



***INTERNATIONAL
MATERIALS
HANDLING
CONFERENCE***

1989

High Speed Conveyor Idlers

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BELTCON 5

1.0 INTRODUCTION

For the purpose of this paper on High Speed Conveyor Idlers we will be taking results obtained from actual on-site tests and tests carried out at Research Laboratories. In addition to these, a number of vibration test results from our in-house test bed will be considered. From on-site experience and the conclusions obtained from the above tests, the final minimum requirements for High Speed Idlers have been ascertained.

During the development of SABS 1313, conveyor systems were running at less than 4 m/s (787,2 ft/min) and the set-down limits and tolerances were in keeping with this requirement.

With the ever-increasing demand for higher capacity conveyors, it has become necessary for the conveyor industry to increase belt widths and belt speeds to a point where 6 m/sec (1181,4 ft/min) is becoming common practice.

These increased speeds and belt widths have introduced a number of problems, which had not been encountered on the lower capacity conveyors.

This has forced the conveyor idler manufacturer to take a more scientific approach to the selection, design and manufacture of both the rolls and bases in order to meet the requirements of users and still produce an idler at a realistic price.

2.0 SCOPE OF PRESENTATION

It is intended to consider only the implications of conveyors running at speeds in excess of 4 m/sec.

As there are conveyor systems operating in South Africa at speeds in the region of 6 m/sec, this paper will use this speed in all given examples. The relevant idler-roll speeds for 6 m/sec are:-

(a) 127mm Dia = 902,3 RPM

(b) 152mm Dia = 753,9 RPM

(c) 165mm Dia = 694,5 RPM

For practical purposes the more common idler roll size of 152mm dia manufactured with a 6mm wall thickness tube will be used.

3.0 GENERAL

The standards given in SABS 1313 which are considered unacceptable for high speed idlers are the Total Indicated Runout (T.I.R.) or Peripheral runout of the rolls, which is represented by the formula:

$$\frac{\text{Roll length} + 0,55}{600}$$

and the out-of-balance forces created by this T.I.R.

All vibration tests and results use the Mird-Mechanalysis International vibration severity chart as a basis of comparison (see graph).

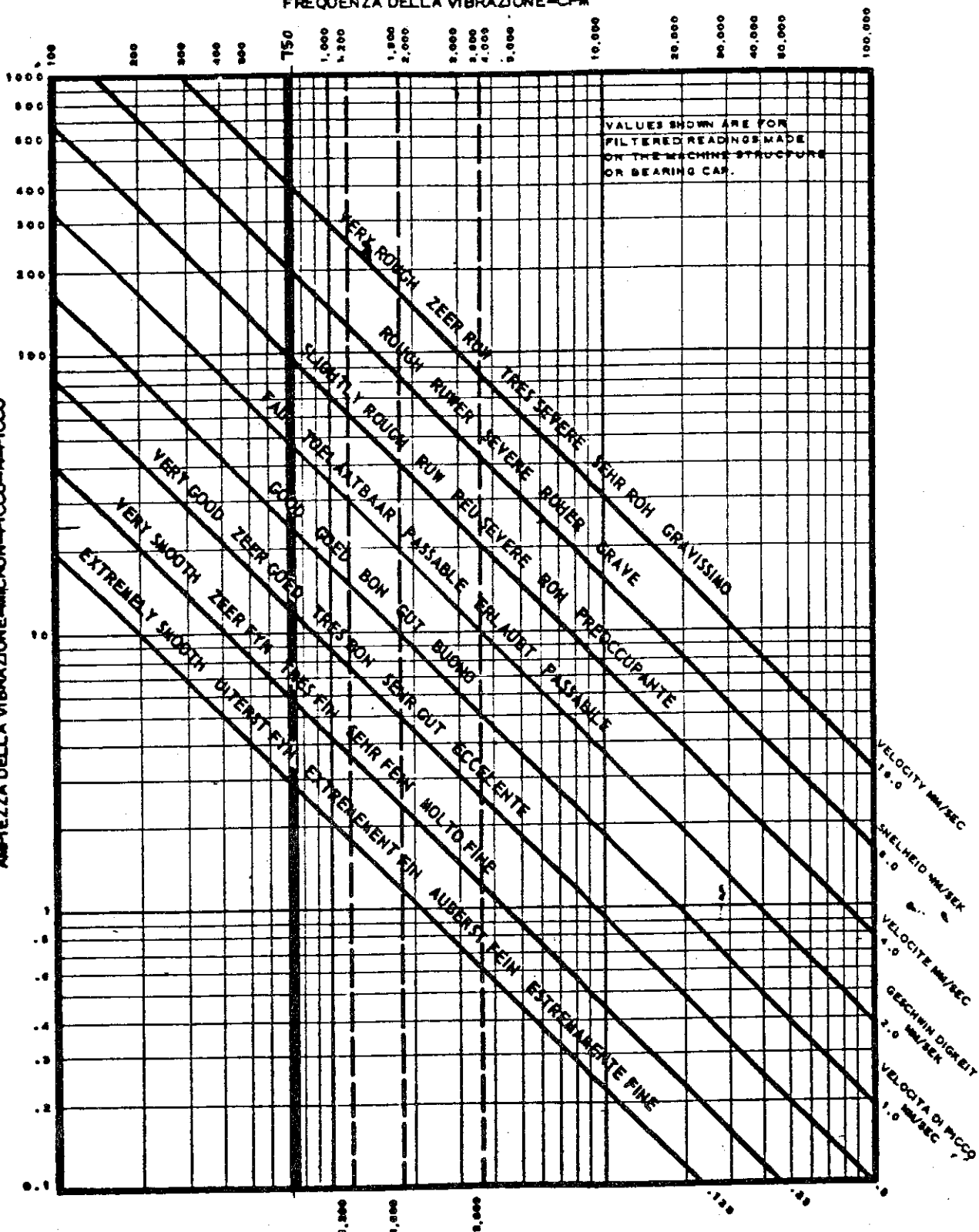


IRD-MECHANALYSIS INTERNATIONAL

GENERAL MACHINERY VIBRATION SEVERITY CHART
 TRILLINGS STRENGHEIDSKAART VOOR ALGEMENE MACHINES
 TABLEAU DE SEVERITE VIBRATOIRE POUR LES MACHINES EN GENERAL
 SCHWINGUNGSSTRENGE TABELLE FÜR ALLGEMEINE MASCHINEN
 NOMOGRAMMA GENERALE DI PERICOLOSITA DELLA VIBRAZIONE

VIBRATION FREQUENCY—CPM
 TRILLINGS FREKVENTIE—CPM
 LES FREQUENCES DE VIBRATION—CPM
 SCHWINGUNG FREQUENZEN—CPM
 FREQUENZA DELLA VIBRAZIONE—CPM

VIBRATION DISPLACEMENT—MICRONS—PEAK-TO-PEAK
 TRILLINGS VERPLAATSIJNG—MICRONS
 DEPLACEMENT DE VIBRATION—MICRONS
 SCHWINGUNGS VERLAGERUNG—MICRONS
 AMPIEZZA DELLA VIBRAZIONE—MICRON—PICCO—A—PICCO



Confirming the above is an extract from Mr. R.N. Taylor's paper "Practical and Operating aspects of the Richards Bay Coal Terminal (case study) dated 3-9-85.

FINDINGS OF INVESTIGATION INTO CAUSES OF IDLER ROLL FAILURE

The investigation into the idler roll failure revealed two rather unusual findings. Over and above the usual problems imposed upon the idler roll by economic, design and manufacturing constraints and the execution of these, there were other significant factors which contributed to the idler roll demise. These were:

- out of balance considerations of idler roll
- idler roll end cap rigidity.

(1) Out of balance considerations

With belt conveyors moving with the speed of those of the Coal Terminal, out of balance of idler rolls seems to contribute most significantly to idler roll failure. It is believed that this problem is not particular to or compounded by the use of taper roll bearings but is equally true for any bearing selection.

(2) Idler Roll end Cap Rigidity

The Coal Terminal's access conditions are often such to necessitate a fair measure of rehandling and ultimate hand transportation. Unfortunately the Coal Terminal rolls are relatively heavy and awkward. These two factors tend to result in the idler rolls being dropped or receiving blows to the end cap or exposed shafting.

The following graph Figure 1 shows the rigidity of some commercially available idler roll end caps. As can be seen considerable variation in idler end cap rigidity exists between manufacturers."

