IMPROVED CONTROL OF A SOLENOID VALVE AND DRAIN COUPLING

By: M. Otrebski
E. Putter
M.F. Scheffer

SUMMARY

The paper looks at a new, improved control concept for a fluid coupling operating in conjunction with a solenoid valve.
Verification of the method has been done by means of conveyor dynamic simulations and some of the results are presented in the paper.

1. INTRODUCTION

Drain fluid couplings have been available in South Africa since 1987 and found their way and acceptance in the bulk solids handling industry. However, initial high expectations have not always been fulfilled, mainly due to some applications which showed a lack of understanding of coupling's limits of performance and / or deficient control systems. The drain coupling can be assisted in its performance by various means [1] and in this respect is exciting to work with for an engineer, as the only limit may be one's imagination and / or finances.
The most common method of torque control relies on a solenoid valve which operates in an on - off mode supported by a PLC system which supervises overall performance within pre - determined limits. The resulting torque curve is characterised by a saw tooth pattern as presented by Fig. 2. The upper and lower torque limits can be adjusted as determined by specific demands.
This technique is sufficient in most instances, however, it may be source of undesirable side effects such as those associated with so called "bang-bang" control techniques. Despite these difficulties, the solenoid valve remains a cost effective and robust device, well suited to the environment of materials handling industry.
Subsequently an attempt has been made to utilise this specific device to its full potential and to develop a suitable control system which would allow improved control of a conveyor during start up.

2. BACKGROUND INFORMATION

For some time research has been performed into continuous control by means of discrete on-off adjustment of control parameters. Trials were performed and systems implemented in areas as far afield from bulk solids handling as switch mode power electronics and agriculture [2, 3]. As an example of the potential of this approach [2] stated that it was possible to maintain pre - determined level of interdependent parameters such as temperature and humidity in a greenhouse by means of discrete action of fans, heaters and sprinklers.
While accepting definite differences in the dynamics of climatic change in a greenhouse and conveyor start up, it was interesting to develop and test the similar concept for the application of a solenoid valve and a drain fluid coupling.

3. THE CONTROL SYSTEM ALGORITHM

The control system arena has in recent years been filled with publications on modern control techniques, such as fuzzy logic control [4] and neural networks [5] as solutions to non-linear and multi-variable control problems. Despite of all the success stories, more classical techniques such as Proportional-Integral-Derivative Control (PID)[3,6], still offers many possibilities for systems with less non-linearities and less interdependent variables. An advantage of PID-control, is that less processor capacity and time is required, than with the intelligent control techniques. However, even though many solutions have been suggested [6], tuning a PID-controller still remains a difficult task, especially in systems where trail-and-error methods are not acceptable. A disadvantage of PID-control is its restricted ability to accommodate system non-linearities.
The solenoid valve and drain coupling lends itself toward application of PID-control. By controlling the switching of the valve, the motor / coupling torque, could maintain a pre-determined pattern. The system dead time is short enough not to contribute significantly to system non-linearity, in which case PID-control may be applied. The control equation is described as follows [7]:

$$\Delta T = K_c \times [\, E(n) - E(n-1) + T_s/T_i \times E(n) + T_d/T_s \times (E(n) - 2 \times E(n-1) + E(n-2)] \quad (1)$$

where:

E - error, difference between set point and observed value.
ΔT - required torque change [Nm];
K_c - proportional gain;
T_d - derivative time constant [s];
T_i - integral time constant [s];
T_s - sample time constant [s].
n - sample number.

Contrary to the conventional PID approach Equation (1) determines the change required relative to the current torque (ΔT) and not the amount of torque required as a function of the error. This is a quicker and more convenient way to determine the output, since no numeric integration or differentiation is required.
The number n refers to the current error measurement, n-1 to the previous measurement and n-2 to the one before that. The time between measurements is critical. A too short sampling time can result in excessive equipment cycling, while a too long sampling time can result in overshoot and instability [8]. The constants K_c, T_i, T_d are originally determined theoretically according to the well known Ziegler - Nichols method [6]. These values serve as a starting point from where further fine tuning was done experimentally. The effect of each of these constants on the controller's performance have been described by Smith [3] and can be summarised as follows [7]:

> A small value of K_c produces large overshoot but gives good stability, while larger values of K_c reduce the overshoot but increase equipment cycling.
>
> Small values of T_i eliminate constant errors quickly, but result in rapid cycling of control equipment. In turn, large values of T_i cause constant errors to occur.
>
> A small value of T_d causes large overshoot, while a large value of T_d increases the reaction time, which results in increased stability.

Although the valve can only be switched on or off, it is still possible to control the switching as if it was a linearly varying valve, by controlling the time for which it is switched on or off. By applying equation (2), ΔT is converted to a duty cycle, which is defined as follows:

$$\text{Duty Cycle} = t_{on}/t_{max} \times 100 \quad [\%] \qquad (2)$$

where:
t_{on} - time for which the valve is switched on ($0 \le t_{on} \le t_{max}$) [s];
t_{max} - time between each sample which also represents maximum time for which the valve can be switched on during one interval ($t_{max} = T_s$) [s].

In fact the conversion may be done in several ways ranging from a simple proportional relationship between ΔT and time of valve operation to a rather complex one where rates of torque/ oil flow change are taken into account. Development and / or selection of the concept may be governed by several factors of which simplicity of the control software and limitations of the hardware are the two most obvious ones.
For this specific application, the valve may not be switched more than 5 times per second or 5 Hz. The optimum length of the maximum duty cycle (t_{max}) may be determined by trial and error according to specified criteria or by dynamic simulations.

4. CONVEYOR SIMULATIONS VERIFYING THE PERFORMANCE OF THE DEVELOPED CONCEPT

The concept has been tested and verified by means of conveyor dynamic simulations.
The first set of simulations utilised a computer model of a conveyor which was a subject of a detailed design investigation by Dynamika Materials Handling two years ago. The conveyor was supplied with drain couplings working in conjunction with a three way valve. Detailed information about the system and the conveyor may be found in [9]. The coupling characteristics utilised are similar to that presented in [11] as it is a representative of drain fluid couplings used in South Africa.
The torque based starting strategy was analysed in significant detail. Acceptable performance results were obtained by adjusting the duty cycles and control settings. This led to further exploration of a velocity based starting strategy.
To complete the investigation, the simulations were extended to conveyors of greater length and higher installed power. Some of the graphs presented in this paper refer to an 8,5 km long overland conveyor with head and tail drives [10]. Although the actual system operates with scoop couplings, in these simulations the conveyor was modelled with drain couplings operating in conjunction with solenoid valves.

4.1 RESULTS OF THE CONVEYOR DYNAMIC SIMULATIONS

The influence of various lengths of the duty cycle on the performance, were investigated. In Fig. 4 to Fig. 10 results of simulations for the three different duty cycles (CYCLE 1, CYCLE 2, CYCLE 3) are shown. For each duty cycle two groups of control settings, named A and B, are compared. For reference purposes results of simulations for conventional mode of on-off valve control are included with limits set at +- 5% (Fig. 1 and Fig. 2) and +- 10% (Fig. 3).
Torque was ramped at a rate of 116,0 Nm/s which is suitable for a 3,3 km long conveyor with steel cord belting (2100 Nm should be reached in 15 sec). The resulting ramp is too steep for the coupling during the initial stages, specifically during the initial 6 seconds of pump operation. Consequently, the control system comes into action with an approximate delay of 12 seconds.

