DYNAMIC ANALYSIS OF CONVEYOR BELTS

AT ISCOR'S MINING DIVISION

SYNOPSIS

In recent years ISCOR's Mining Division has experienced dynamic problems with long and/or high inertia conveyor belts.

Dynamic analysis of conveyor belts has been under study at the department of Mechanical and Electrical Engineering, Mining since 1989. The Finite Element Method is used to perform modal analysis and natural frequencies of the system are calculated and used to predict the dynamic response of the system, as a superposition of normal modes. Results obtained compare favourably with measurements on site and in the literature. ISCOR's method had successfully described dynamic problems not explained by traditional dynamic analysis techniques.

Modal analysis not only represents a novel approach to the dynamic analysis of conveyor belts, but also allows the engineer to understand the behaviour of the system better.

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1.0 INTRODUCTION

1.1 Iscor Mining Division

ISCOR's Mining Division operates 9 mines in South Africa and Namibia. Materials produced comprise (1); iron ore (22,5 million tons per year), coking coal (3,5 million tons per year), power station coal (9,7 million tons per year), dolomite (1.5 million tons per year), zinc and other products.

1.2 Conveyor Belts at ISCOR

ISCOR operates more than 500 conveyor belts in its different mines and works. Particularly in the Mining Division, long and high-capacity conveyors are common, therefore the analysis and evaluation of dynamic phenomena in conveyor belts is often required.

2.0 DYNAMIC PHENOMENA ON CONVEYOR BELTS

2.1 Longitudinal transient stress

Longitudinal dynamic forces appear during the starting and stopping of a conveyor belt. Under given circumstances the transient stress generated by these forces can damage or even destroy the system. These forces are associated with the formation of travelling elastic waves (2) caused by the acceleration or deceleration of the belt. Although not frequently mentioned in the literature, the formation of standing waves (3) during steady state operation (as a resonant phenomena) is also possible.

Although huge safety factors (between 5 and 10) are used on static analysis, sometimes these factors are not enough to avoid dynamically caused problems and failures.

2.2 Transverse Vibrations

The failure of components such as idlers in conveyor belts may be largely attributed to dynamic loads caused by transverse belt vibration during steady state operation (4).

Transverse vibrations can also appear due to transient tension drops when the conveyor belt is stopped. This can produce severe spillage problems and, in extreme cases, damage to the belt (5.6).

3.0 CURRENT METHODS FOR DYNAMIC ANALYSIS OF CONVEYOR BELTS

3.1 Longitudinal Forces

A method of calculating the propagation speed of stress waves in conveyor belts was proposed by Harrison (6). This method is based on prismatic bar physics. Other methods are based on Finite Element Analysis (3,7,8,9), wave propagation (10), or combined systems (11). These methods are available as a consultory service offered at up to 2% of the value of a given installation.

3.2 Transverse Vibrations

Plate flexural behaviour is used to calculate transverse vibrations on conveyor belts. The only method published in the literature is the one proposed by Dr Harrison.

4.0 DESIGN OF CONVEYOR BELTS AT ISCOR

4.1 Static Design

ISCOR uses its own code of practice for the design of conventional conveyor belts. This standard is based on a combination of overseas standards, manufacturers' recommendations and ISCOR's own operating experience.

Both mainframe and PC-based programs have been developed for designs based on the ISCOR specifications.

4.2 Dynamic Design

Until recently, dynamic analysis was limited to quasi-static tension analysis and casting calculations. However, the dynamic problems experienced at ISCOR installations in the past few years have called for the development of dynamic analysis methods for long and/or high-capacity conveyors. A novel in-house method for dynamic analysis of conveyor belts based on finite element modal analysis techniques, has been developed with external academic support (12).

The complexity of the dynamic phenomena as well as of the published methods for analysing them makes the use of consulting services that produce a "black box" of results, with a high associated cost, difficult. Furthermore, although computer programs are available locally, the expertise required for their utilisation is based overseas.

5.0 MODAL ANALYSIS OF CONVEYOR BELTS

Finite Element Modal Analysis is used to predict the dynamic response of conveyor belts (longitudinal and transverse behaviour). The aim of the work is to obtain approximate quantitive values of dynamic forces, using available information. This method has been developed by the author as part of an M.Sc Thesis (12).

5.1 Longitudinal Vibrations

Although, traditionally, longitudinal vibrations on conveyor belts have been treated as elastic waves (an approach easily understood by electrical engineers), transient phenomena on starting and stopping can be properly represented using a linear vibration model. The value of a linear model has been extensively proven by Drs Funke and Harrison's work, while the vibration approach is only a modification of the wave approach. Both can be derived from basic theory of elasticity.

The use of a "vibration" model allows the design and maintenance engineers (usually mechanical engineers) to speak a common language (using familiar words like resonance and damping). Transient problems can be avoided if the excitation of the vibration modes of the belt is prevented.

For the numerical evaluation of belt forces and elongations, finite element models are used (see fig. 1). These models are based on easily available information like belt mass and elasticity, while parameters like damping are determined empirically. Analysis is done by modal superposition techniques. Approximate values are obtained (± 20%). This information allows the reasonably accurate design of support structures, take-up systems and also a realistic evaluation of belt safety factors.

5.2 Transverse Vibrations

Dr Harrison's method is used to evaluate the natural frequency of the belt given the belt tension and idler spacing. The possibility of resonant vibrations is evaluated by means of the amplification factor (fig. 2). Once again, damping is determined empirically.

Site measurements (using "bump" tests) correlated very closely with results obtained using finite element models and Dr. Harrison's formula. As this last method is easier to apply, it was adopted for future calculation.