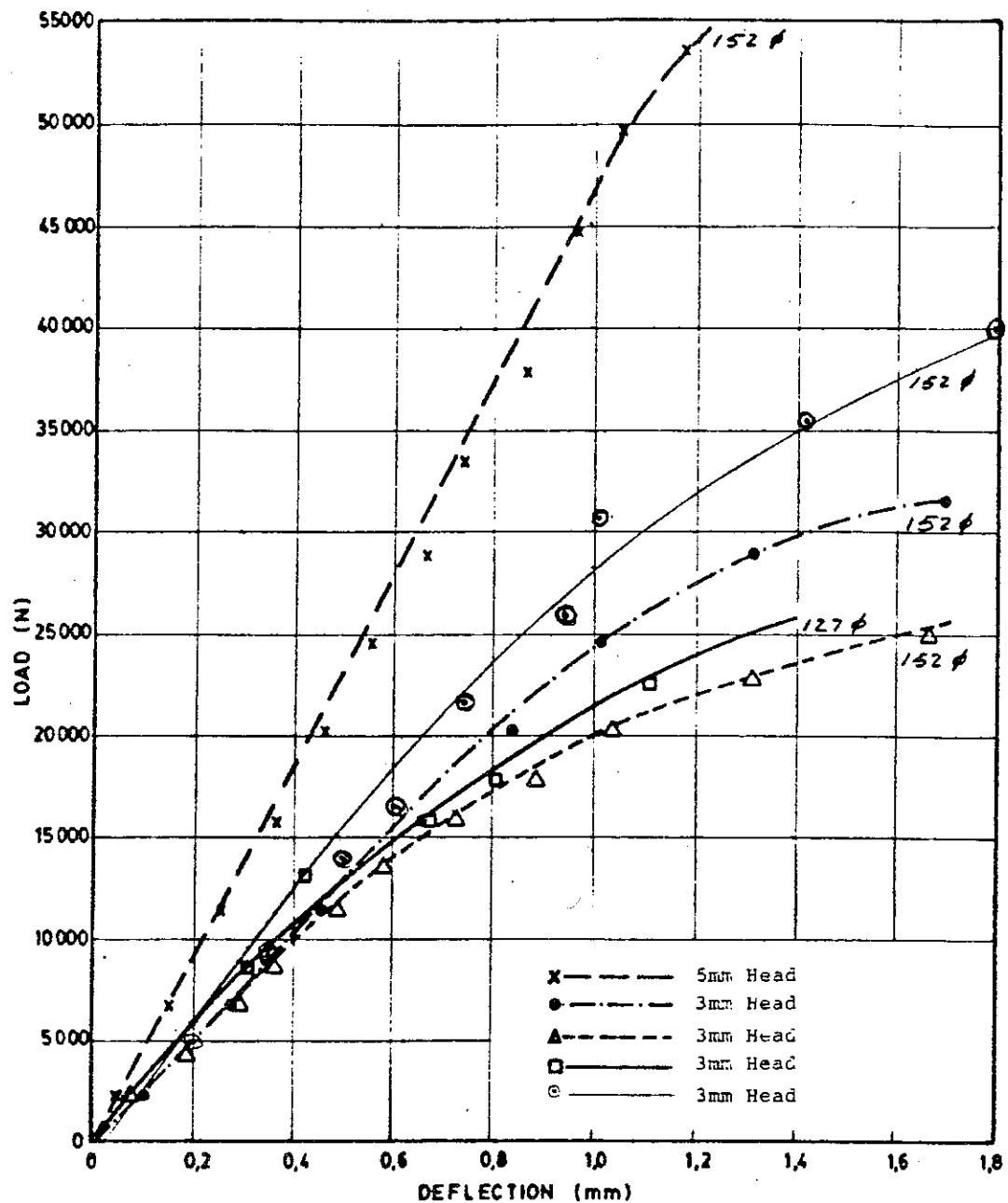


FIG. 1 LOAD-DEFLECTION CHARACTERISTICS OF 5 MAKES OF CONVEYOR IDLER HEADS

4.0 IDLER SELECTION

In the selection of idlers for high speed conveyors it is necessary to take into account all the factors required for the selection of idlers for any standard conveyor system. However, in addition it is essential to consider the overall conveyor structure for alignment and rigidity to ensure minimal vibration.

In order to complete an accurate idler selection the following information is necessary:

- (a) Conveyor capacity
- (b) Type of material
- (c) Bulk density
- (d) Maximum lump size
- (e) Belt speed
- (f) Belt width
- (g) Belt type & mass
- (h) Belt tensions T_1 & T_2
- (j) Environmental conditions

From the above information we can determine:

- (a) The most suitable idler configuration (3 Roll, 5 roll and trough angle)
- (b) Idler spacing
- (c) L-10 bearing life

(d) Shaft deflection due to load

(e) Idler base deflection

5.0 Possible problem areas which we consider should be looked at are listed together with the relative comments. (Taken from actual tests)

PROBLEM	COMMENT
1. End Cap Deflection	Limited importance.
2. Undersize/Oversize Shaft Specification.	Not an important factor.
3. Back Seal Design and Bearing Grease.	Small contributory factor to failures.
4. Bearing End Float Setting. (Taper Roller Bearing)	Small contributory factor to failures.
5. Bearing Seating.	Not a factor.
6. Backseal Damage on Assembly.	Not an important factor.
7. Looseness of Bolts Holding down Idler Bases.	Contributory factor to failures.

8. Conveyor Support Structures.	Could be a major contributory factor to failures. If structures are not stiff enough.
9. Flexibility of Conveyor Bases.	Major contributory factor. Flexibility of bases needs to be reduced to a minimum.
10. Belt tracking.	Small contributory factor.
11. T.I.R. and Unbalance of Rolls.	Major contributing factor to all failures.

From the above list it can be seen that the problems which cause the idler manufacturer most concern are:

- (a) End cap deflection
- (b) Loose base holding down bolts
- (c) Bases
- (d) Balance / T.I.R.

From our findings on-site it was evident that if the idler selection was correct the highest failure rate would take place on the least loaded rolls. In effect the rolls which have least load are being subjected to the highest T.I.R./out-of-balance forces, causing very high vibrations which in turn cause premature bearing failure.

Any one of the above points on its own is not a major problem, but in practice they combine to form a major problem.

(a) End Cap Deflection (Fig. 1 & 2)

Static testing in a press at Boart Research Centre indicated that a force of approximately 2 tonnes was necessary to cause sufficient plastic deformation of a 3mm thick end cap, to induce excessive end float in the bearings of the idler rolls.

An interesting point to come out of the investigation was that because of the weight of wide belt conveyor rolls, they do not have to be dropped from any great height in order to induce dynamic forces which are of the same magnitude as the 2 tonnes static force. The implication of this is that bad handling of the rolls somewhere between the factory and the site could result in excessive end float being induced in the roll. It is interesting to note that a 5mm thick head requires a static force of approximately 5,5 tonnes before any plastic deformation results. This characteristic would be very desirable in a head used on a taper roller bearing roll. However, if an idler roll were to be subjected to this type of shaft-end loads, major damage would be done to the bearings and premature failure would result.

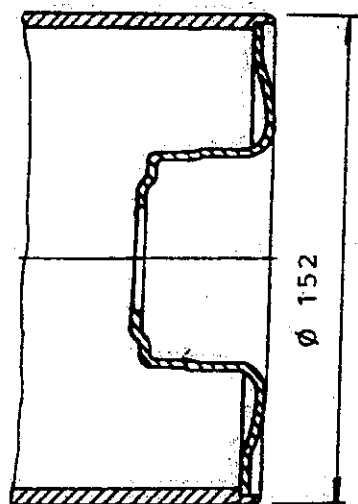
(b) Looseness of Bolts holding down Idler Bases (Fig. 3)

Vibration analysis carried out revealed that with loaded conveyors no extra vibration resulted from the bolts being loose. However, with the belts running unloaded the idler bases tended to vibrate excessively in the direction of belt travel. This would explain why some bases crack near their mounting points.

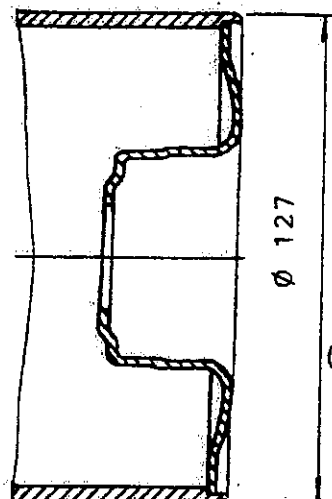
(c) Conveyor Idler Bases (Fig. 4, 5 & 6)

Figures 4 and 5 give an indication of the vibration levels present at various points on an idler base. The readings show the vibration levels on the top of the side wing roll support as well as on the gusset plates of a base which is too flexible.

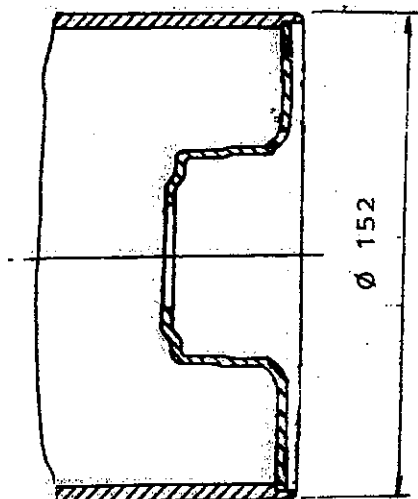
THE FIVE HEADS TESTED ARE AS FOLLOWS



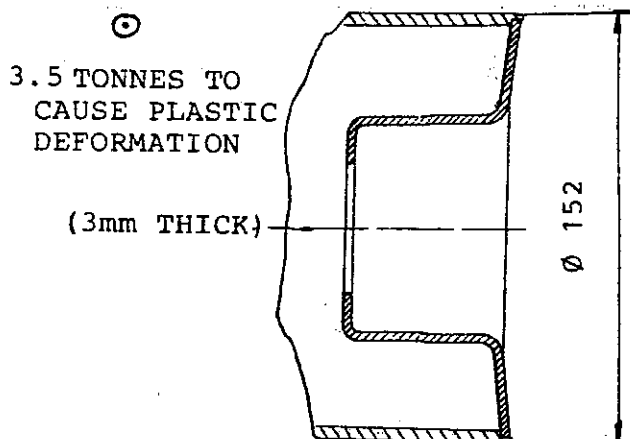
Δ
2 TONNES TO
CAUSE PLASTIC
DEFORMATION
(3mm THICK)



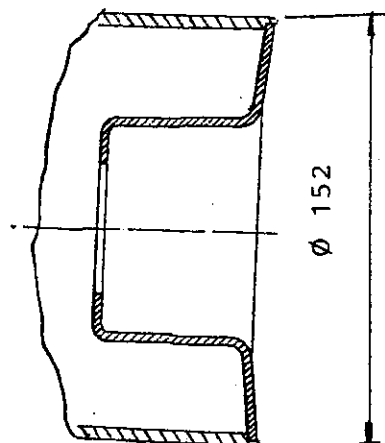
□
2.4 TONNES
TO CAUSE
PLASTIC
DEFORMA-
TION
(3mm THICK)



3 TONNES TO CAUSE
PLASTIC DEFORMATION
(3mm THICK)



⊙
3.5 TONNES TO
CAUSE PLASTIC
DEFORMATION
(3mm THICK)



x
5.5 TONNES
TO CAUSE
PLASTIC
DEFORMATION
(5mm THICK)