Attention is drawn to the results of CYCLE 2, control settings A (Fig.6) and CYCLE 3, control settings A (Fig.8) and B (Fig.9), where the control system is able to produce a smooth line of torque rise. In all the other graphs a distinctive step can be noted, resulting from on / off action of the valve.
Importance of proper fine tuning of the control constants is underlined by the results produced by CYCLE 1 with control settings B, where a specific combination of duty cycle and control constants produced results which were inferior even to normal mode of operation (compare Fig.1, Fig.3 and Fig. 5).
Duty cycle, CYCLE 3, with control settings B was selected for a simulated conveyor start up using a linear velocity ramp. Fig.11 and Fig.12 present results for average acceleration of 0,06 m/s^2 and 0,04 m/s^2 .

Two problems are apparent from the presented graphs:

1. Initial low torque delivery does not allow instant acceleration of the conveyor. This is possible to be rectified by adjustments in the software as is the case for the simulations. It can be noted that the velocity ramp and controlling action starts some 8 seconds after motors and 6 seconds after couplings' pumps were energised.

2. The rate at which oil is discharged from the coupling is limited and may be insufficient to counter act dynamic reaction of the conveyor and as a result, a certain degree of overspeed can be noted in the initial stages of the start up.

Despite the mentioned problems, it may be stated that the results of the velocity ramp start up are significantly better than can be expected from a discrete type of control operation.
Further tests, utilising different velocity curves and / or refined tuning of the control constants may produce better results.
As mentioned earlier, simulations were done to verify above findings in case of longer conveyor. Fig.13 shows the results of simulations for an 8,5 km long overland conveyor with 5 drives situated at both the head and tail ends.
The overall results are acceptable, however, it can be noted that the performance of tail end drive is somehow better that those at the head end.
For each drive it is possible to notice a period of increased oscillations of torque / tensions between the period of 12 and 22 seconds in the start up. Once again, by adjusting settings and / or the length of the duty cycle, the results can improved.

5.0 CONCLUSIONS

Based on the results of simulations the following conclusions can be made:

- The objectives which were set for the control concept of the solenoid valve have been achieved to a large extent. It was possible to convert the discrete action of the valve into a precise, continuous output.
- New system allows application of the solenoid valve for velocity based starting strategy, however, control may not be as precise as for a torque based strategy, specifically during initial stages of the start up.
- Care must be taken in fine tuning of the control parameters. Simulation results have shown that badly selected control parameters may lead to poor performance results.
- As was shown the concept may be applied both to very long and medium length conveyors. Existence of the tail drive is not a restriction for the application, however, only torque based starting strategy has been tested in this case.
- PID-control is a classic control technique which is widely applied in the process industry and which has provided good solutions to diverse control problems, provided that it is well-tuned.

6.0 ACKNOWLEDGMENTS

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REFERENCES

1. Otrebski M.: Torque Control Using Drain Fluid Coupling; bulk solids handling Vol. 16 (1996) No. 1, pp 39 - 41.

2. Putter E., Gouws J.: An Automatic Environmental Controller for Greenhouses; Elektron (Journal for South African Institute for Electronic Engineers) August 1996, pp 66-67.

3. Smith C.L.: Digital Computer Process Control; Intext Educational Publishers, New York, 1972.

4. Zadeh L.A.: Similarity Relations and Fuzzy Orderings; Information and Science, Vol 3 1971, pp 177-200.

5. Kosko B.: Neural Networks and Fuzzy Systems - A Dynamical Systems Approach to Machine Intelligence; Prentice-Hall International Editions, New Jersey 1992.

6. Franklin G.F., Powell D.J., Emami-Naeini A.: Feedback Control of Dynamic Systems; Addison-Wesley Publishing Company, 1986.

7. Putter E. : Multi-variable Control Techniques for Greenhouses; Master's Thesis (Electrical and Electronic Engineering), Rand Afrikaans University 1996, South Africa.

8. Macdonald R.D., Hawton J., Hayward G.L.: A Proportional-Integral-Derivative Control System for Heating and Ventilating Livestock Buildings; Journal of CSAE, Vol 31, 1989, No. 1, pp 45-49.

9. Pretorius J.L.: Design of a Long Conveyor System with Combined Ascending and Descending Sections; bulk solids handling Vol. 16 (1996) No. 2, pp 195 - 201.

10. Fauerbach R., Otrebski M.: The Syferfontein Overland Conveyor System at the Sasol Secunda Plant in South Africa; bulk solids handling Vol. 13 (1993) No. 2, pp 289 - 295.

11. Page J. L. et al.: Design of Long Overland Conveyor with Tight Horizontal Curves; bulk solids handling Vol 14 (1994) No. 1, pp 21-29.

MARIAN OTREBSKI graduated in 1976, and obtained a Master's degree in Mechanical Engineering at the Technical University of Wroclaw. Whilst in Poland he worked for the brown coal industry, both in the operation and design of the machines. In 1982 Mr. Otrebski emigrated to South Africa and has since been involved in the design of bulk materials handling systems for various South African projects.

ESTELLE PUTTER obtained a degree in Electrical and Electronic Engineering in 1994 at the Rand Afrikaans University, Johannesburg, South Africa. Her studies was furthered towards a Master's Degree in Control Systems, specializing in multi-variable control techniques for application in greenhouses. She is currently studying towards a PhD in Control Systems. She is a member of the South African Council for Automation and Control, as well as the Air & Waste Management Association in the U.S.A., and is one of the founder members of the South African Association for Women in Engineering. Her fields of interest include Non-linear Modelling, Intelligent Control Techniques, Machine Vision, Chemical Process Control and Control Systems for Materials Handling.

MARTEN F. SCHEFFER obtained a degree in Electrical and Electronic engineering in 1991 at the Rand Afrikaans University, Johannesburg, South Africa. In 1993 he was awarded a M. Eng in Control System Engineering, specialising in real - time and intelligent control methods. He was awarded a Ph D in Teletraffic Methods and Systems in 1997. He is a member of the IEEE and the South African Council for Automation and Computation. His fields of interest include Non-linear Modelling and Control, Bulk Solids Handling Systems and Intelligent Control.