6.0 APPLICATIONS

6.1 Overland Conveyor 1 (Figure 3a)

During a full-load start-up, the counterweight tower of this conveyor was damaged (Figure 4). Before this accident, lifting of the belt in the concave radius before the drive station was experienced during start-up (up to 3 meters lift according to eye-witnesses, including the author). Unfortunately, the belt failed before measurements could be taken.

Calculations using the modal analysis method, showed that the natural period of the belt coincided with the time delay between the start-up of the motors, which partly explained the significant lift of the belt that was observed. From the calculated tensions, a belt lift between 1,5 and 2m can be expected, which is broadly in accordance with site observations.

The start-up of the fully loaded conveyor was simulated, using the belt characteristics and data supplied by the manufacturer of the fluid coupling to reproduce the conditions prevailing during the accident.

The results showed the expected displacement of the counterweight to be greater than the available travel, which explained the damage to the take-up tower.

Results obtained with the simulation were used to recommend drive and loading modifications. These modifications were successfully implemented.

After the modifications, site measurements where taken and compared with calculated values, in order to asses the accuracy of the model used. The belt speed was recorded by means of a tacho-generator while take-up displacements were recorded on video. The results obtained compare favourably with calculated results (see Figures 5 to 6).

6.2 **Overland Conveyor 2 (Figure 3b)**

This conveyor has been the subject of analysis by different external consultants (4 in total), with two of them presenting formal reports.

One of the problems experienced with this conveyor was the excessive slippage of the drive pulleys during start-up. This was attributed to the slow response of the electric winch. A post-mortem analysis confirmed this theory (Figure 7). The installation of a faster response winch and the change of the slip-ring drive steps solved the problem.

Another problem experienced was the repeated failure of primary drive gearboxes, while significative oscillations of the drive itself and the current drawn by the electric motor were measured.

The oscillation of the drive was attributed by external consultants, to resonance of the drive supporting structure, this theory was supported by the existence of noticeable oscillations on the walkway above the drives. A modal analysis indicated a natural frequency of 1.9 hertz (compared to the 1.5 hertz measured on the drive itself). This frequency implies sideways displacement, explaining the oscillation of the walkway as a consequence of the drive oscillation.

However, the longitudinal displacement natural frequency of the structure is 9.1 hertz, therefore the resonance of the structure in the longitudinal direction is very unlikely to occurr.

Gearbox failure was attributed by external consultants, to gearbox overload. However, strain gauge measurements during start-up showed the maximum gearbox torque to be 70% of the nominal gearbox value, proving the overload assumption to be wrong. A similar value was obtained by computer simulation, (see figure 8). Further research on the gearboxes indicated that the failures were produced by lubrication deficiencies.

Similar oscillations, although of a lesser amplitude, were measured in another conveyor with the same kind of drive arrangement which forms part of the same discard system.

Severe oscillations were experienced on the cantilever section of the head structure. External consultants attributed the problem to structural resonance, but while the structure oscillated with main frequency components at 1.5 and 3.2 hertz, the natural frequency of the structure was calculated (by means of finite element analysis) as 8 hertz.

Modal analysis of the belt showed that the first two natural frequencies of the return strand between the head and the drive were very similar to the main components of the head and drive oscillations.

It was concluded that the drive oscillation excited the belt on its two first fundamental frequencies, so that the torque variation was amplified and transmitted to the head in form of a belt tension variation that produced the oscillation of the structure. The tension variation, resulting from the measured torque variation in the drive was used as an input in a finite element simulation of the head structure response. The calculated displacement is similar to measured values before and after structural modifications (See Figures 9 and 10).

As confidence in the computer simulation was obtained by means of the good correlation between measured and calculated results, a new simulation was done in order to determine the required back-stop for full load conditions. The worst condition was considered to be an aborted start-up just after tension has built up. The maximum run-back tension is produced when the head reaches zero speed. The calculated run-back force is roughly 3 times larger than the static run back force, but only one third of the rating recommended by the manufacturer. In other words, a static calculation could have led to an underdesigned unit, while the manufacturer's "rule of thumb" rating would have led to overdimensioning.

After two years of work by ISCOR personnel and external consultants, the problems experienced on this conveyor have been successfully analysed and solved.

7.0 <u>CONCLUSIONS</u>

7.1 Conveyor Belt Design Standards

Numerous codes and methods for the static design of conveyor belts exist and they are widely documented and relatively well understood by the designers and users of conveyor belts. In other words, although different methods are used for static design, they are in general equivalent to each other. Discussion and/or comparison of different designs is commonly done without major difficulties, engineering criteria being the factor that really makes the difference.

On the other hand, although abundant literature is available, information on dynamic design of conveyor belts is not explicit. The dynamic analysis itself is treated as a matter of "know how" (in the words of Dr Funke).

Furthermore, the average designer and user have very little understanding of the dynamic phenomena. The stress wave theory is regarded as to "theoretical to be used in practice" (in the words of fellow conveyor designer). These factors lead to the fact that the different dynamic design criteria are difficult to compare and discuss. As a consequence, dynamic design and trouble shooting often become speculative rather than accurate and exact.

Unfortunately, an understanding of the dynamic phenomena is indispensable when designing and operating long and/or high-capacity conveyors. Therefore it is the duty of consultants, manufacturers and users as a first step to the use of common criteria for both static and dynamic design, to improve their understanding of the dynamic behaviour of conveyor belts.

7.2 Modal Analysis of Conveyor Belts

Finite element modal analysis of conveyor belts, as presented in this paper, represents a viable method of predicting the dynamic behaviour of conveyor belts. Calculated values compare favourably with site measurements within the accuracy levels used in normal engineering practice.

Modal analysis is not only more economical in a computational sense, than other methods currently used (like second-degree direct integration), but the results are also easy to evaluate. The required variables at different points of the belt at different times can be calculated, allowing the evaluation of complex situations.

However, it must be stressed that any numerical method is only a calculation tool. The paramount factor for a successful dynamic analysis is a deep understanding of the behaviour of conveyor belts.