THE END CAP DEFLECTION WAS NOT CONSIDERED A MAJOR PROBLEM AS IT IS FELT THAT IF THE SHAFT END OF AN IDLER ROLL WERE SUBJECT TO THIS TYPE OF SHOCK LOADS THE BEARINGS WOULD BE DAMAGED

FIG. 2

LOOSE IDLER FRAME SECURING BOLTS

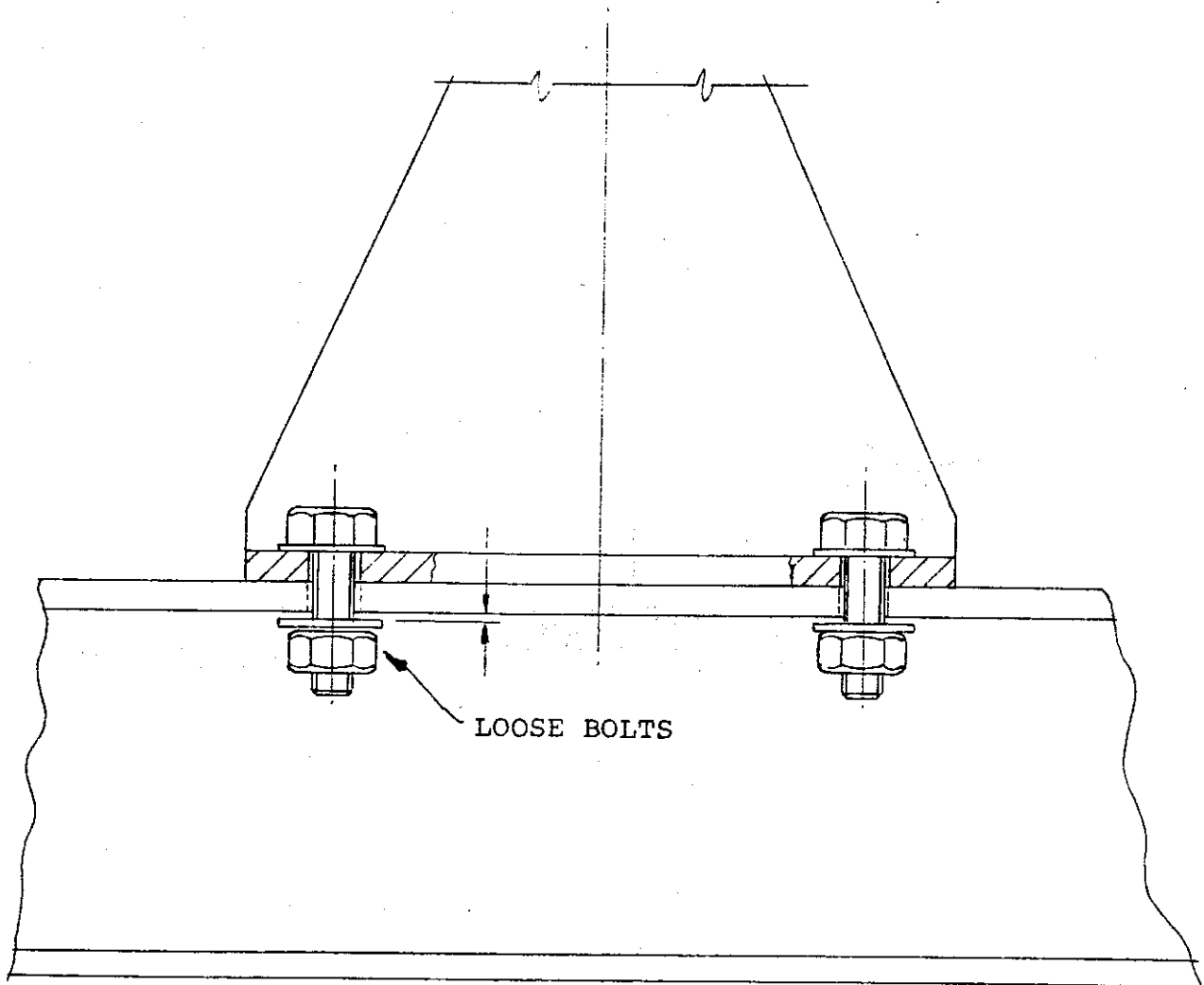
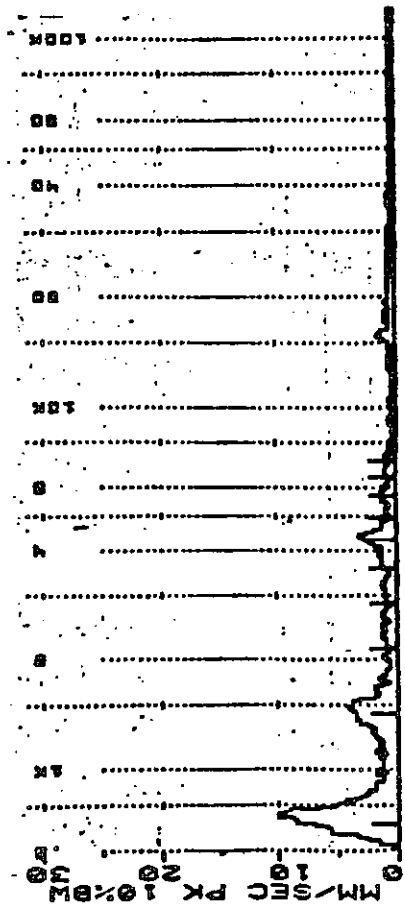
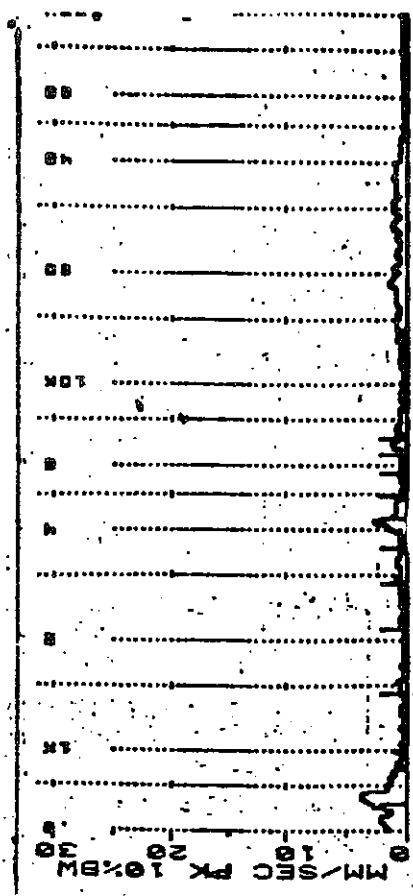