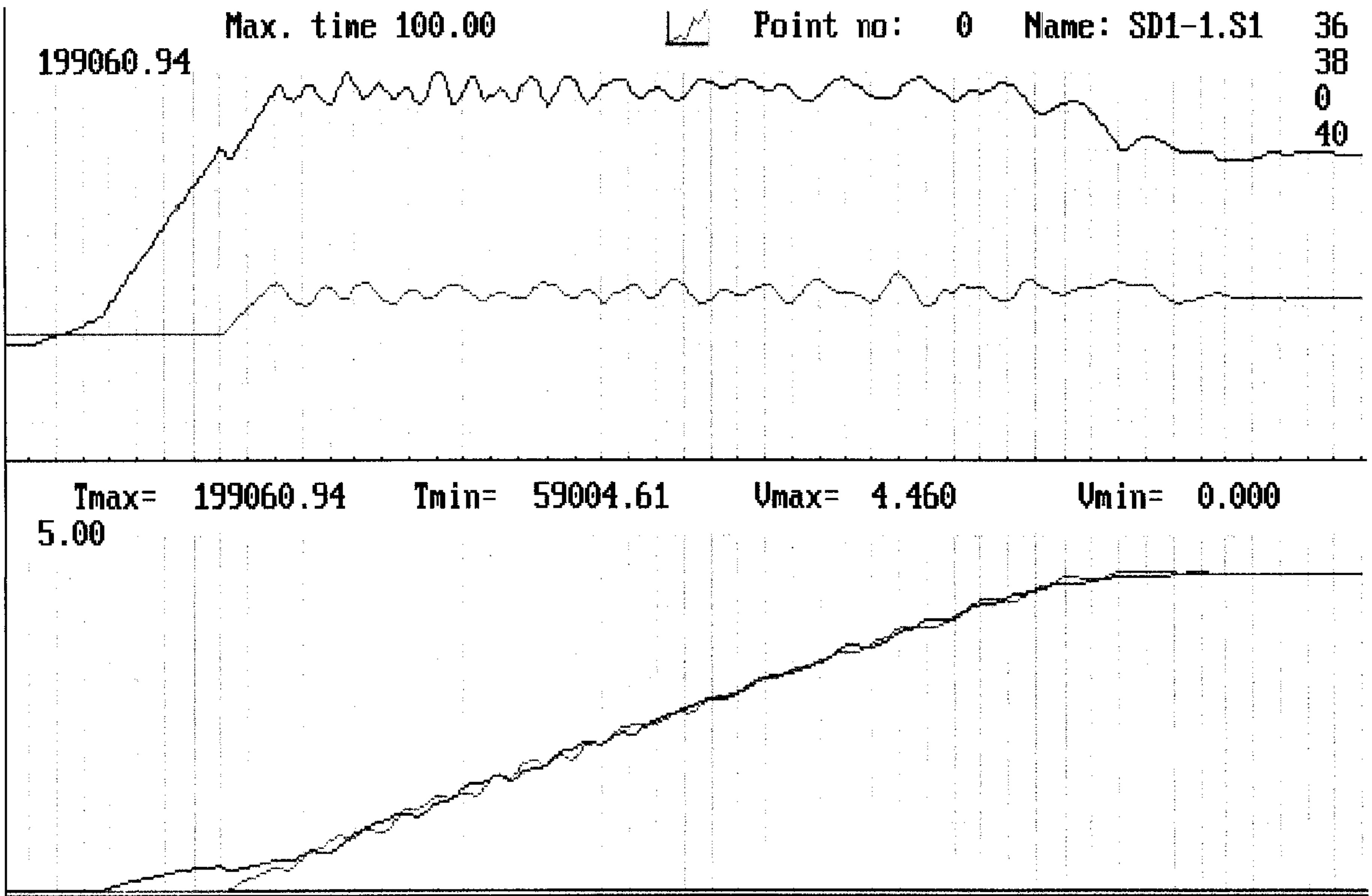

Fig.1 Loaded conveyor start up. Conventional valve operation with limits of torque set at +-5%. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

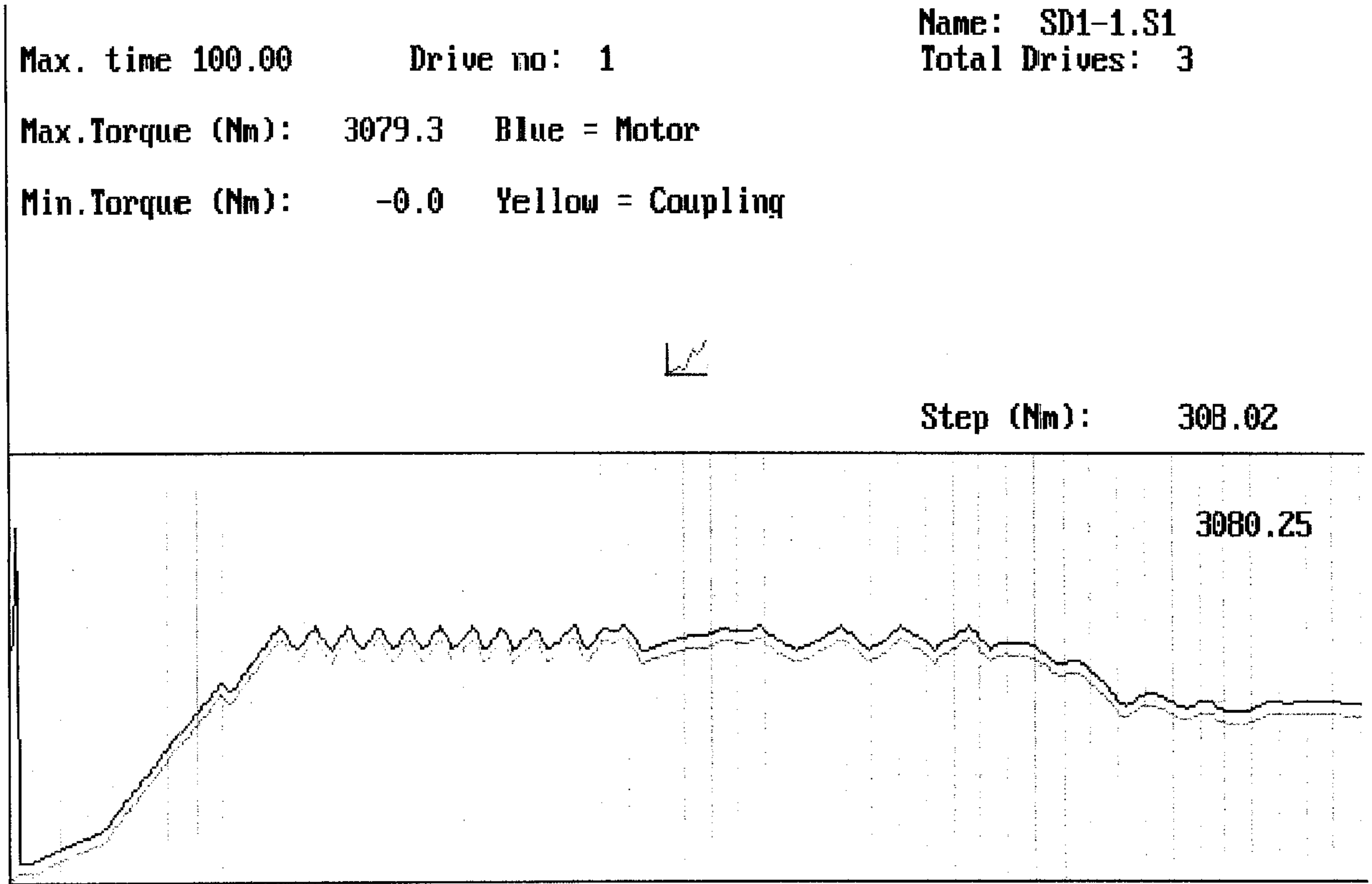

Fig.2 Loaded conveyor start up. Conventional valve operation with limits of torque set at +-5%. Graphs of motor and coupling torques [Nm] of drive no.1

