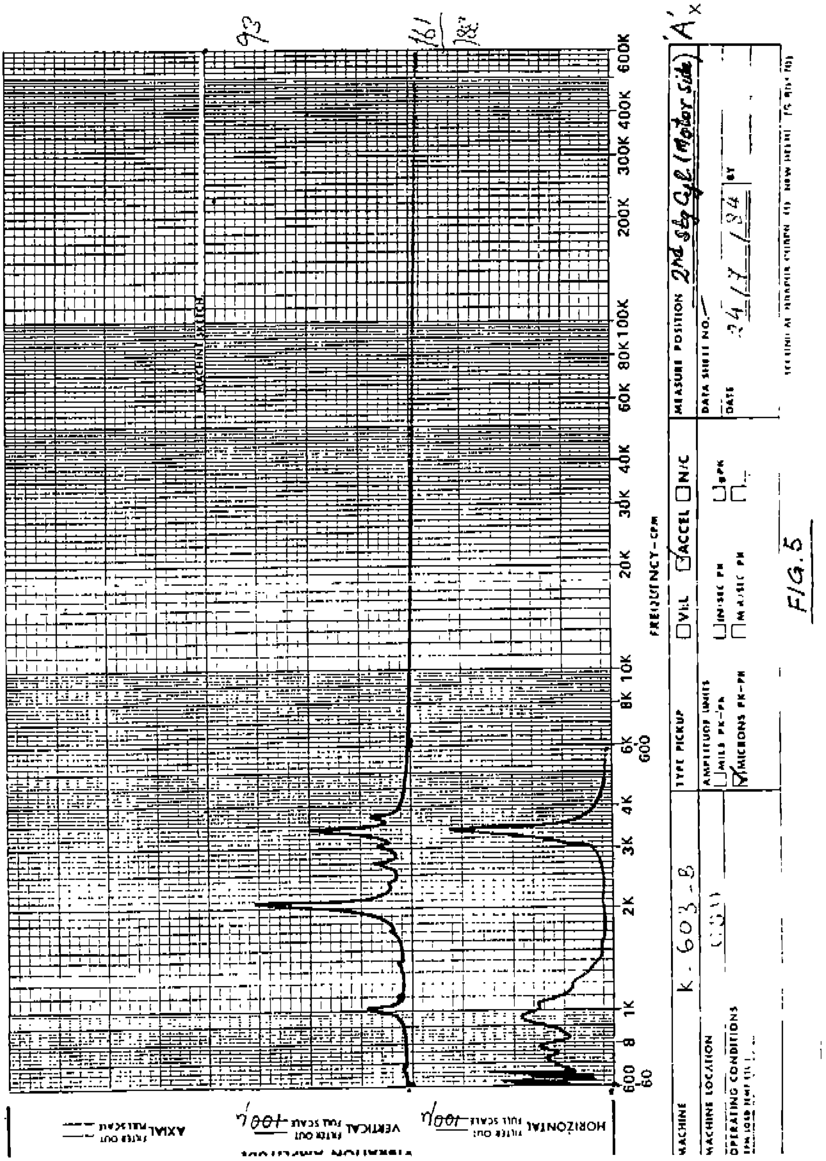
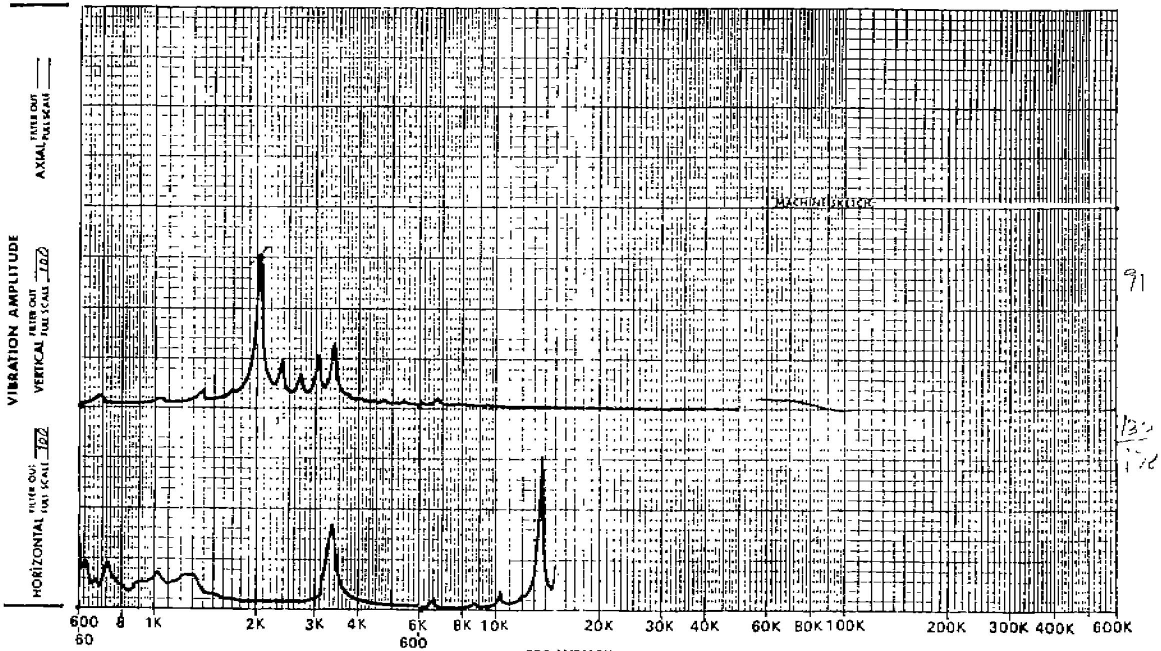