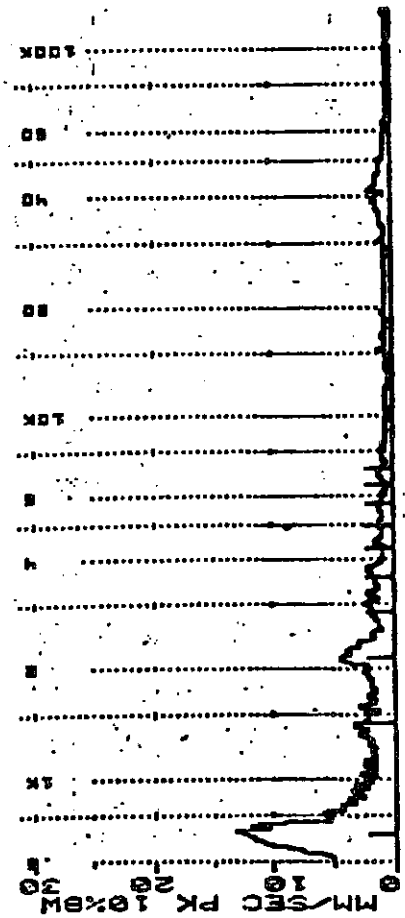
FIG. 3



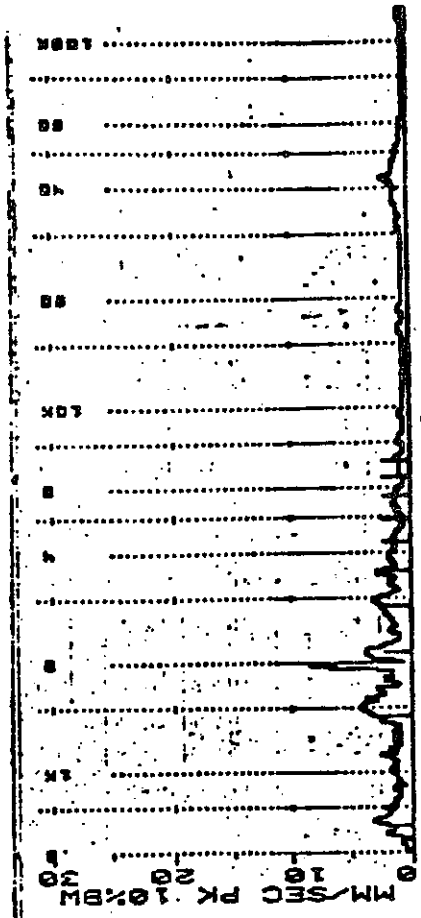
1



2



3



4

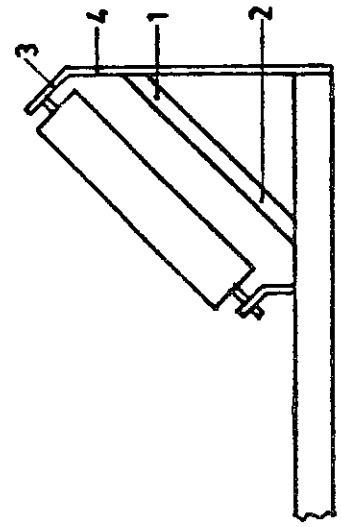


FIG. 4 VIBRATION LEVELS — TYPICAL BASE IN USE WITH 1800 mm WIDE BELT

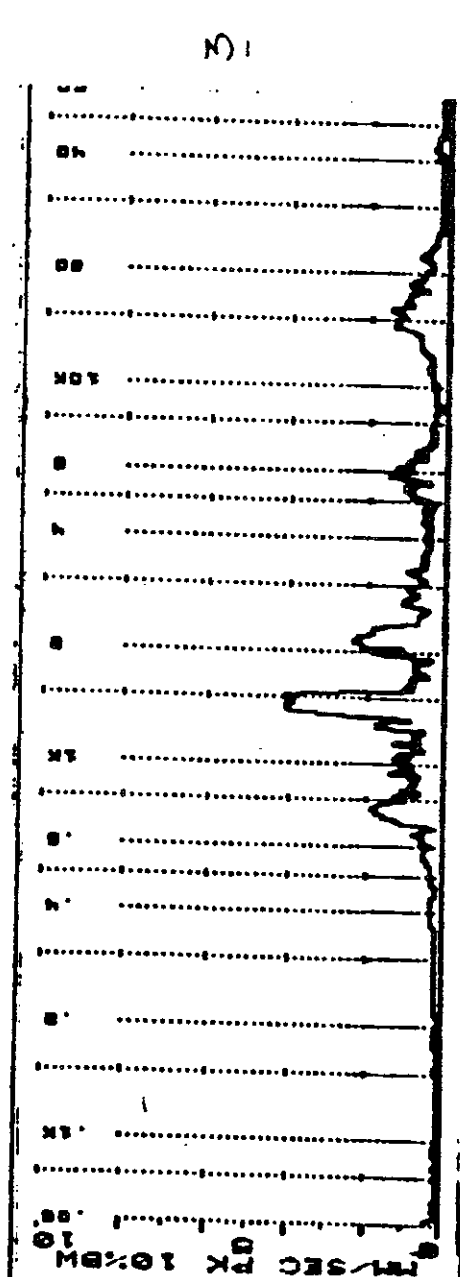
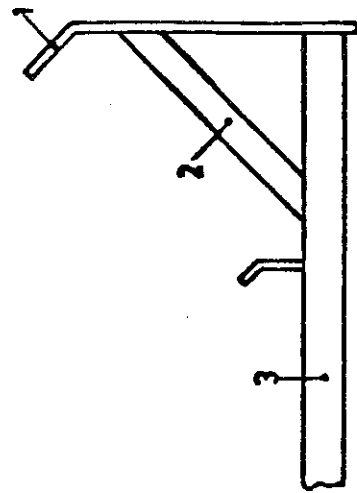
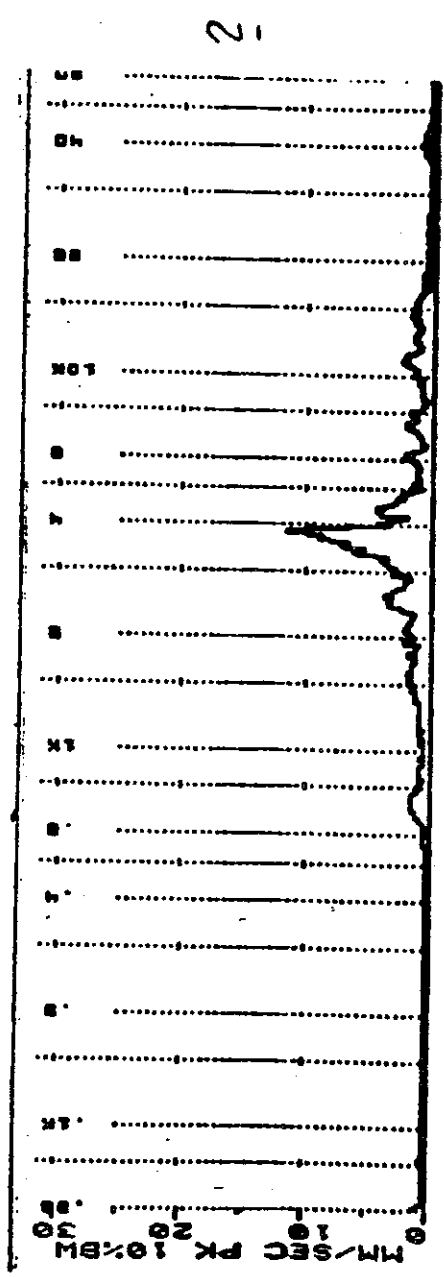
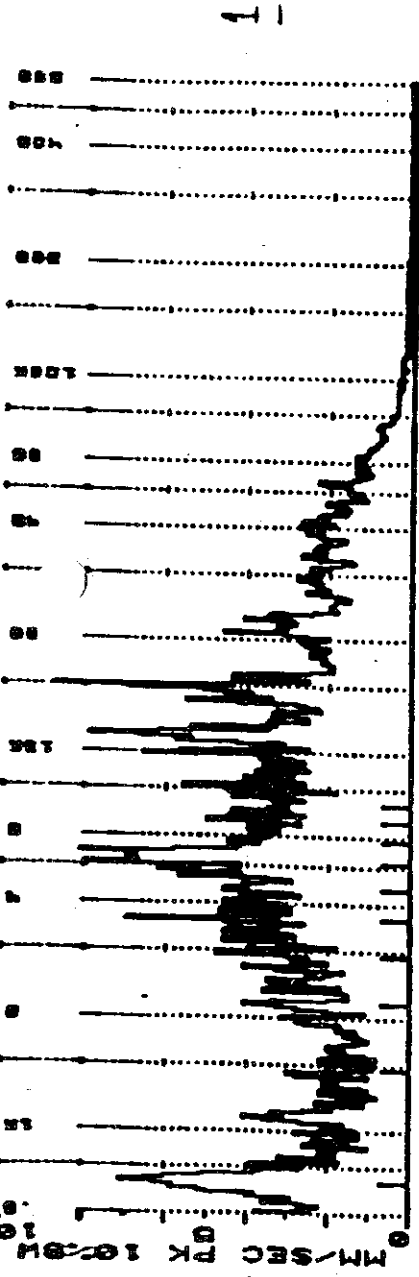


FIG. 5 VIBRATION LEVELS ON A TYPICAL BASE IN USE WITH 2200 mm WIDE BELT

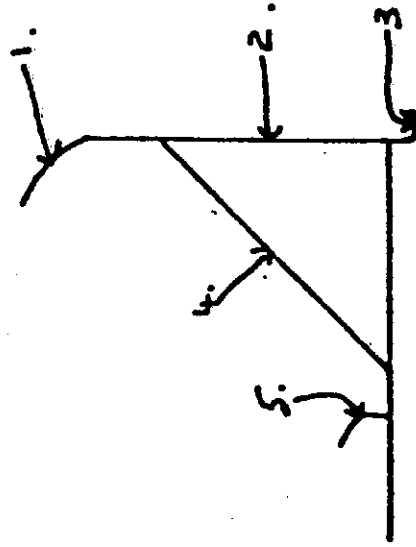
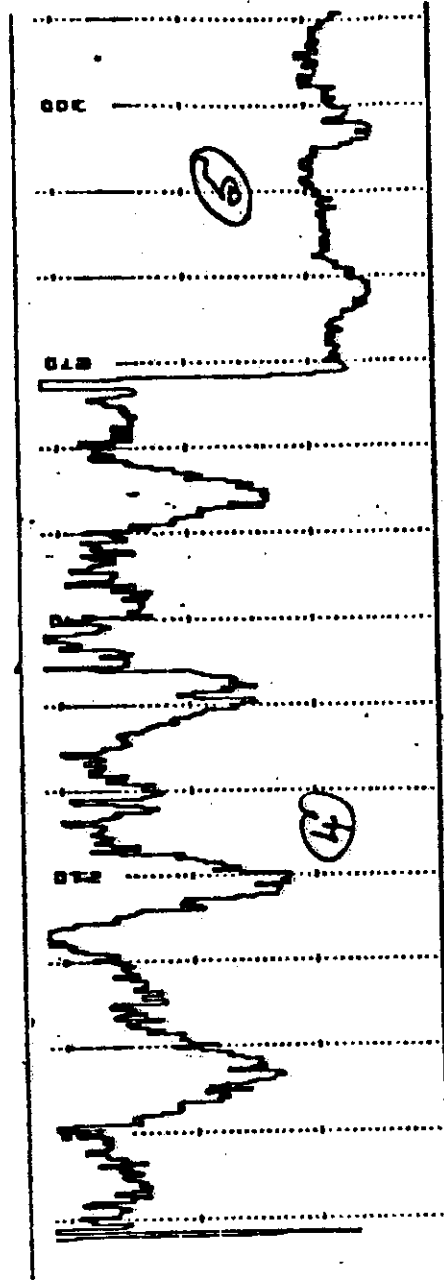
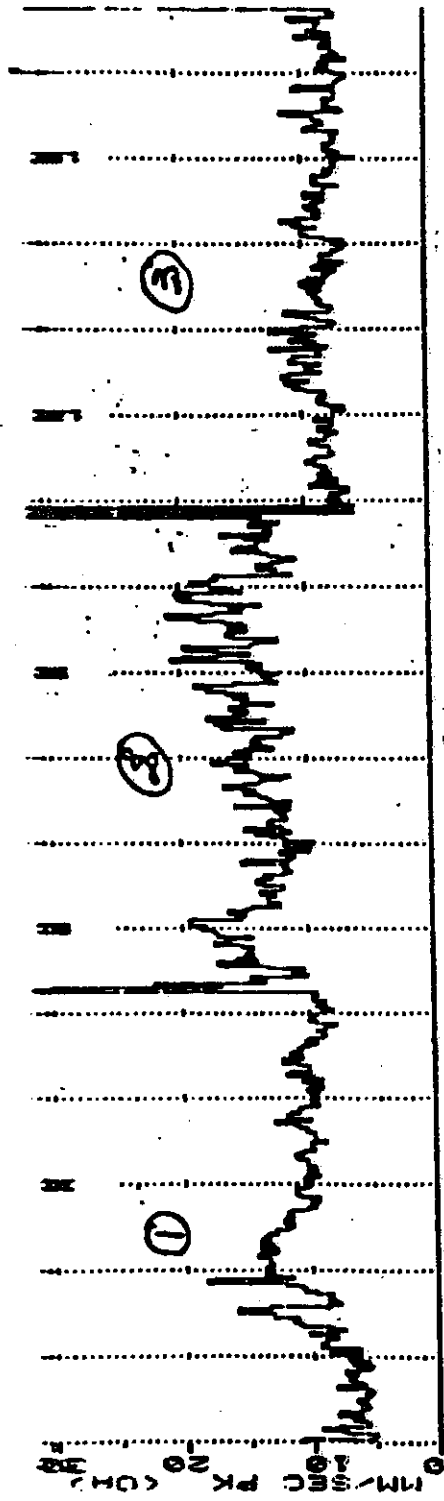