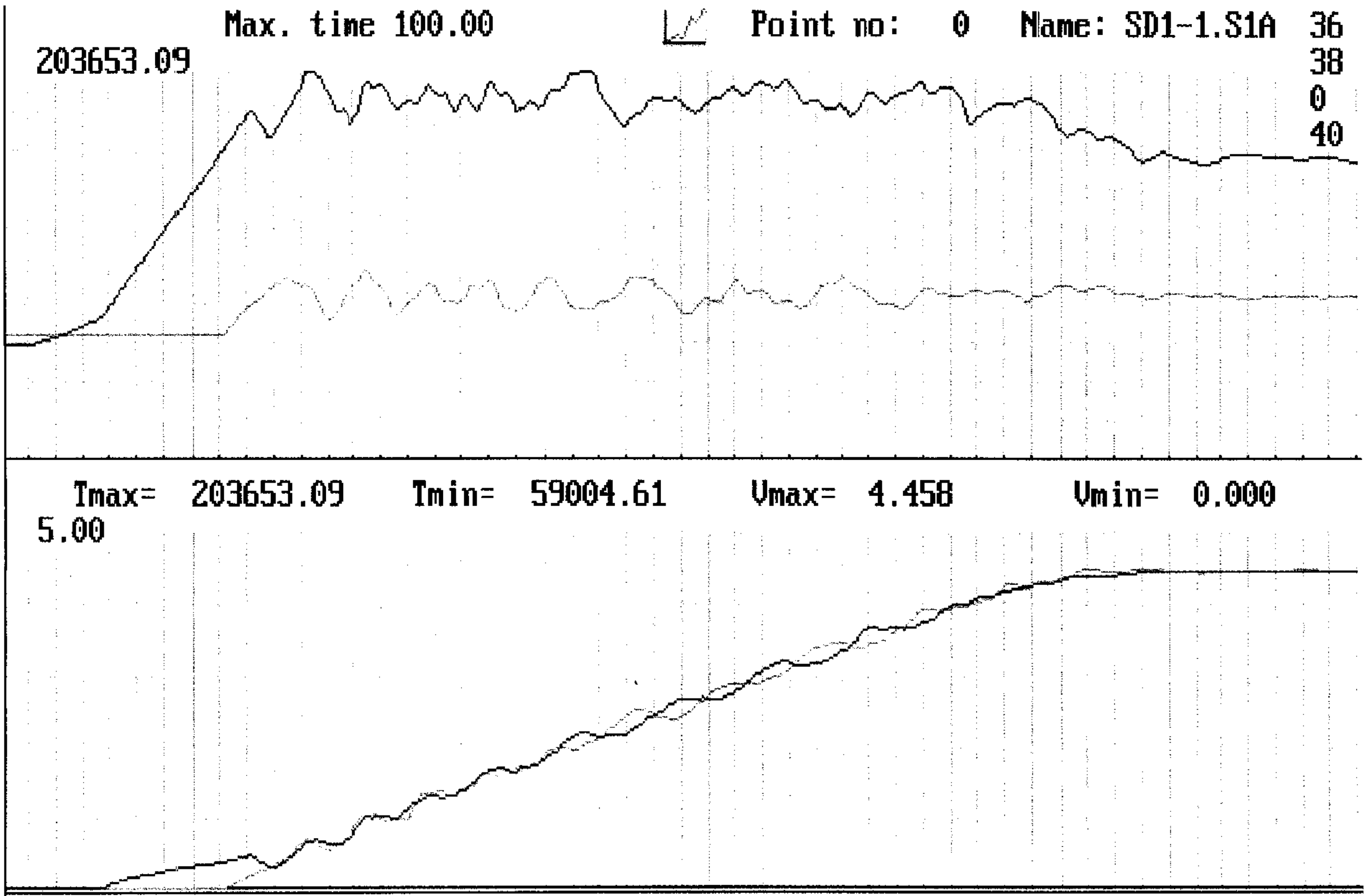

Fig.3 Loaded conveyor start up. Conventional valve operation with limits of torque set at +-10%. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

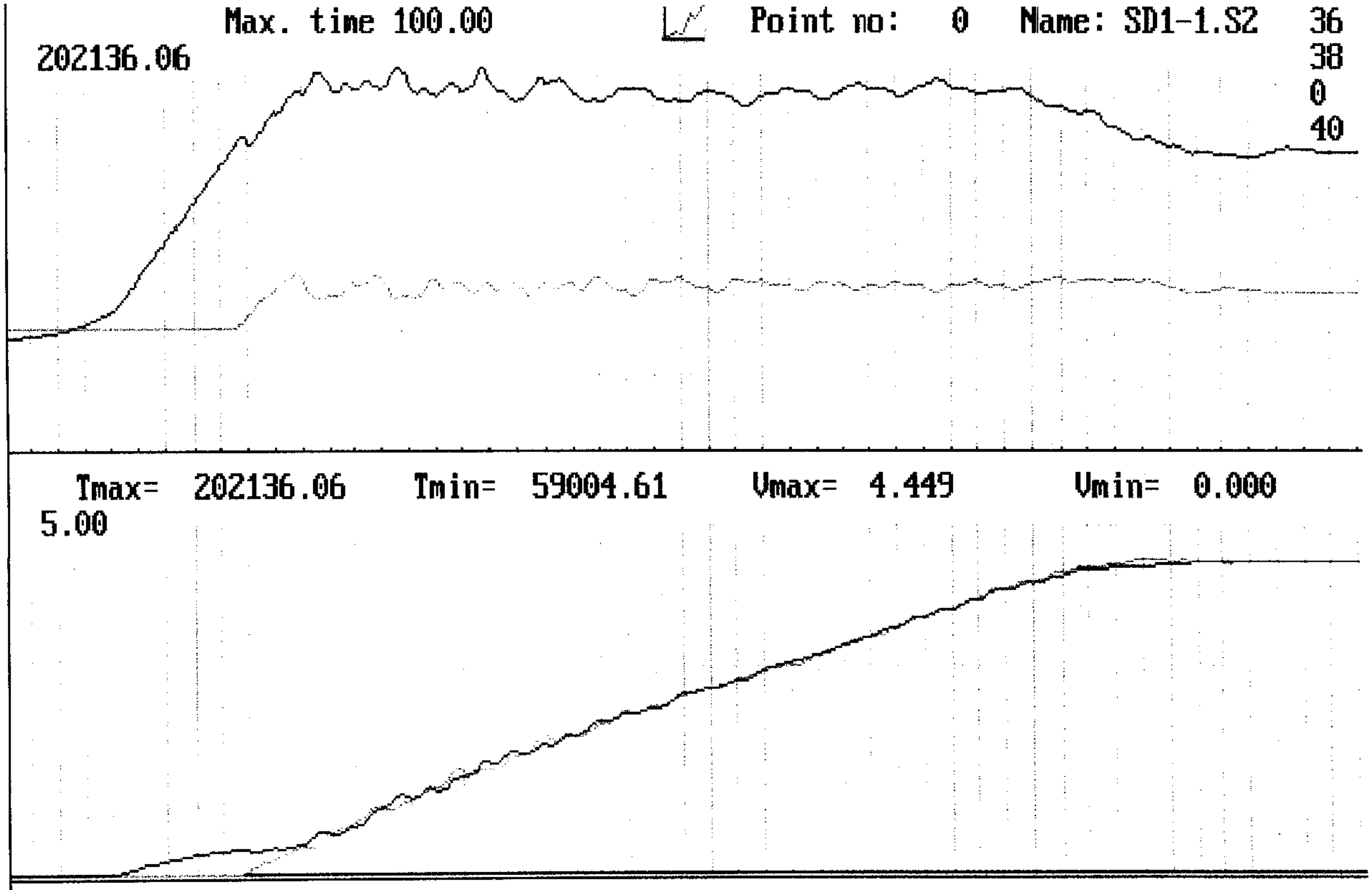

Fig.4 Loaded conveyor start up. Modified valve operation CYCLE1, settings A. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