FREQUENCY-CPM
MACHINE K-603-B
MACHINE LOCATION CSU II
OPERATING CONDITIONS
MEASURE POSITION 2nd Stg Cylinder (Pump end)
TYPE PICKUP
VEL
ACCEL
N/C
AMPLITUDE UNITS
MICRONS PK-PK
DATA SHEET NO.
DATE
HORIZONTAL
VERTICAL
AXIAL
MACHINE SKETCH

FIG. 4



93
161
600K
400K
300K
200K
100K
80K
60K
40K
30K
20K
10K
8K
6K
600
4K
3K
2K
1K
600
60
FREQUENCY-CPM
HORIZONTAL
VERTICAL
AXIAL
100μ
MACHINE
K. 603 B
MACHINE LOCATION
OPERATING CONDITIONS
TYPE PICKUP
VEL
ACCEL
N/C
AMPLITUDE UNITS
MILS PK-PK
MICRONS PK-PK
MEASURE POSITION
2nd stg Cyl (Motor side)
DATA SHEET NO.
DATE
BY
A

FIG. 5



MACHINE	K-6(3 8 (5412/3)	TYPE PICKUP	☐ VEL	☑ ACCEL	☐ N/C	MEASURE POSITION 2nd Cyl.(Motor Size) C
MACHINE LOCATION	(.71	AMPLITUDE UNITS ☐ MILS PK-PK ☑ MICRONS PK-PK	☐ IN SEC PK ☐ MM/SEC PK	☐ PPK ☐		DATA SHEET NO —
OPERATING CONDITIONS RPM LOAD TEMP ETC						DATE 24/7/84 BY

TECHNICAL GRAPHS CORPN. NEW DELHI

FIG. 6

TABLE 1

INITIAL GAS COMPOSITION

The compress-ors shall handle Gas 1 from 1st to 3rd stg and Gas 2 from 2nd to 3rd stage. The gas compositions (Mol %) of Gas 1 and Gas 2 are as follows:

	Case I		Case II	
Component	Gas 1	Gas 2	Gas 1	Gas 2
CO2	2.71	3.01	2.81	3.01
C1	14.74	54.20	15.11	54.20
C2	21.52	20.74	22.10	20.74
C3	31.76	13.79	31.19	13.79
IC4	6.80	2.14	6.50	2.14
N C4	10.90	3.16	10.46	3.16
I C5	2.99	0.78	2.91	0.78
NC5	0.70	0.19	0.70	0.19
C6 +	7.89	1.99	8.22	1.90