FIG. 6 VIBRATION LEVELS PRESENT ON A TYPICAL BASE WITH ROLL MISSING

The readings shown in Figure 5 were obtained with no wing roll present in the same frame, and the magnitude of the vibrations would be cause for concern. Figure 6 is a comparison of five points on the same frame, clearly indicating the flexible points. At point four, deflection of the frame produced amplitudes of a highly destructive nature. Should the amplitudes produced by this support frame be dampened or contained by the idler roll during operation, then the path of least resistance of these vibrations would be through the bearings.

From a physical point of view, the flexibility of the bases could be detected by watching the conveyor in operation. This could be seen under two conditions:

- (i) The bases flexed in the direction of belt travel when the conveyor was running unloaded.
- (ii) The supporting gusset flexed noticeably when a new load first passed over the base.

IDLER BASE TEST BED RESULTS (Fig. 7 to 11)

In order to ascertain the vibratory condition of an idler roll support frame, with and without additional strengthening being done to the upright and cross-piece, we carried out the following investigation in our works:

The instrument used was the IRD Model 880 Microprocessor Analyser with built-in hard copy printer, as was used in all previous site tests.

A motor and belt drive system was used to drive the wing roll at a speed corresponding with those attained in the field under normal operating conditions. The speed in question was approximately 700 RPM. Under the first test, the roll was of a reasonably balanced condition. The wing upright and cross-piece were strengthened in order to increase the mass and stiffness, resulting in a damping of the roll vibration. This test proved highly successful as is indicated by test one.

Data Sheet One of Four indicated 0.8mm/sec, being the amount of vibration on the top of the frame, and all being due to the balance condition; whereas Data Sheet Four of Four is indicative of the actual amount of vibration produced by the roll which was 4.8mm/sec with only 2.5mm/sec representing the amount of unbalance, and the rest being made up of misalignment, mechanical looseness and associated harmonics.

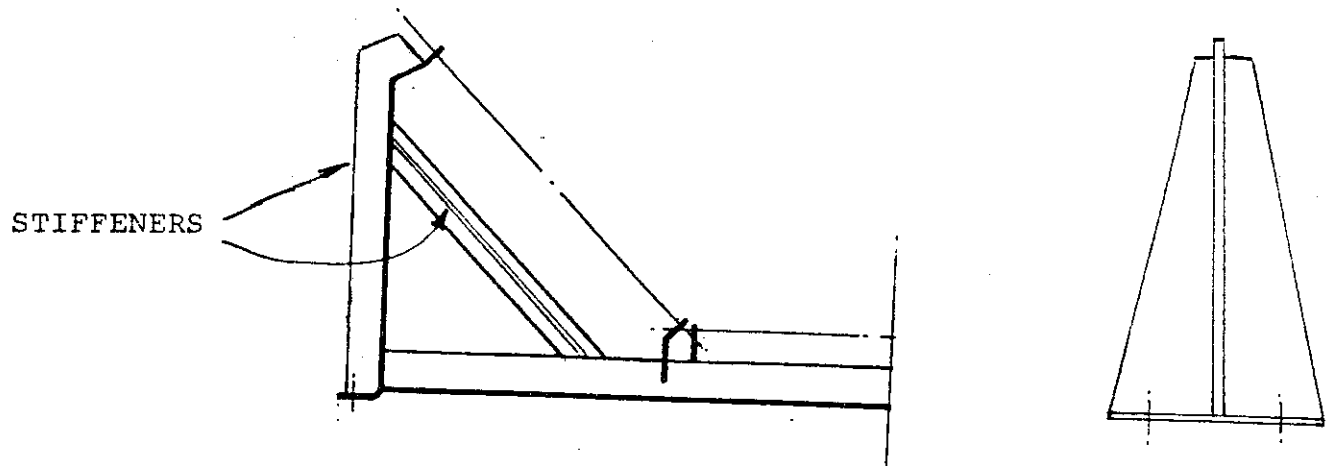
An unbalance was created by welding a piece of metal plate to the roll to create the condition recorded under Test Two. The overall increase was noted on all data sheets with the most significant increase being noted on Data Sheets Two and Three. Data Sheet Two represents the cross-piece which indicates that should any unbalance be present, the amount of vibration created will take the path of least resistance, and as the cross-piece was the most flexible of the entire structure, this was noted as the point with the most vibration. Although the unbalance characteristic was noted to have doubled in severity, the overall increase was seen to be created by the flexibility of the structure.

The unbalance created for this test also saw a drastic increase in the vibration measured on the main upright of the frame, when compared to test one.

For Test Three, the additional strengthening done to the main upright was removed from the base up to the first bend supporting the wing roll, which resulted in a general increase in the vibration levels. In some instances the amplitudes doubled, suggesting that if at any time during operation an imbalance of the roll becomes sufficient to excite the flexible points of the frame, bearing failure would probably occur.

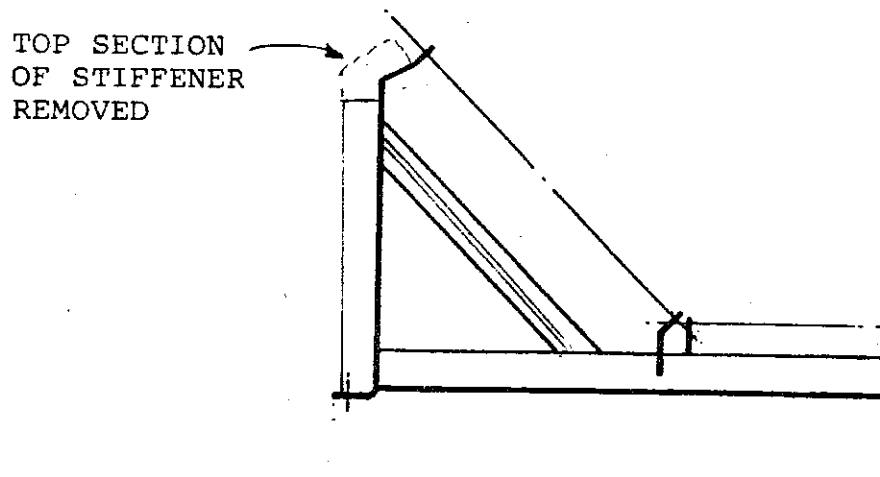
IN ORDER TO ASCERTAIN VIBRATORY CONDITIONS OF AN IDLER FRAME WITH VARIOUS STIFFENERS PROVIDED IN THE FLEXIBLE AREAS INDICATED ON SITE TEST WERE CARRIED OUT IN OUR WORKS

AN IRD 880 MICROPROCESSOR ANALYSER WITH BUILT IN HARD COPY PRINTER WAS USED



STIFFENED BASE

(FOR TEST 1 2 & 4)



(FOR TEST 3)

FIG. 7

MACHINEKİ VIBRASYON SİGNAİLERİ

MACHINE LOCATION.	CORLEQUIP TEST BED, - SPAN ON.	DATA SHEET NO.	THREE OF FOUR.	BY.	ALANIE TAYLOR.
MACHINE.	IDLER ROLL SUPPORT FRAME.	TYPE PICKUP.	MODEL 970 REC.	MODEL	880
OPERATING CONDITIONS.	MOTOR DRIVEN USING ROLL UNDER	AMPLITUDE UNITS.	mm/sec.	VIBRATION ANALYSER.	
VARIOUS SUPPORT CONDITIONS. (VIBRATION IN BASE DUE TO UNBALANCED ROLL)		MEASURE POSITION.	SEE DIAGRAM.	DATE.	26 OCTOBER 84.