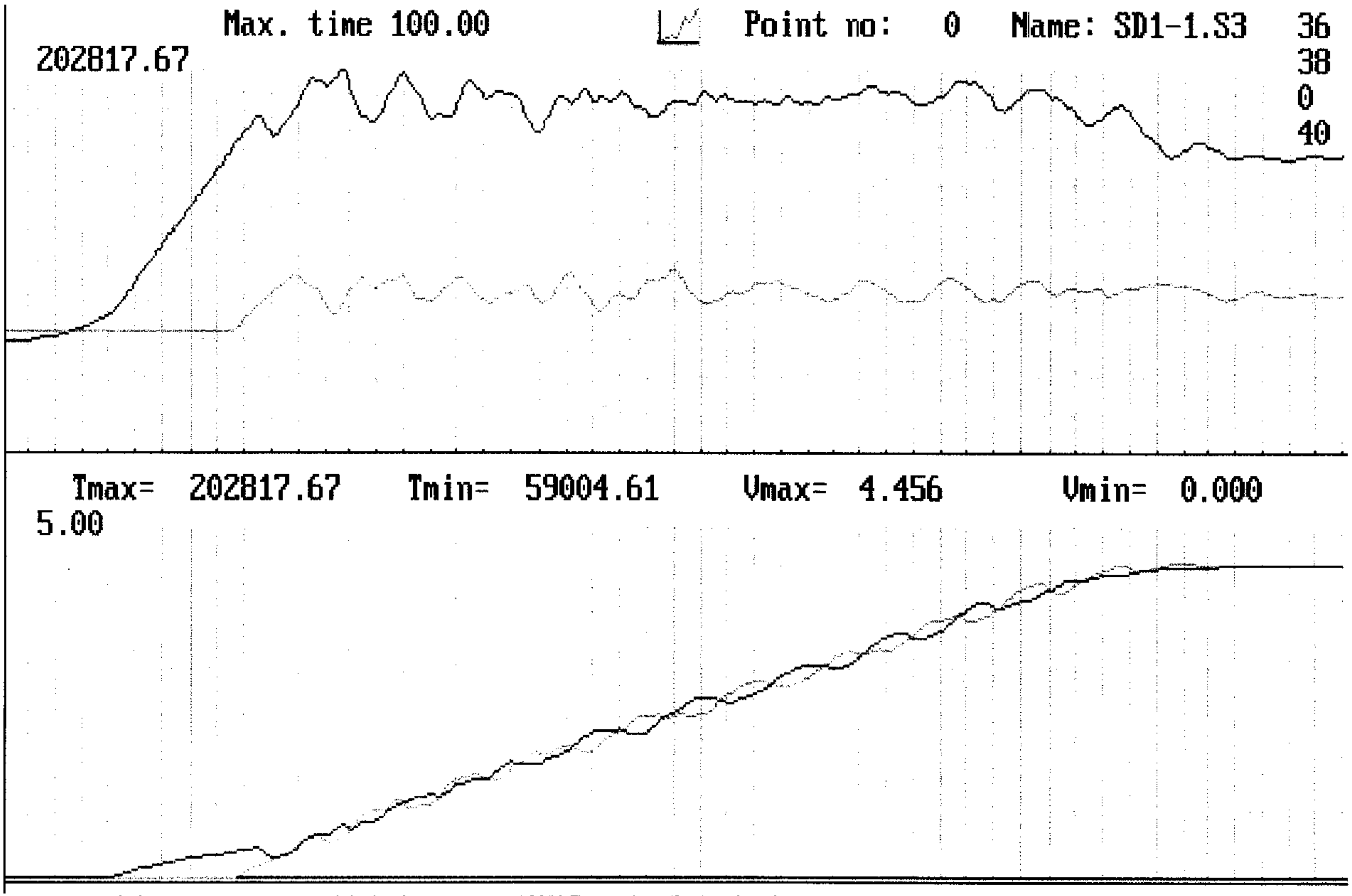

Fig.5 Loaded conveyor start up. Modified valve operation CYCLE1, settings B. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

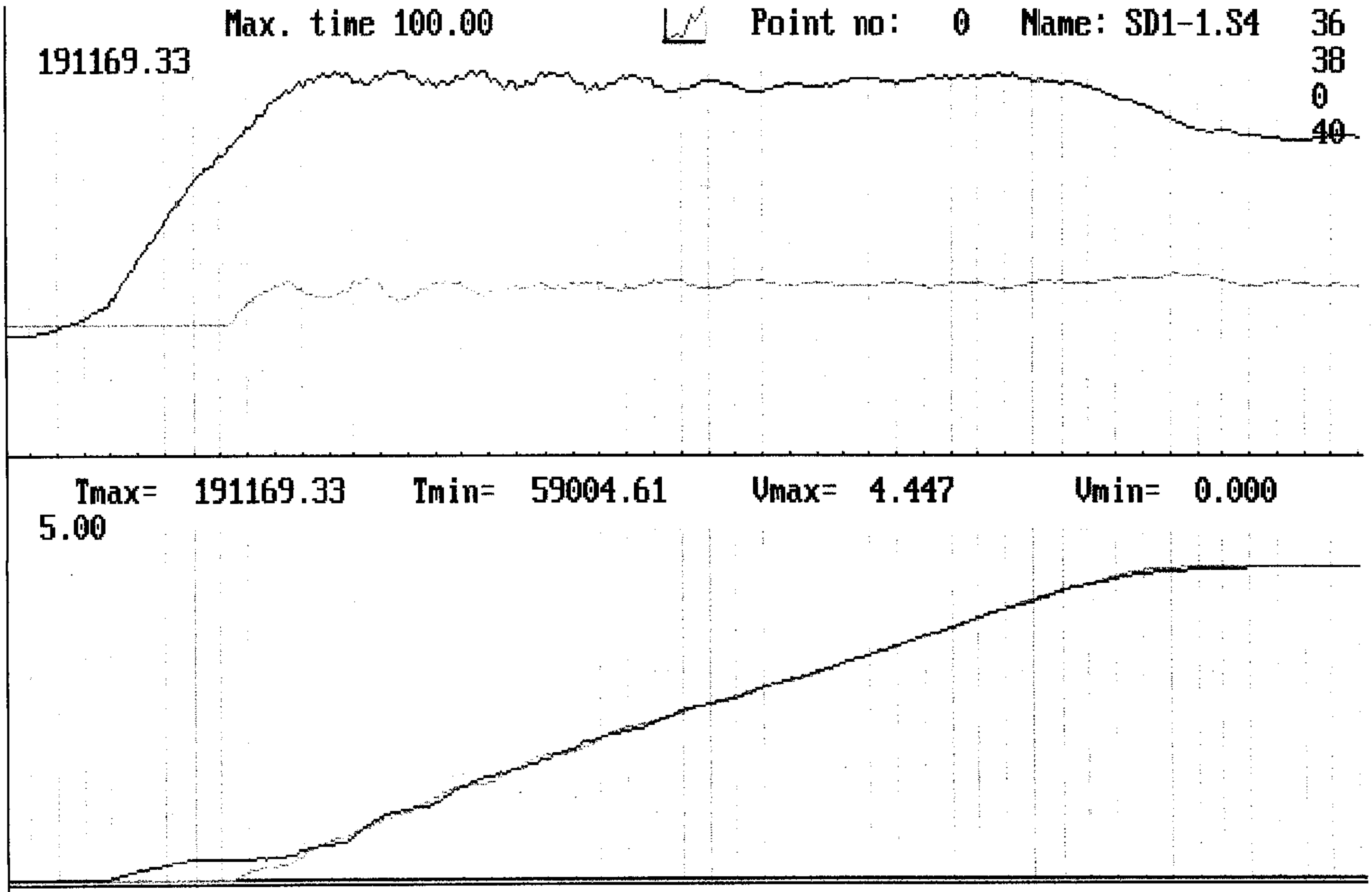

Fig.6 Loaded conveyor start up. Modified valve operation CYCLE2, settings A. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