8.0 ACKNOWLEDGEMENTS

The author wishes to express his sincere gratitude to ISCOR's mining division and Research and Development personnel for their valuable support in the preparation of this paper.

Gratitude is also expressed to SWF Bulk Materials Handling (Pty) Ltd for allowing the author to present this paper.

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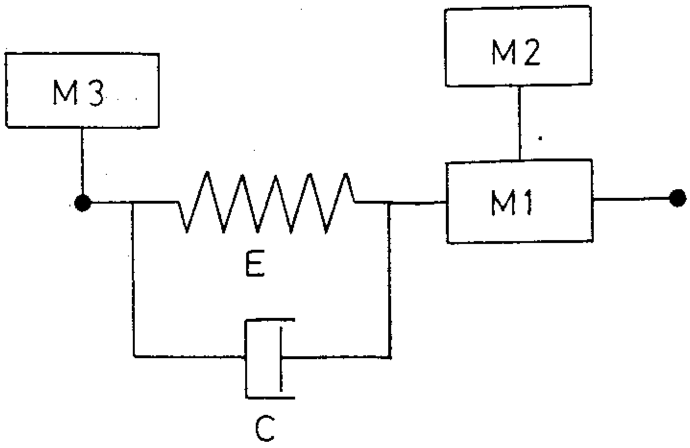


M1 = BELT MASS
M2 = LOAD MASS
M3 = PULLEY / IDLER MASS
E = ELASTIC MODULES
C = VISCOUS DAMPING

Fig.1
FINITE ELEMENT MODEL USED FOR CONVEYOR BELT DYNAMIC SIMULATION

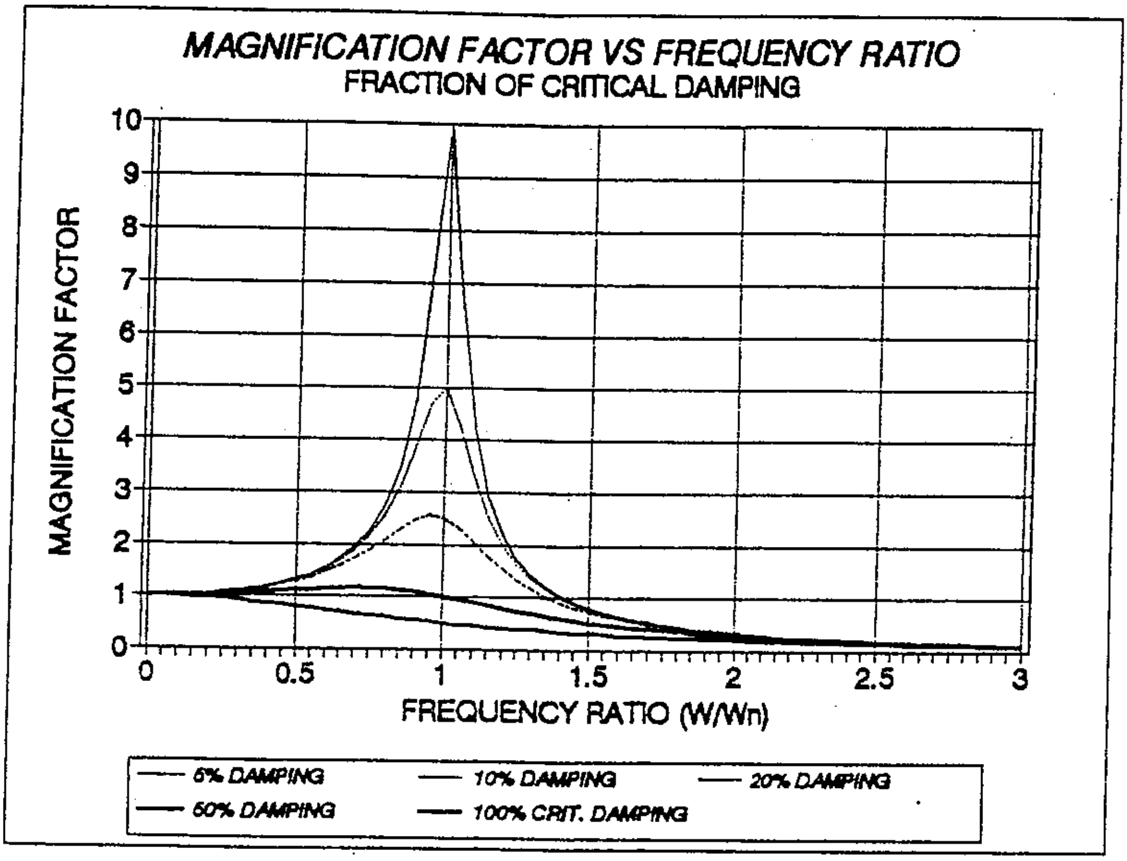
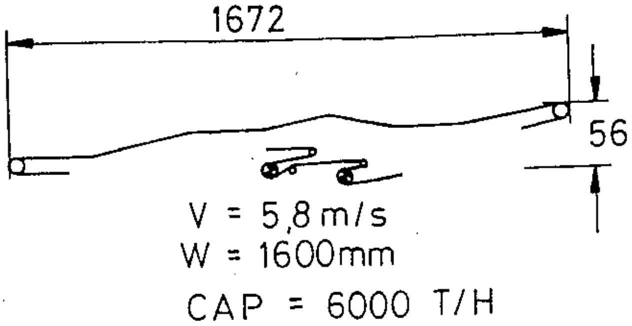
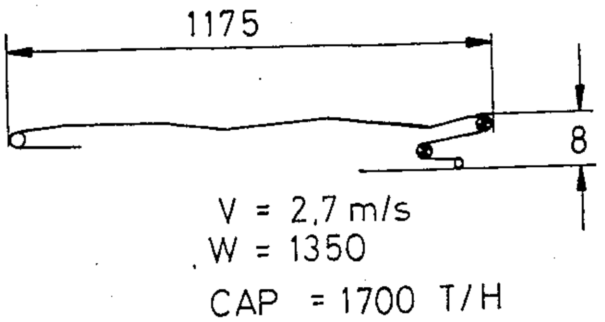