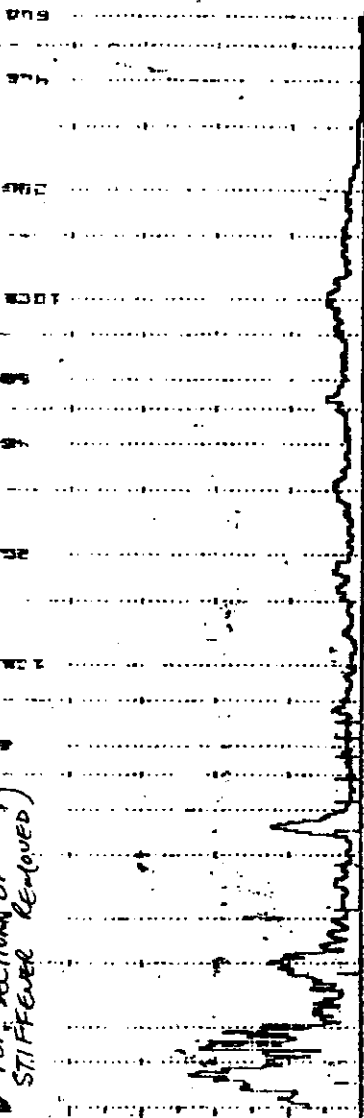
TOP SECTION OF
STIFFENER REMOVED

46-557
SPK 10464

2010

DATE

DISPATCH



390 RPM

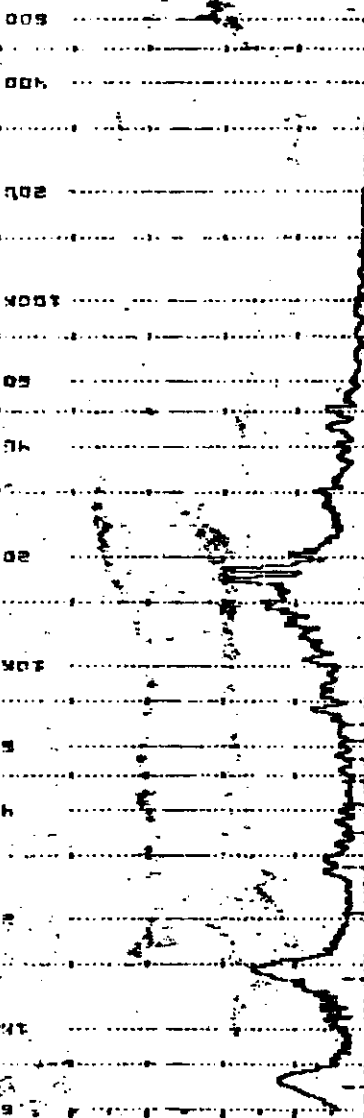
ON 8.6(7-9.5)
M/SEC FX 10%BM
5

SECOND / 1st / 100. 350 RPM

1104 1000

DATE _____
MACH _____

AND MECHANICAL ANALYSIS



17(13-28)
SEC PK 102BM
10
20
30

END/REV. 11/2005.

1925

SECRET

ANALYSIS OF MECHANISMS

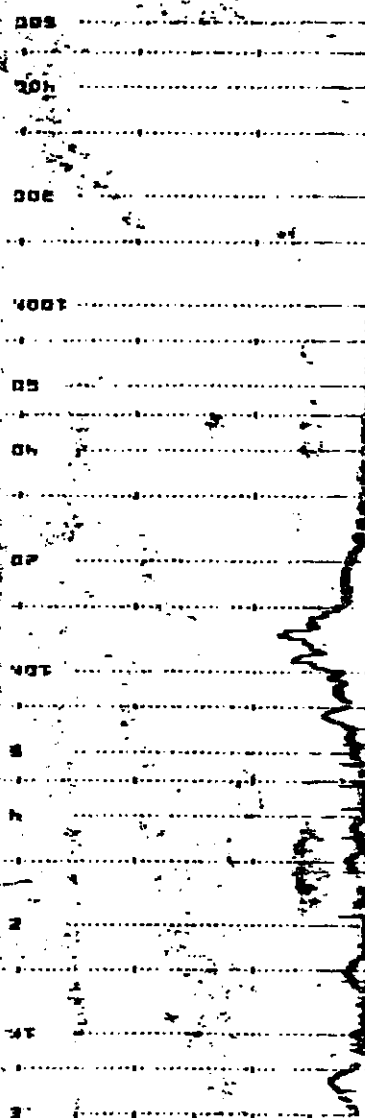


FIG. 11

MACHINERY VIBRATION SIGNATURE

MACHINE LOCATION: <i>COALEQUIP TEST BED - BRADPH.</i>	DATA SHEET NO.: <i>ONE OF FOUR.</i>	BY: <i>NAME TAPPA.</i>
MACHINE: <i>IDLER ROLL SUPPORT FRAME.</i>	TYPE PICKUP: <i>MODEL 970 REC.</i>	MODEL: <i>880</i>
OPERATING CONDITIONS: <i>MOTOR DRIVEN WING ROLL UNDER</i>	AMPLITUDE UNITS: <i>MM/SEC.</i>	VIBRATION ANALYZER:
<i>VARIOUS SUPPORT CONDITIONS. (VIBRATION ON FRAME DUE TO STD. ROLL.)</i>	MEASURE POSITION: <i>SEE DIAGRAM.</i>	DATE: <i>26 OCTOBER 84.</i>

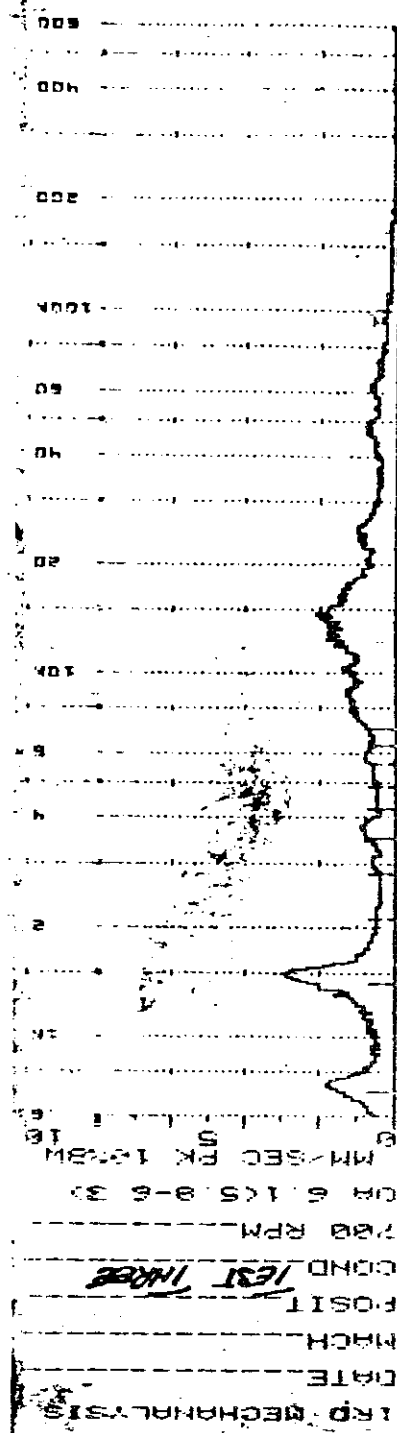
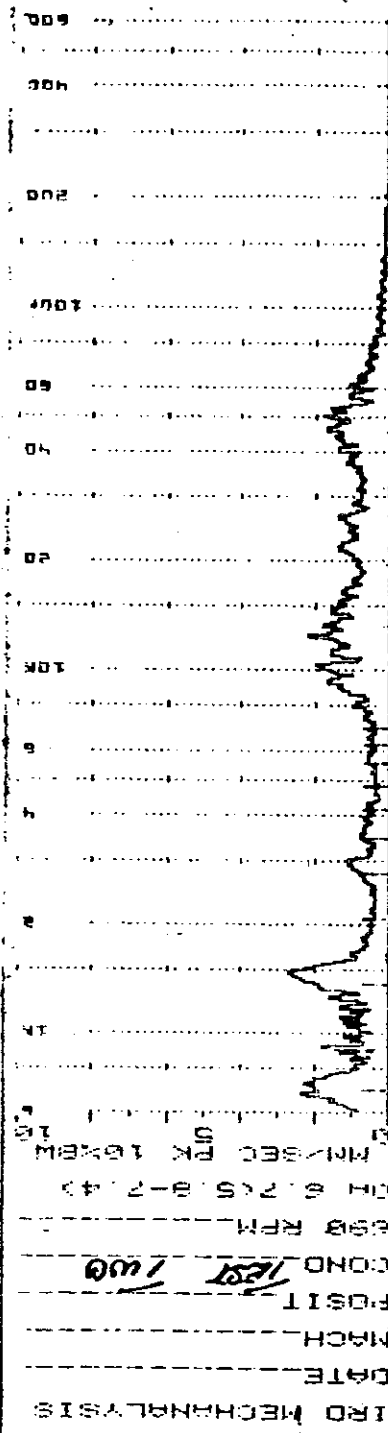
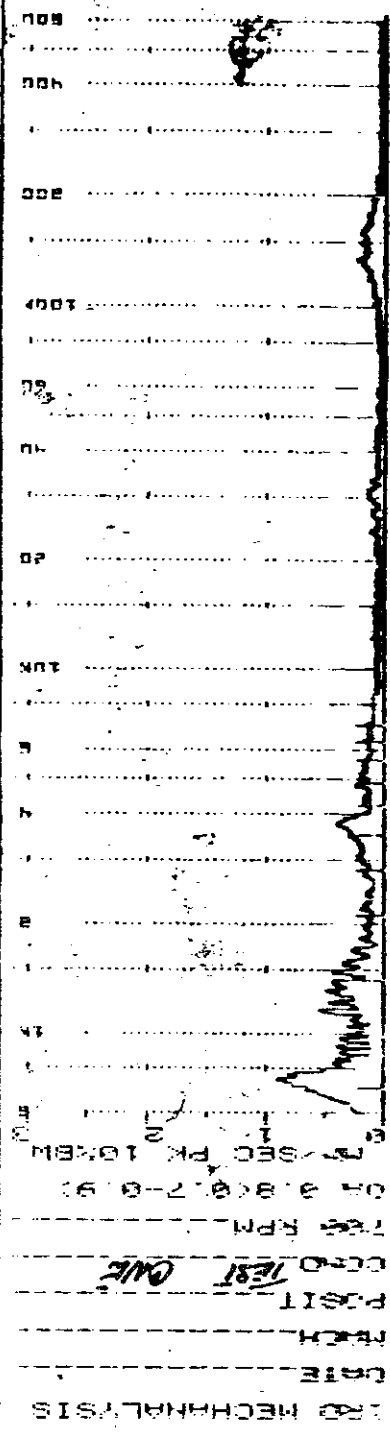


FIG. 8

The conclusion drawn from the tests conducted, is that, as the rolls are not prebalanced before installation takes place, any unbalance forces being generated by the roll are likely to excite the vibratory condition of the structure, and as the bearing is the most sensitive part of the entire installation, irreparable damage would be the only result. This was further substantiated by the fact that if the structures are too flexible, and are not able to contain or dampen any vibrations which are likely to excite their own resonant or natural frequencies, this would result in more forces being placed on the bearings.