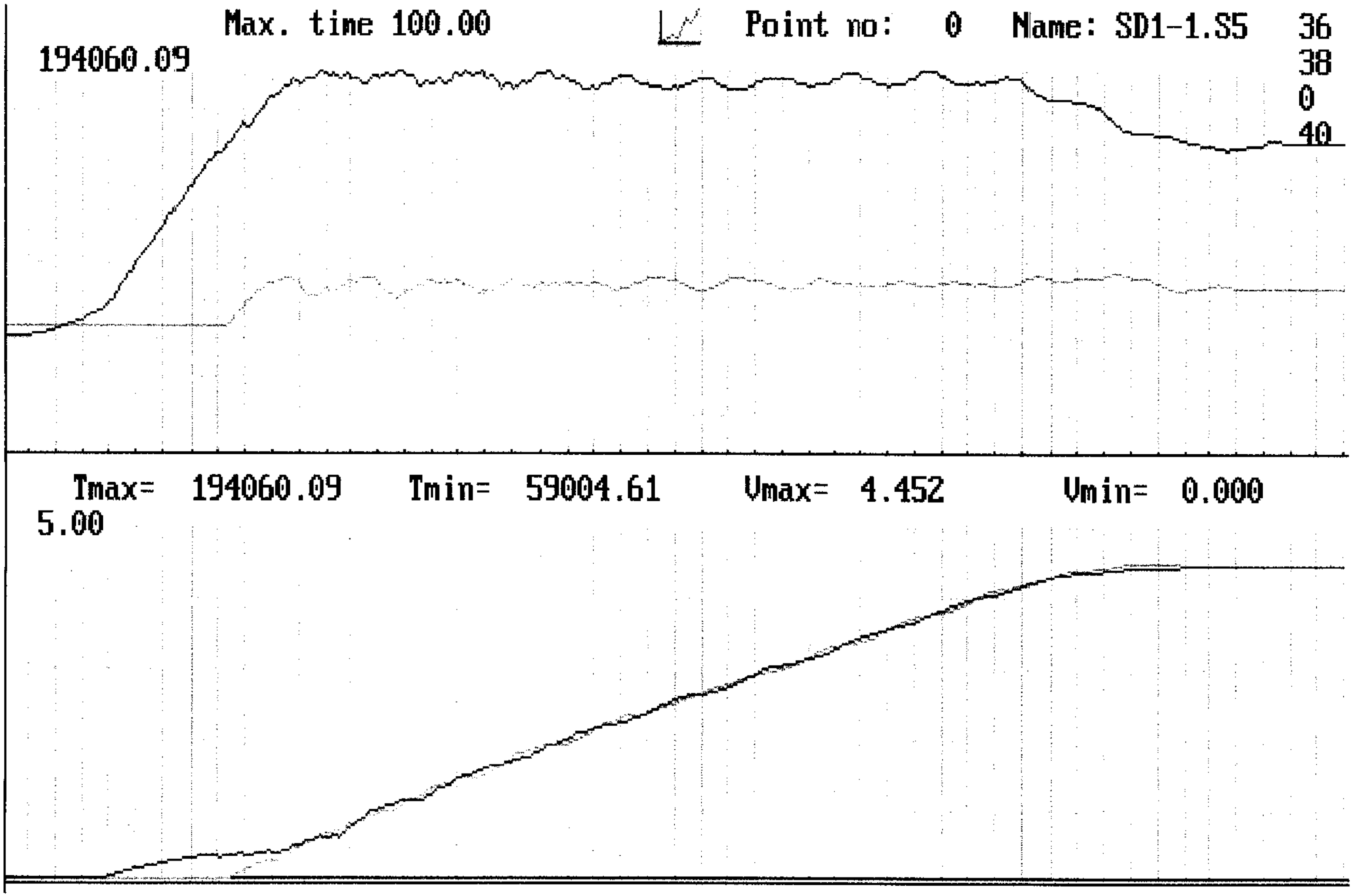

Fig.7 Loaded conveyor start up. Modified valve operation CYCLE2, settings B. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

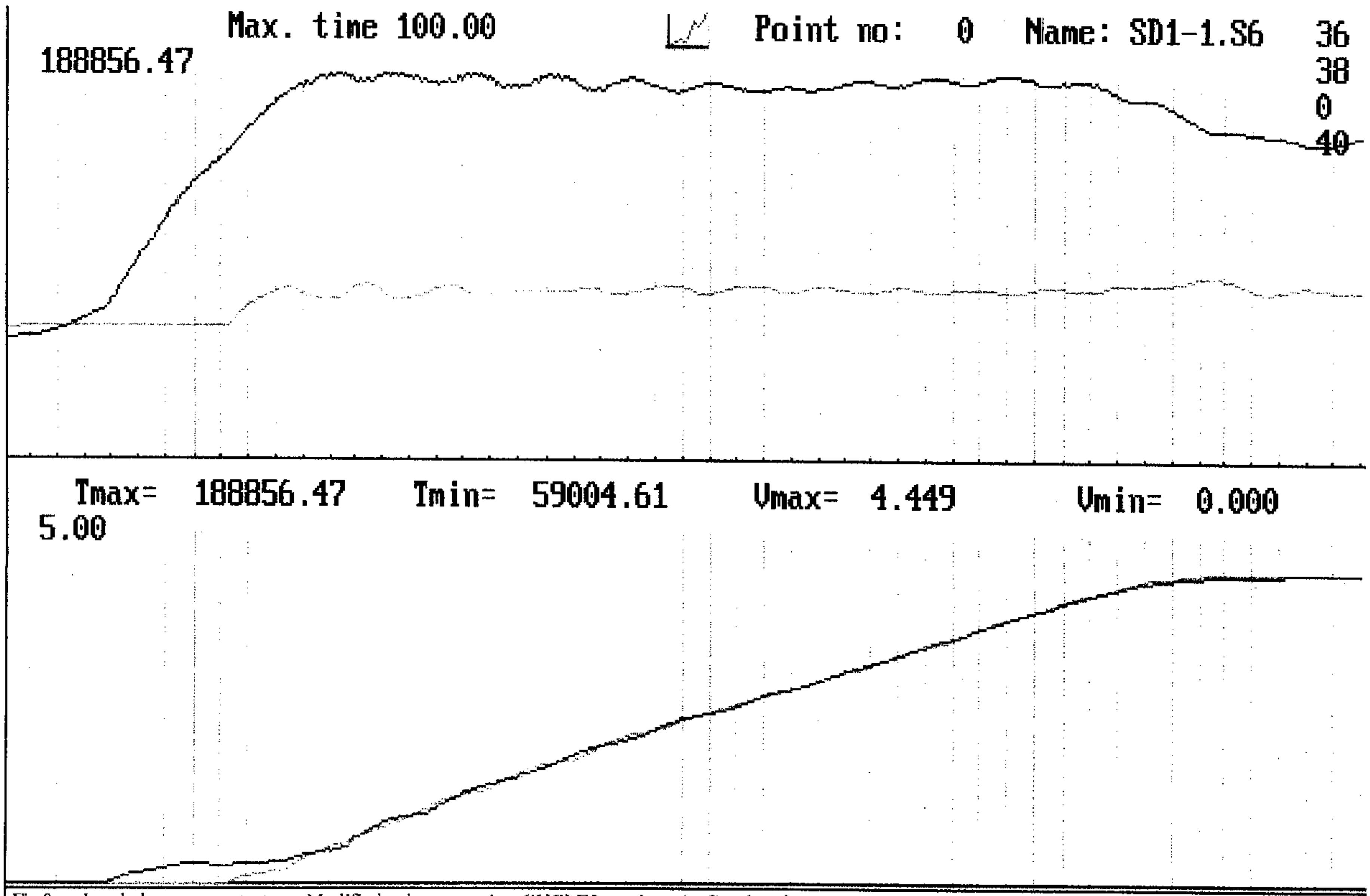

Fig.8 Loaded conveyor start up. Modified valve operation CYCLE3, settings A. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