Fig. 2
AMPLIFICATION FACTOR USED FOR RESONANCE CALCULATIONS



(B) CONVEYOR N° 2



(A) CONVEYOR N° 1

Fig. 3
OVERLAND CONVEYORS USED FOR ANALYSIS

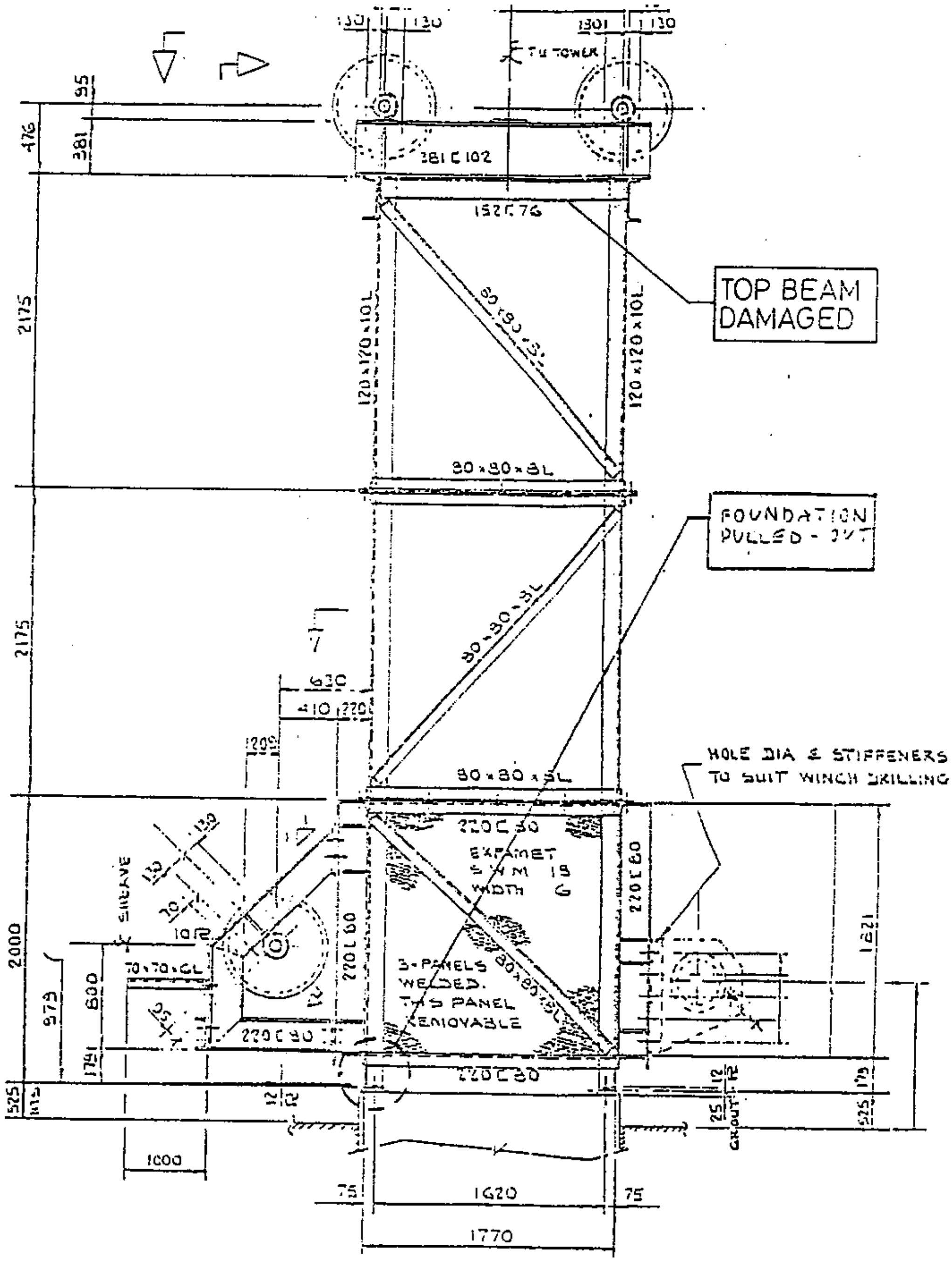


Fig. 4