(d) T.I.R. and Unbalance of Rolls

(The major causes of excessive T.I.R. and unbalance are shown in fig. 12)

Although the excessive vibration on a conveyor base and support structure is due to their being too flexible, the vibrations must have a source, which in most cases is the idler rolls, especially when they are rotating in the 700 rpm range and with T.I.R., inside SABS specification and a 6mm tube wall but in excess of 0,5 mm.

Two rolls which had a T.I.R. in excess of 1mm but were within SABS spec. of 1,86 mm were placed on a soft bearing balancing machine to determine the amount of unbalance on the rolls. This was found to be 7 kg and 16 kg respectively (at 700 rpm). This indicates that it is necessary to keep the T.I.R. to less than that specified in SABS, and to ensure that variation in tube wall thickness, out-of-balance, and bearing concentricity are kept to a minimum for high speed conveyors.

6.0 A WORKED EXAMPLE GIVING THE COMBINED EFFECTS OF A ROLL WITH MAX T.I.R.

Taking a 152mm Dia idler roll of 6mm wall thickness with a 3-roll 45° configuration at 6m/sec on a 2,2m wide belt of 50 kg/m mass we have:-

MACHINERY VIBRATION SIGNATURE

MACHINE LOCATION. <i>CORLEQUIP TEST BED - BRADON.</i>	DATA SHEET NO. <i>FOUR OF FOUR.</i>	BY <i>ALANIE TAYLOR.</i>
MACHINE. <i>IDLER ROLL SUPPORT FRAME.</i>	TYPE PICKUP. <i>Model 970 REC.</i>	MODEL <i>970</i>
OPERATING CONDITIONS. <i>MOTOR DRIVEN WING ROLL UNDER</i>	AMPLITUDE UNITS. <i>mm/sec.</i>	VIBRATION ANALYSER.
<i>Various Support Conditions. (Vibration of 30 cm)</i>	MEASURE POSITION. <i>See Diagram.</i>	DATE. <i>26 October 84.</i>

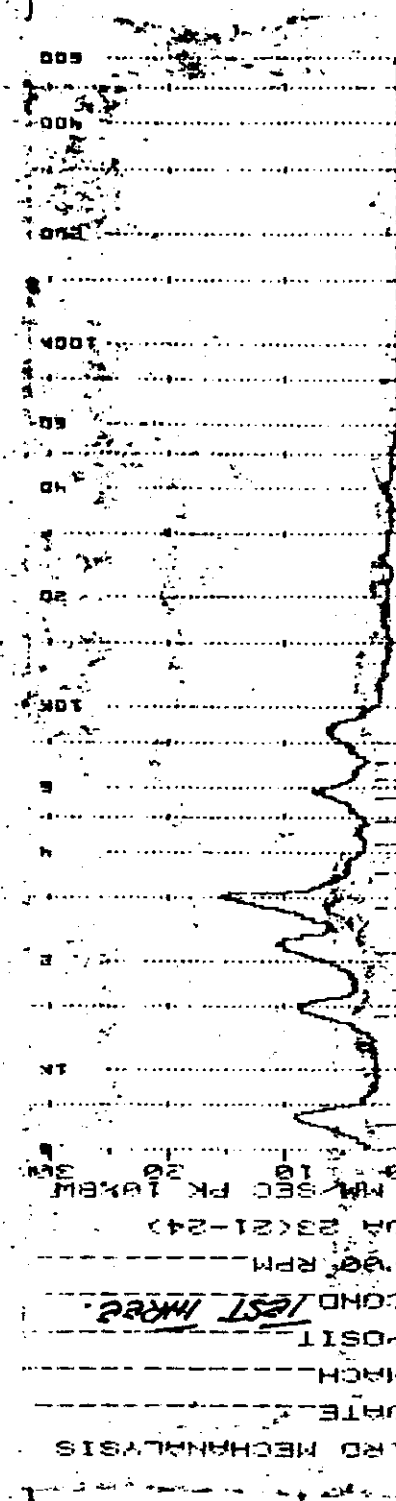
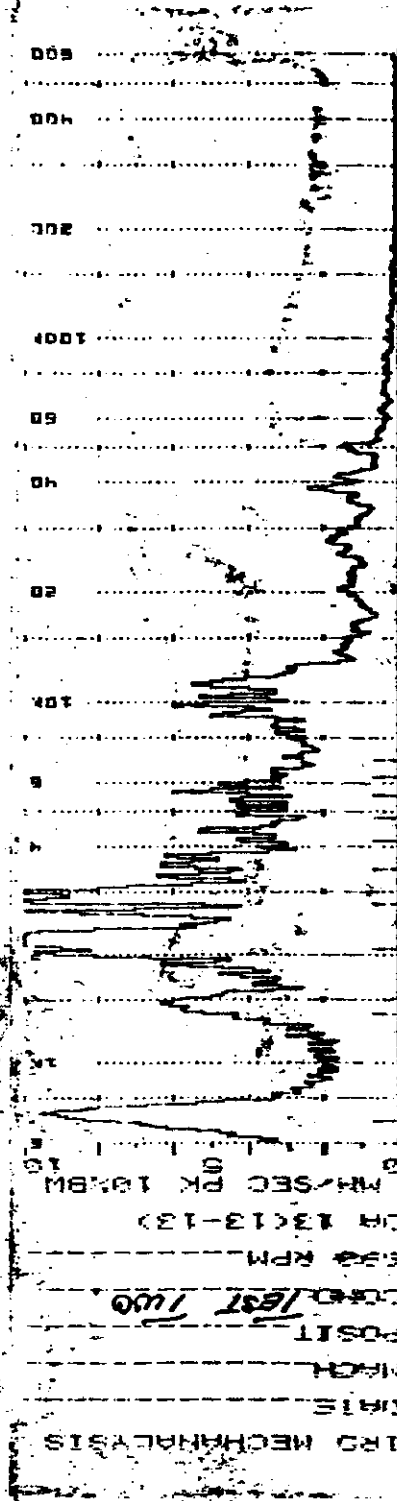
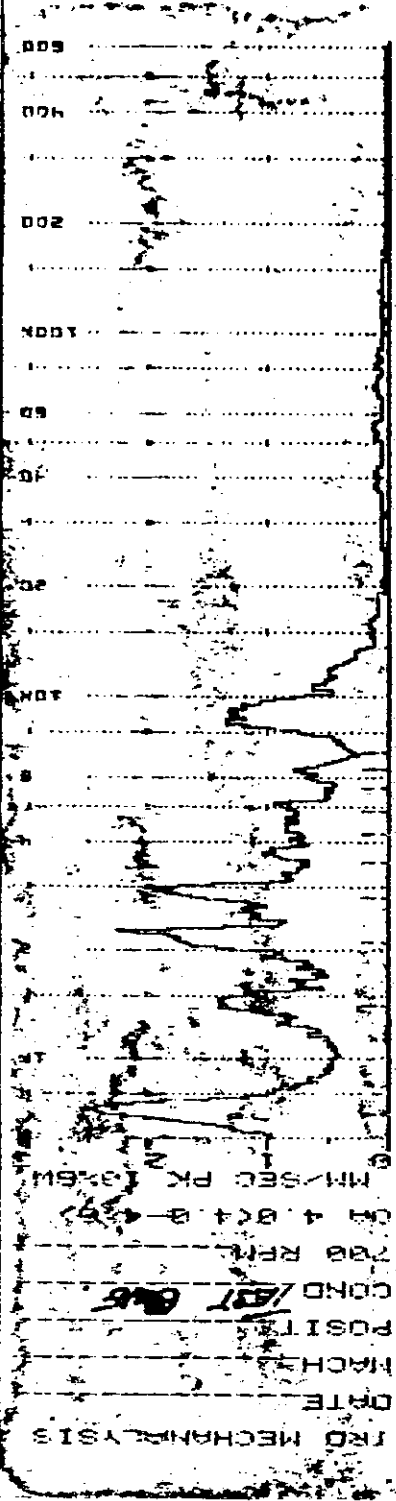
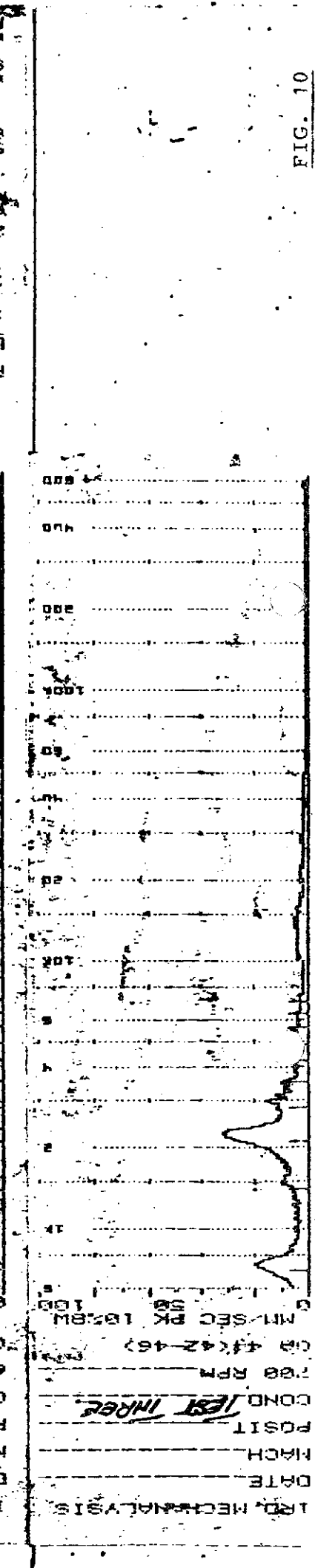
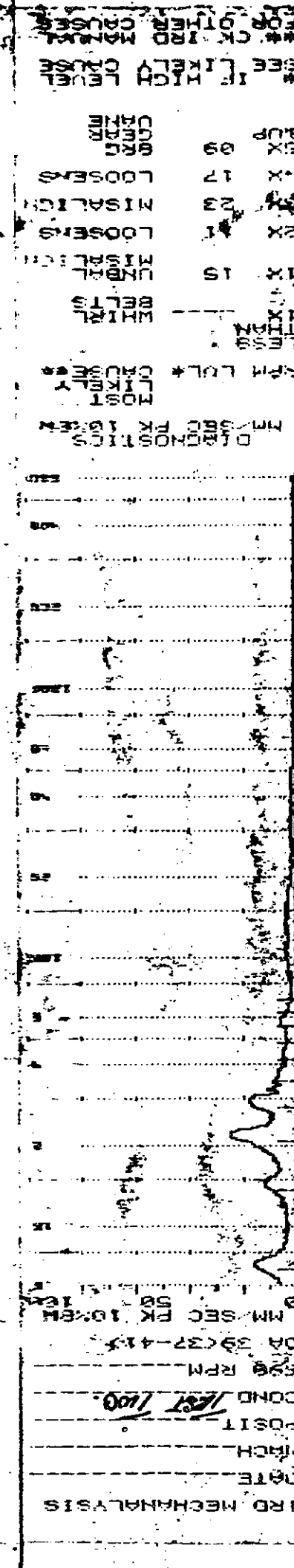
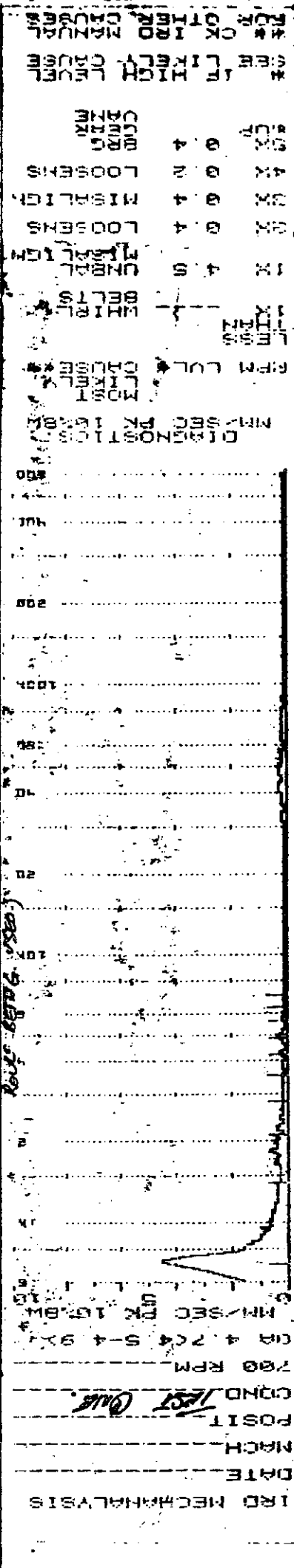