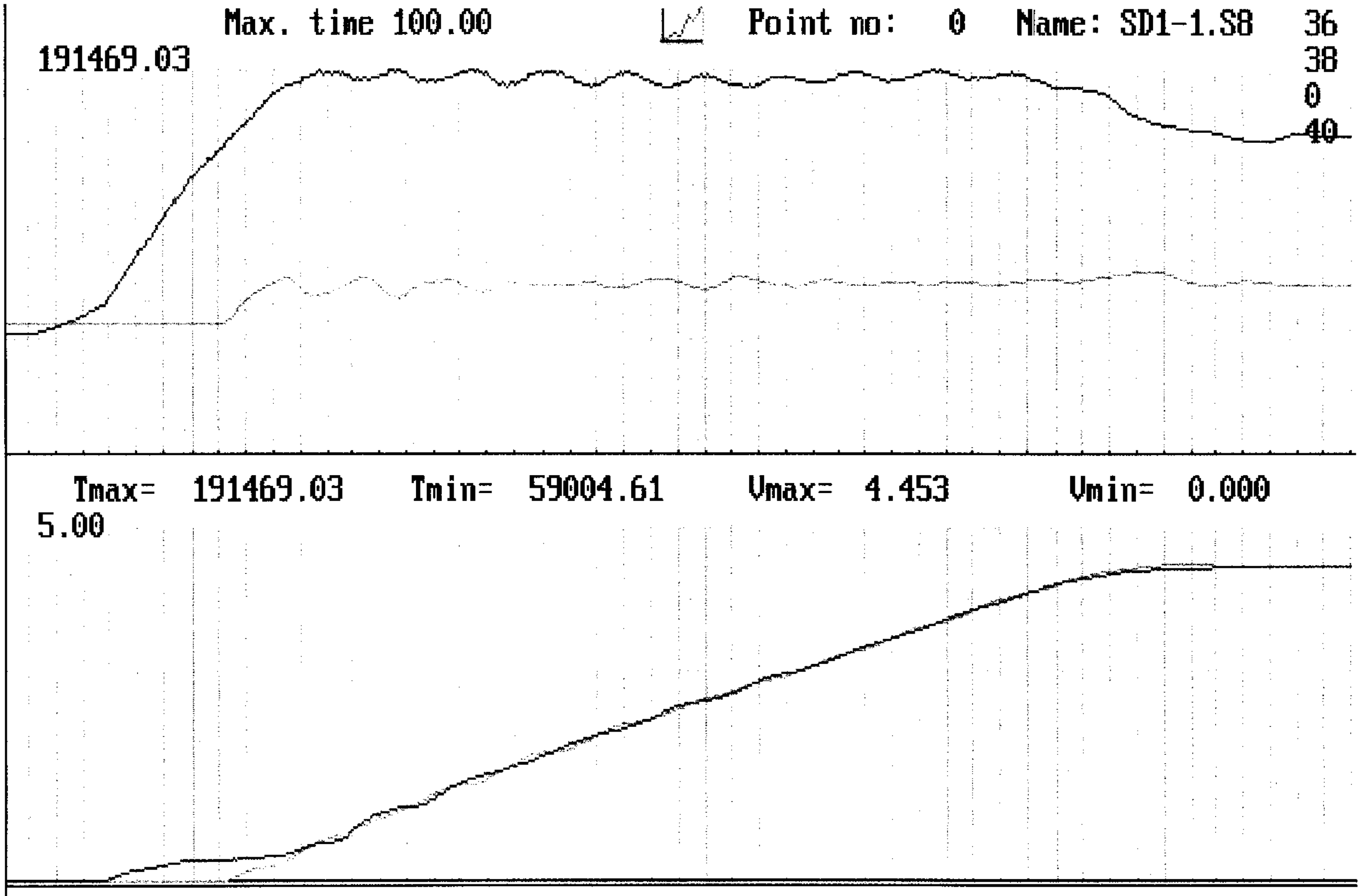

Fig.9 Loaded conveyor start up. Modified valve operation CYCLE3, settings B. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

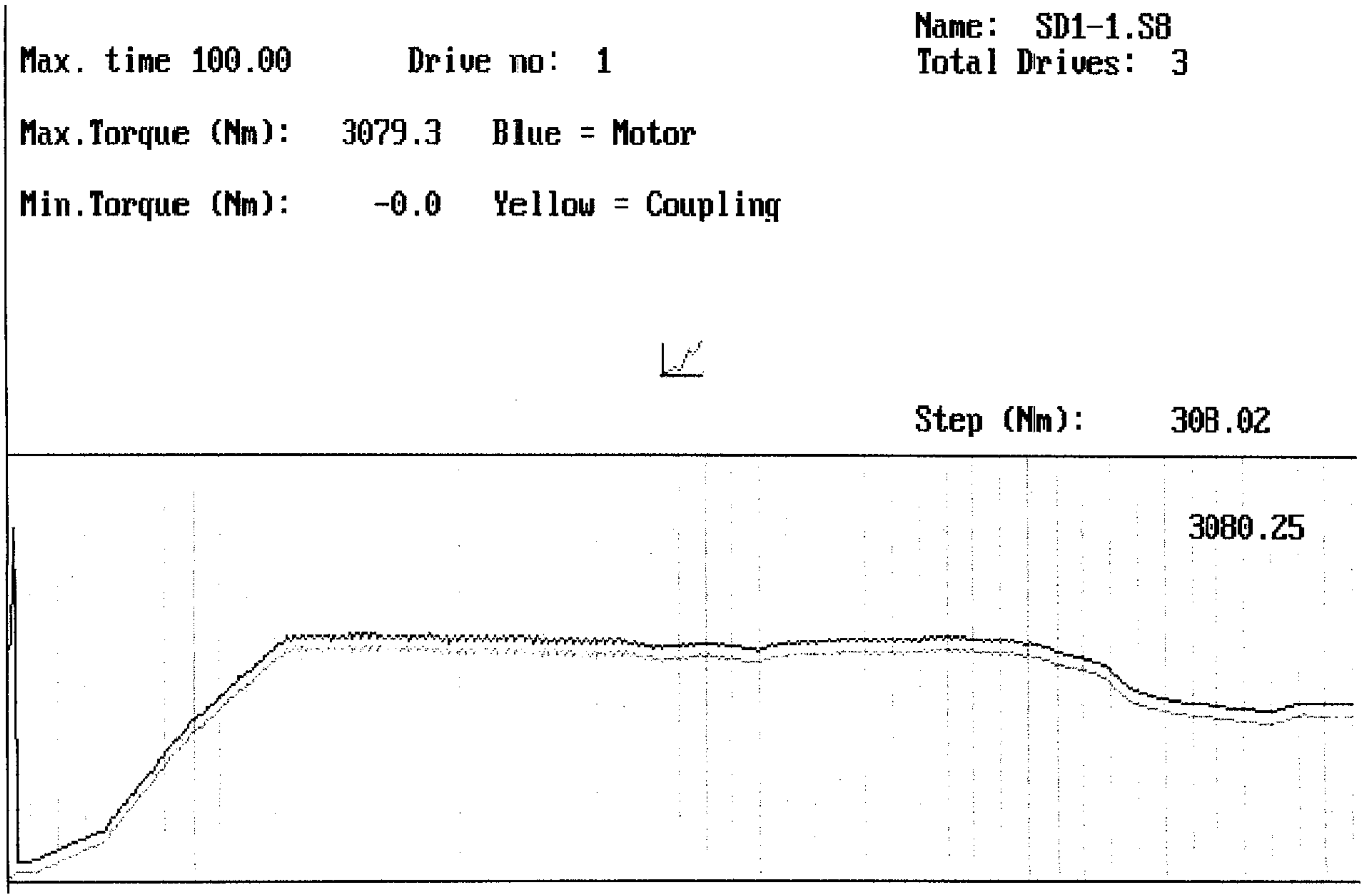

Fig.10 Loaded conveyor start up. Modified valve operation CYCLE3, settings B. Graphs of motor and coupling torques [Nm] of drive no.1