DAMAGE ON CONVEYOR Nº 1 TAKE-UP TOWER

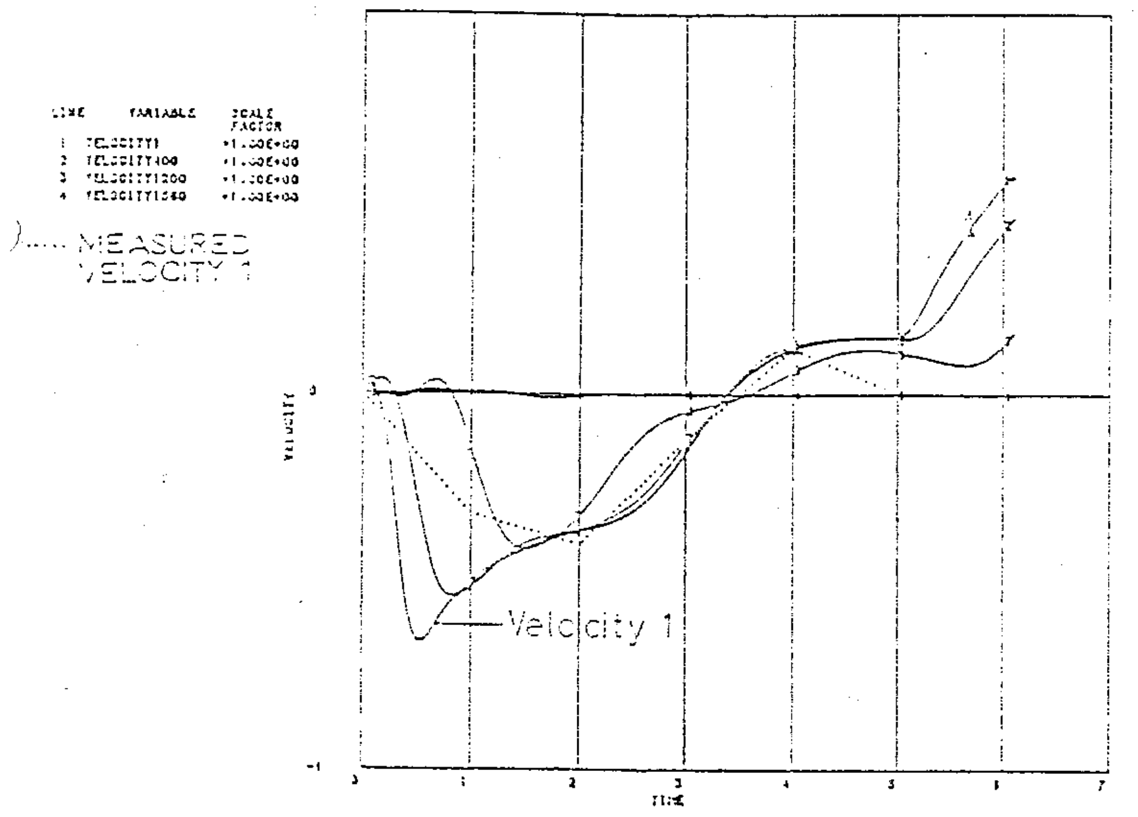


Fig. 5
MEASURED VS CALCULATED SPEED DURING
CONVEYOR N° 1 START - UP

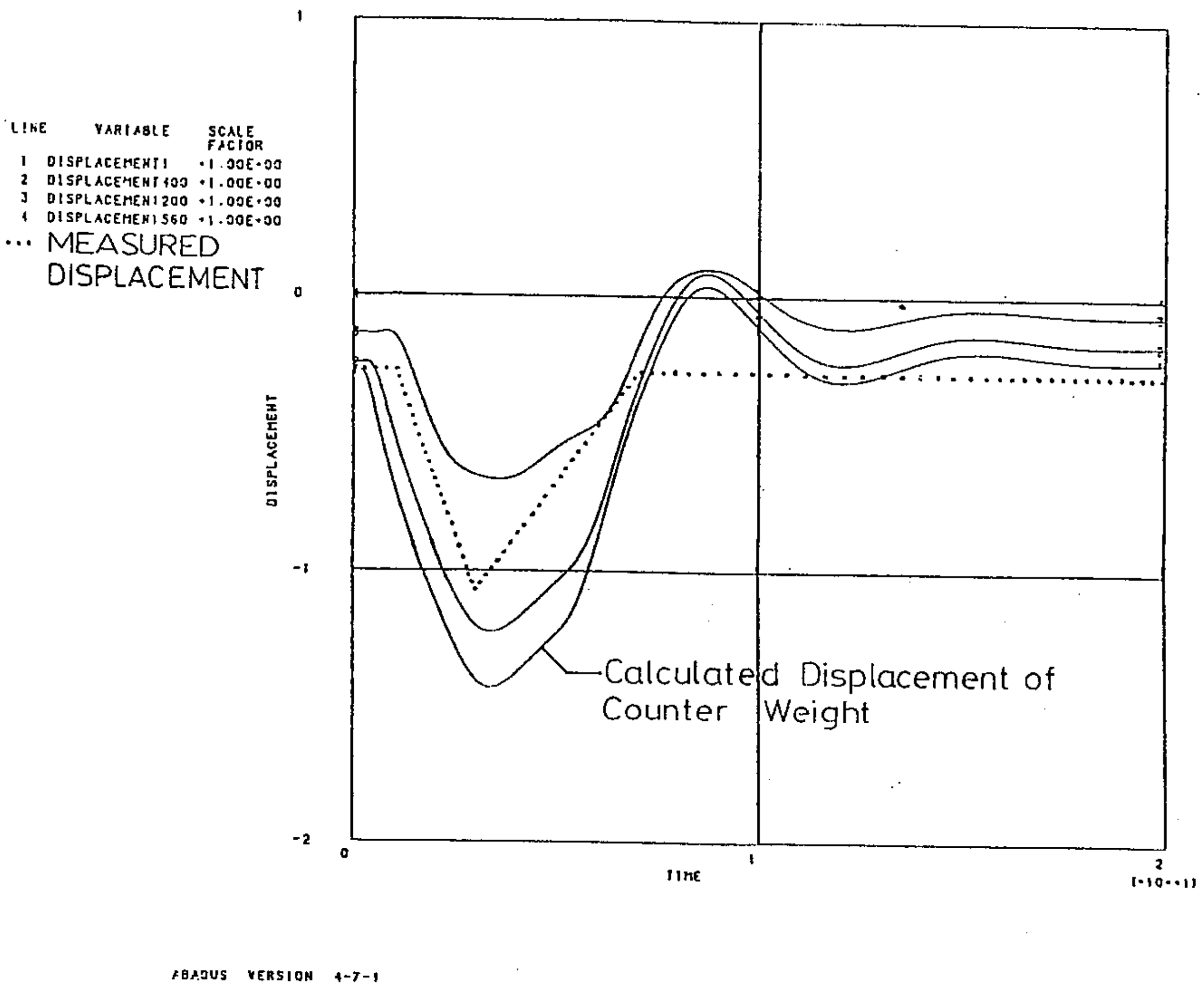


Fig. 6
TAKE - UP DISPLACEMENT (CONV.1)