FIG. 9

MACHINERY VIBRATION SIGNATURE

MACHINE LOCATION. <u>CORNAQUIP TEST BED - BRACON.</u>	DATA SHEET NO. <u>TWO OF FOUR.</u>	BY <u>WAVIE TAYLOR.</u>
MACHINE. <u>TOWER ROLL SUPPORT FRAME.</u>	TYPE MICH. <u>WAGON 940 APC.</u>	MODEL <u>800</u>
OPERATING CONDITIONS. <u>MOTOR DRIVEN WING ROLL UNDER</u>	AMPLITUDE UNITS. <u>MM/SEC.</u>	VIBRATION ANALYSIS.
<u>VARIOUS SUPPORT CONDITIONS. (VIBRATION IN CROSS THE ROLL BEING USED.)</u>	MEASURE POSITION. <u>SEE DIAGRAM.</u>	DATE. <u>26 October 84.</u>



- (a) Roll speed = 753,9 RPM
- Max T.I.R. to SABS = 1,86mm
- (b) Eccentric effect (Inertia Force) = 194,82 N
- (c) Balance force due to (TIR) upto = $16 \times 9,81 = 156,9 \text{ N}$

This combined effect could cause the following reactions.

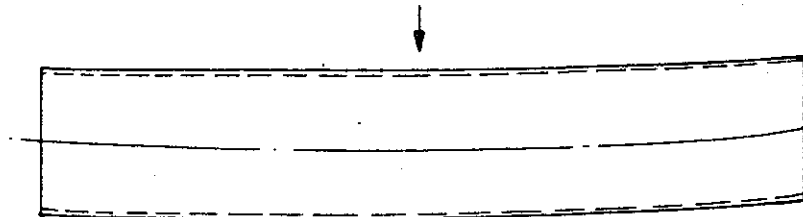
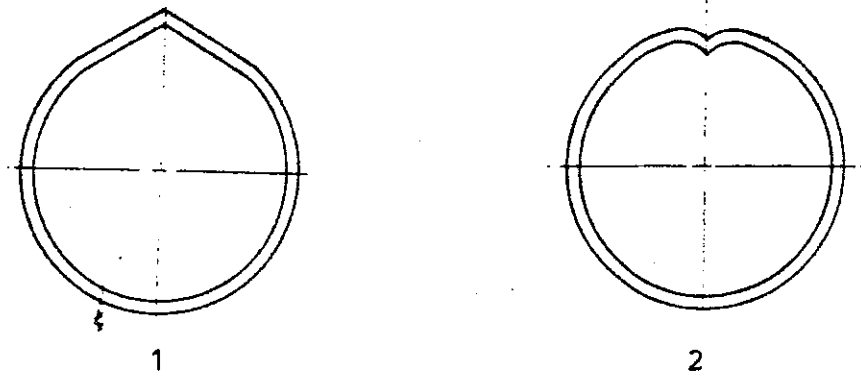
- (a) When a number of rolls of similar T.I.R. are installed in consecutive bases the eccentricity of the roll could in fact lift the belt off the roll allowing the roll to vibrate in the base.
If the base and supporting structure is not of sufficient strength and or rigidity, the vibration is carried through the entire conveyor system. In some instances this vibration comes into resonance with the belt, causing very high stress.

7.0 CONCLUSIONS

- (a) In order to manufacture an idler for high speed conveyors, it is important to ensure that the idler roll is manufactured to tolerances tighter than those given in SABS 1313. The T.I.R. for high speed idlers should be limited to 0,5mm. This will in turn reduce the out-of-balance force to acceptable limits.

It is not satisfactory to machine the O.D. of the roll after manufacture to correct the T.I.R., as this reduces the wall thickness and does nothing to reduce the out-of-balance effect.

- (b) A problem experienced by South African idler manufacturers is that the tubing available for idler roll manufacture is produced to S.A.B.S. 657 Part III, which limits the tube ovality for a 152mm dia tube to 0,6mm which on long rolls is already outside the 0,5mm limitation.
- (c) There are various manufacturing methods used. Some of these are more suitable for high speed idler roll manufacture than others.



3. Straightness of 1 in 1000 ovality of 0.6mm max. (To SABS 657 PT III)

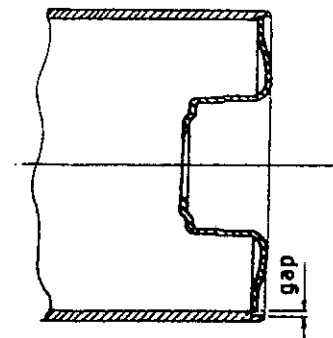
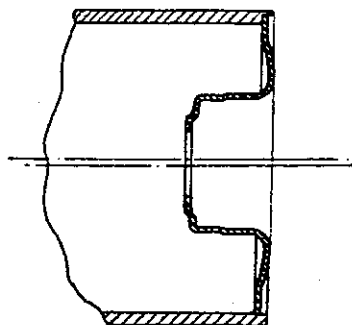
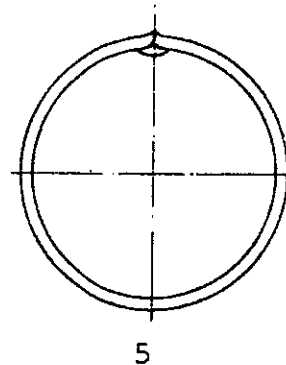
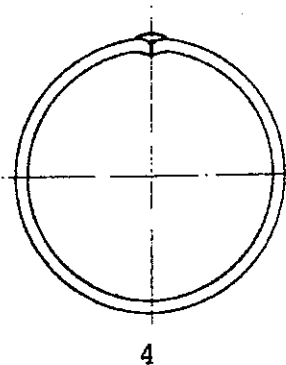


FIG. 12

(d) The L-10 life of an idler roll is in general requested to be 100 000 hrs. It is generally accepted that a calculated L-10 life of 100 000 hrs is the bearing life running under laboratory conditions. In the case of conveyor idlers the rolls will probably fail due to ingress of dirt or tube failure. Therefore a life of 35 000 hrs is more than likely to be in keeping with actual practice.

8.0 ACKNOWLEDGEMENTS

The author wishes to acknowledge the permission of Mr. R.W. Taylor of Richards Bay Coal Terminal Company Ltd to quote from his paper "Practical & Operating Aspects of the Richards Bay Coal Terminal (case study)".

Boart Research Centre (Dr. D.W. Paul)
Vibration Consultant Pty. Ltd. (Wayne P. Taylor)

The help and assistance in the checking of mathematical examples - by Bruce Watson BSC (Hons.) Wits University.

The help and assistance in the preperation of the paper of various Osborn MMD Employees, most particularly Mr. Luqman Ebrahim and Mrs. Deline Strydom.

9.0 REFERENCES

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