TABLE 2

VIBRATION MEASUREMENTS BY B&K INSTRUMENTS (MICRONS PEAK TO PEAK)

LOCATION	1ST STG CYL (PUMP SIDE)	1ST STG CYL (MOTOR SIDE)	II STG CYL (PUMP SIDE) Overall	II STG CYL (PUMP SIDE) Major Peaks Observed	II STG CYL (MOTOR SIDE) Overall	II STG CYL (MOTOR SIDE) Major Peaks Observed	III STG CYL Overall	III STG CYL Major Peaks Observed
A_X	100	100	220	70 at1xRPM 110 at1xRPM	180	70 at1xRPM 80 at6xRPM	230	160 at 1xRPM 40 at 6xRPM
B_X	-	-	180	50 at1xRPM 65 at6xRPM	140	-	120	-
C_Z	45	40	360	65 at1xRPM 280 at6xRPM	340	75 at1xRPM 280 at6xRPM	220	70 at 1xRPM 80 at 6xRPM 130 at 9xRPM
D_Z	-	-	260	-	240	-	160	-
E_Y	180	185	280	220 at 1xRPM 80 at 6xRPM	240	-	340	300 at 1xRPM 45 at 8xRPM
F_Y	240	240	550	95 at 1xRPM 360 at 6xRPM	500	-	400	85 at 1xRPM 50 at 3xRPM 300 at 8xRPM

NOTE: Measurement locations as shown in fig. 2 .
R P M = 333.

TABLE 3

SHAKING FORCES ON COMPRESSOR DAMPENERS
(BASED ON INITIAL GAS COMPOSITION)

I STG. SUCTION DAMPNER (D1)	: $\pm$ 510 kg
II STG. DISCHARGE DAMPNER (D2)	: $\pm$ 380 kg
III STG. SUCTION DAMPNER (D3)	: $\pm$ 560 kg
II STG DISCHARGE DAMPNER (D4)	: $\pm$ 520 kg
III STG SUCTION DAMPNER (D5)	: $\pm$ 595 kg
III STG DISCHARGE DAMPNER (D6)	: $\pm$ Negligible.

TABLE 4

FINAL RANGE OF GAS COMPOSITION (MOLE PERCENT)

COMPONENT	FIRST SUCTION (Press. 1.068 kg/cm^2 abs.)	SECOND SUCTION (Press. 4.013 kg/cm^2 a)	THIRD SUCTION (Press. 15 kg/cm^2 a)
CO2	2.51 - 3.71	2.81 - 3.63	3.21 - 4.03
C1	20.04 - 44.11	34.06 - 48.62	38.83 - 50.10
C2	13.40 - 23.00	15.45 - 21.59	16.87 - 23.70
C3	18.70 - 28.20	17.20 - 22.60	17.20 - 22.65
IC4	3.70 - 4.47	2.62 - 3.57	2.54 - 3.01
NC4	3.24 - 5.04	2.80 - 4.31	2.63 - 3.33
IC5	0.39 - 0.74	0.33 - 0.54	0.27 - 0.33
NC5	0.30 - 0.63	0.26 - 0.49	0.19 - 0.24
NC6	0.12 - 0.28	0.10 - 0.17	0.04 - 0.07
NC7	0.01 - 0.03	0.01 - 0.01	0.00 - 0.002
C8 +	6.33 - 18.98	5.72 - 12.58	3.85 - 4.97

TABLE 5

COMPRESSOR OVERALL VIBRATION READINGS
(MICRONS PEAK TO PEAK)

		I STG CYL. (PUMP SIDE)	I STG CYL. (MOTOR SIDE)	II STG CYL. (PUMP SIDE)	II STG CYL. (MOTOR SIDE)	III STG CYL.
BEFORE MODIFICATIONS	X	110	100	180	150	230
	Y	180	185	300	330	340
	Z	45	40	360	340	220
AFTER MODIFICATIONS	X	140	110	110	100	100
	Y	220	200	220	200	340
	Z	60	60	80	100	130

X axis is along crank-shaft axis.
Y axis is along cylinder axis.
Z axis is along vertical axis.