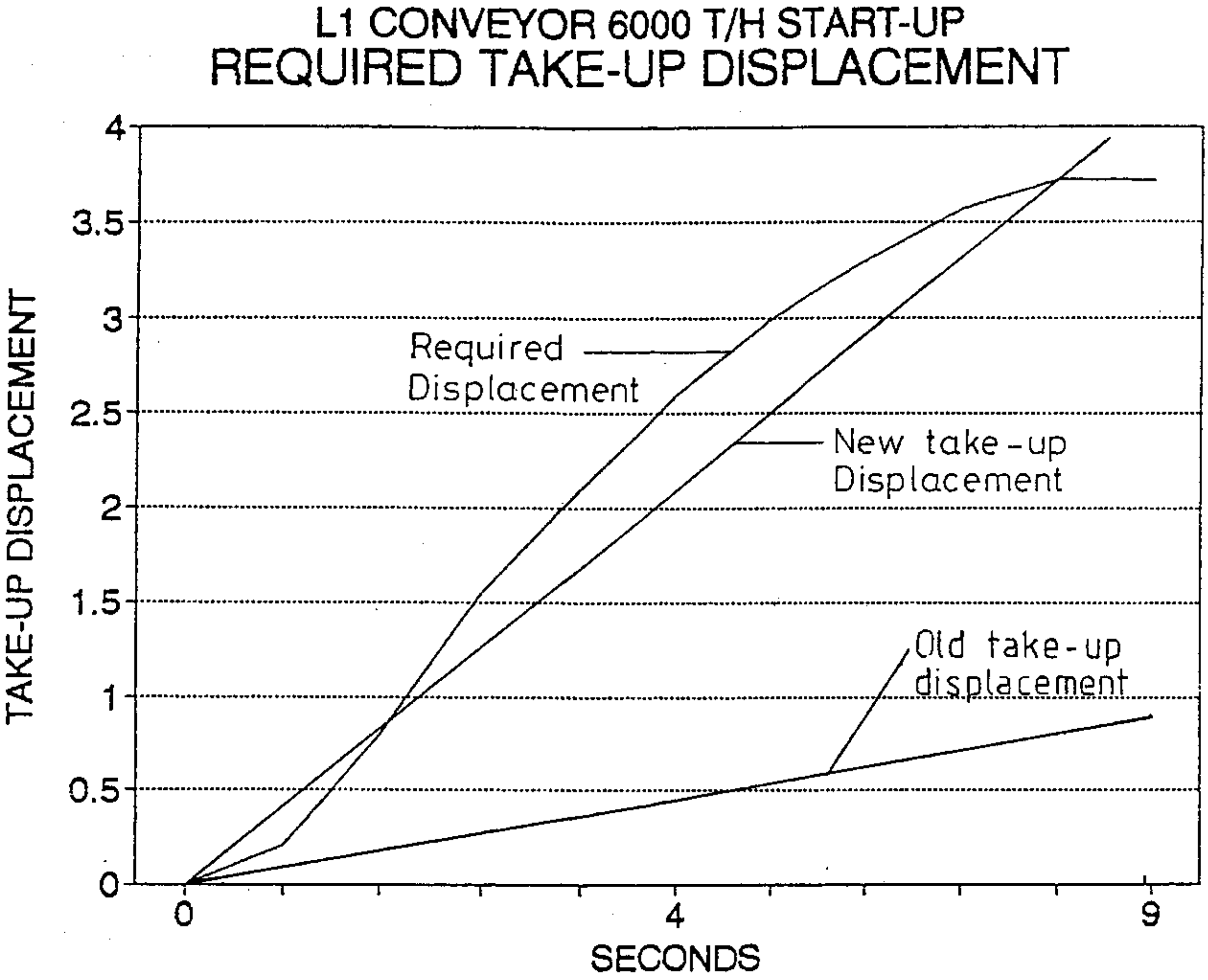
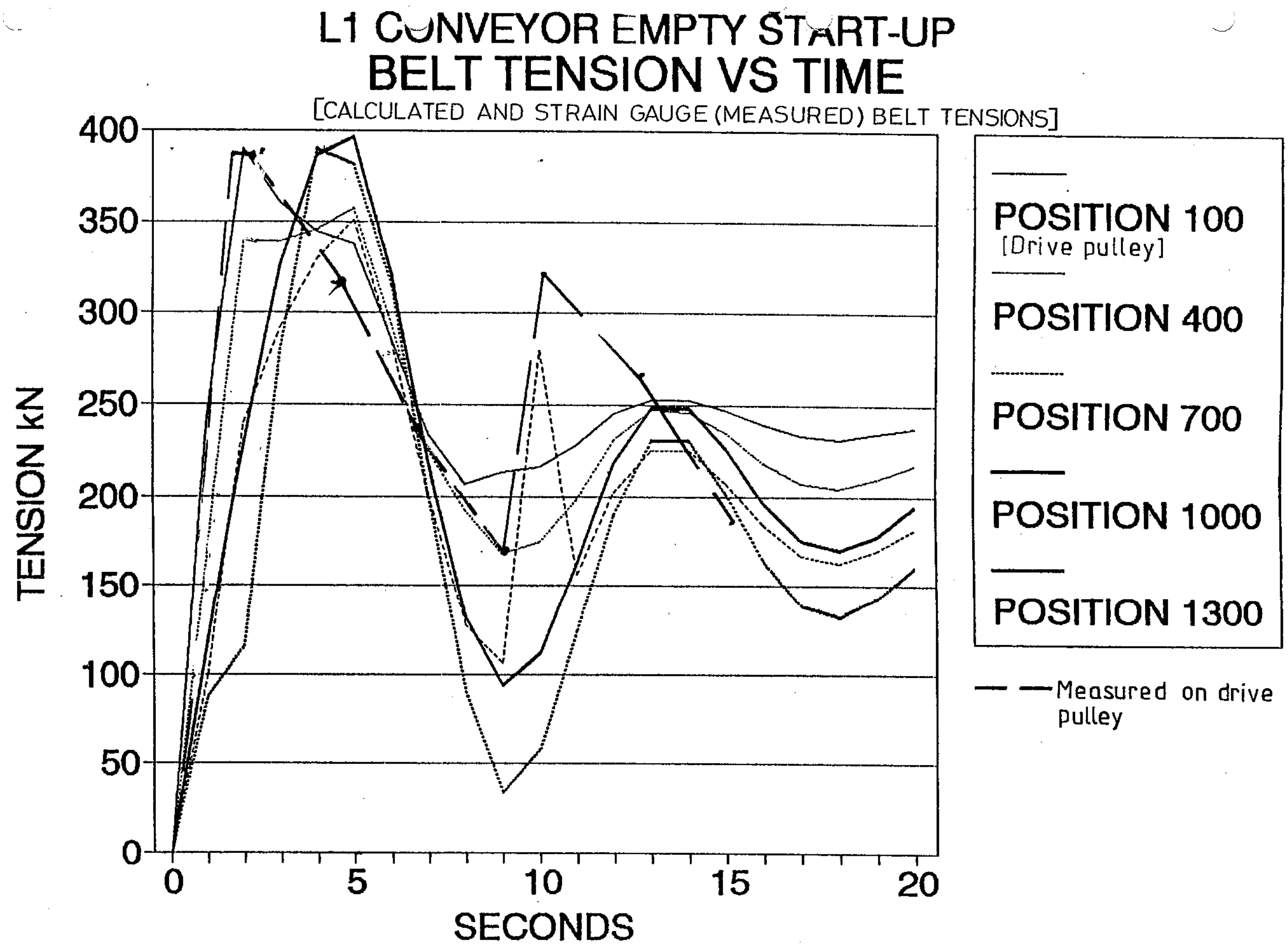

L1 CONVEYOR 6000 T/H START-UP
REQUIRED TAKE-UP DISPLACEMENT
TAKE-UP DISPLACEMENT
SECONDS
Required Displacement
New take-up Displacement
Old take-up displacement
0
0.5
1
1.5
2
2.5
3
3.5
4
0
4
9

Fig. 7



L1 CONVEYOR EMPTY START-UP
BELT TENSION VS TIME
[CALCULATED AND STRAIN GAUGE (MEASURED) BELT TENSIONS]
TENSION kN
400
350
300
250
200
150
100
50
0
0
5
10
15
20
SECONDS
POSITION 100
[Drive pulley]
POSITION 400
POSITION 700
POSITION 1000
POSITION 1300
Measured on drive pulley

Fig. 8

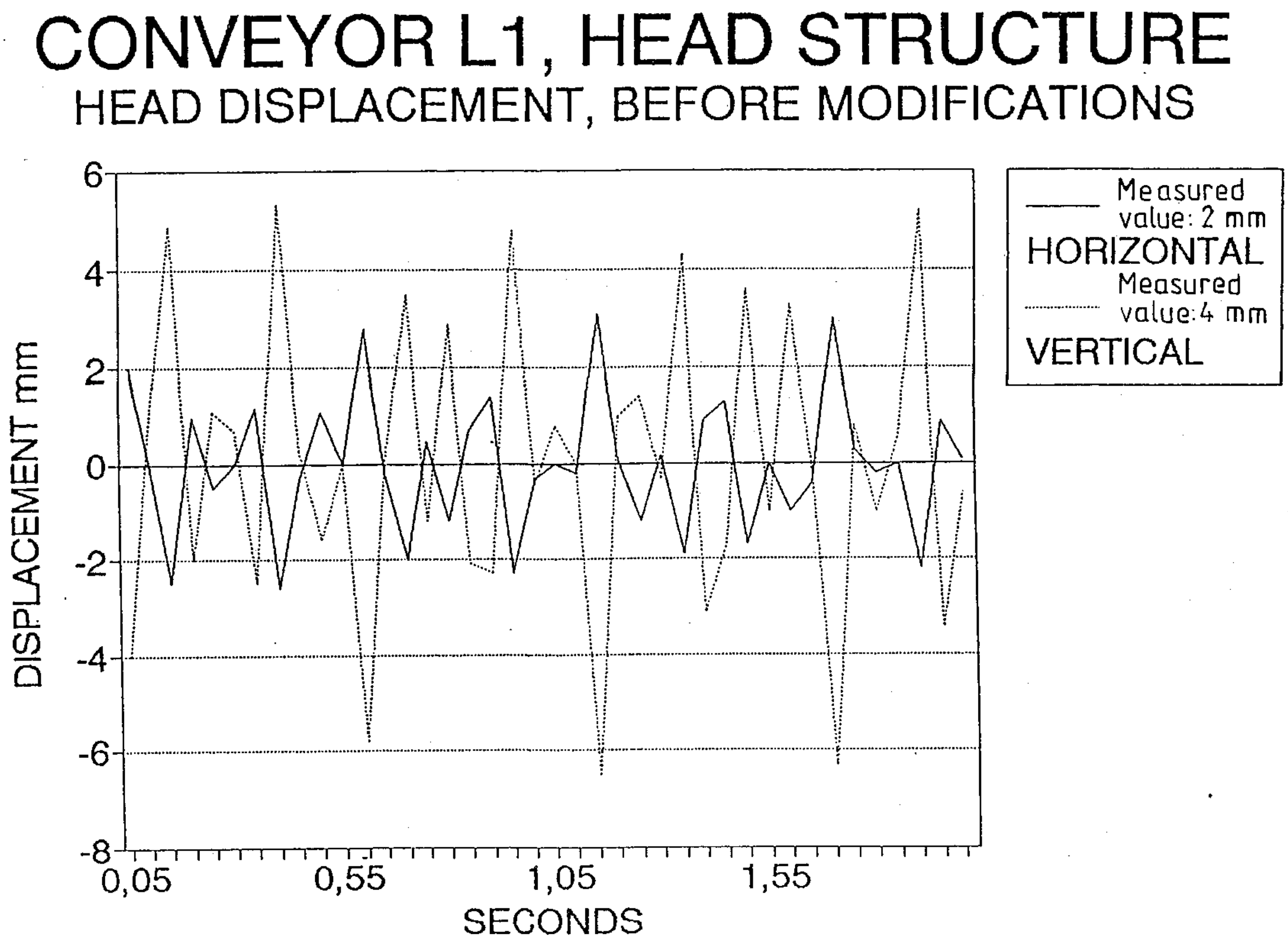
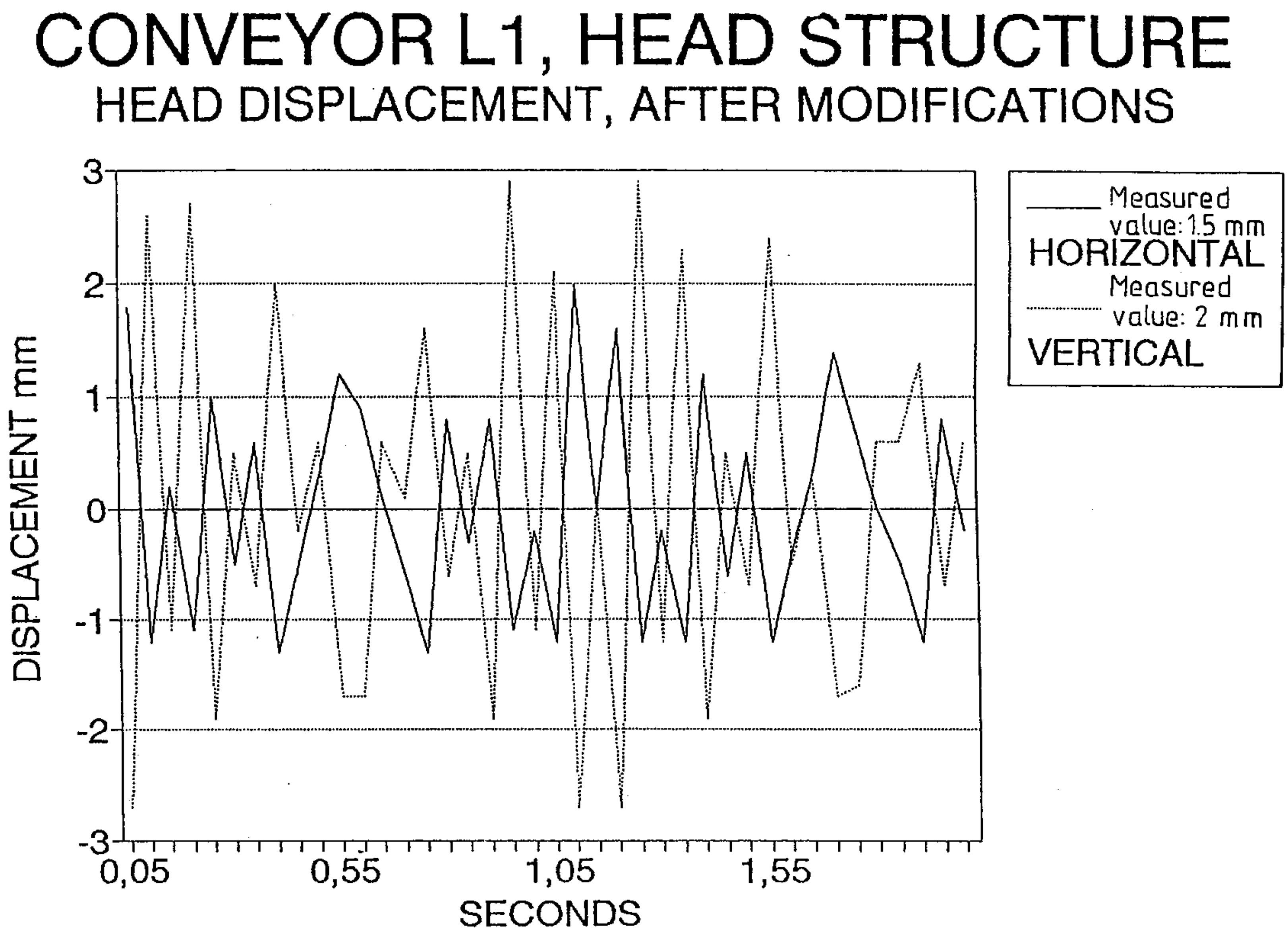

CONVEYOR L1, HEAD STRUCTURE
HEAD DISPLACEMENT, BEFORE MODIFICATIONS
Measured value: 2 mm
HORIZONTAL
Measured value: 4 mm
VERTICAL
DISPLACEMENT mm
6
4
2
0
-2
-4
-6
-8
0,05
0,55
1,05
1,55
SECONDS

Fig. 9

Fig. 10



CONVEYOR L1, HEAD STRUCTURE
HEAD DISPLACEMENT, AFTER MODIFICATIONS
DISPLACEMENT mm
3
2
1
0
-1
-2
-3
0,05
0,55
1,05
1,55
SECONDS
Measured value: 1.5 mm
HORIZONTAL
Measured value: 2 mm
VERTICAL