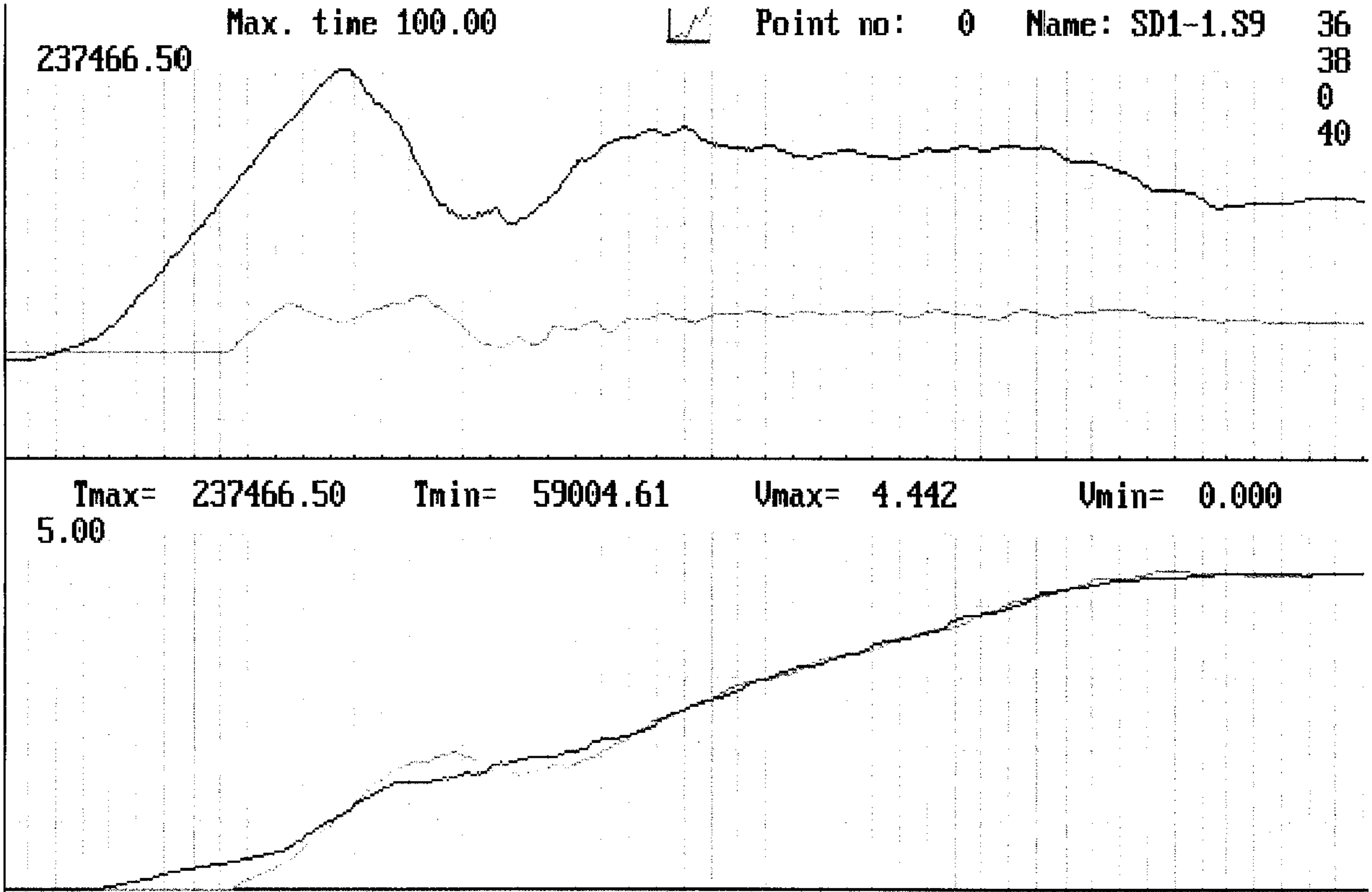

Fig.11 Loaded conveyor start up. Modified valve operation.Linear speed ramp 0,06 m/s². Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

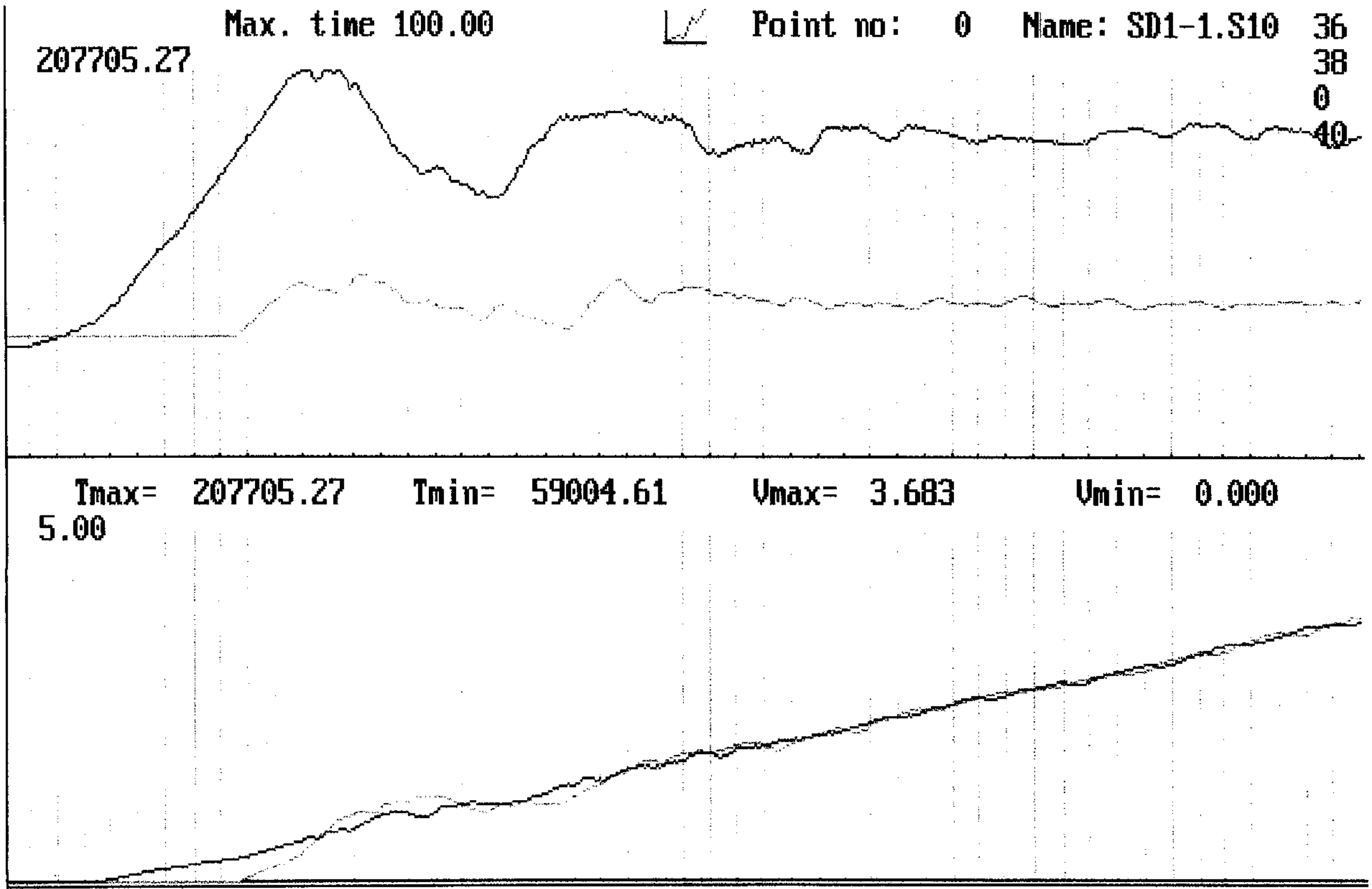

Fig.12 Loaded conveyor start up. Modified valve operation.Linear speed ramp 0,04 m/s^2. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends.

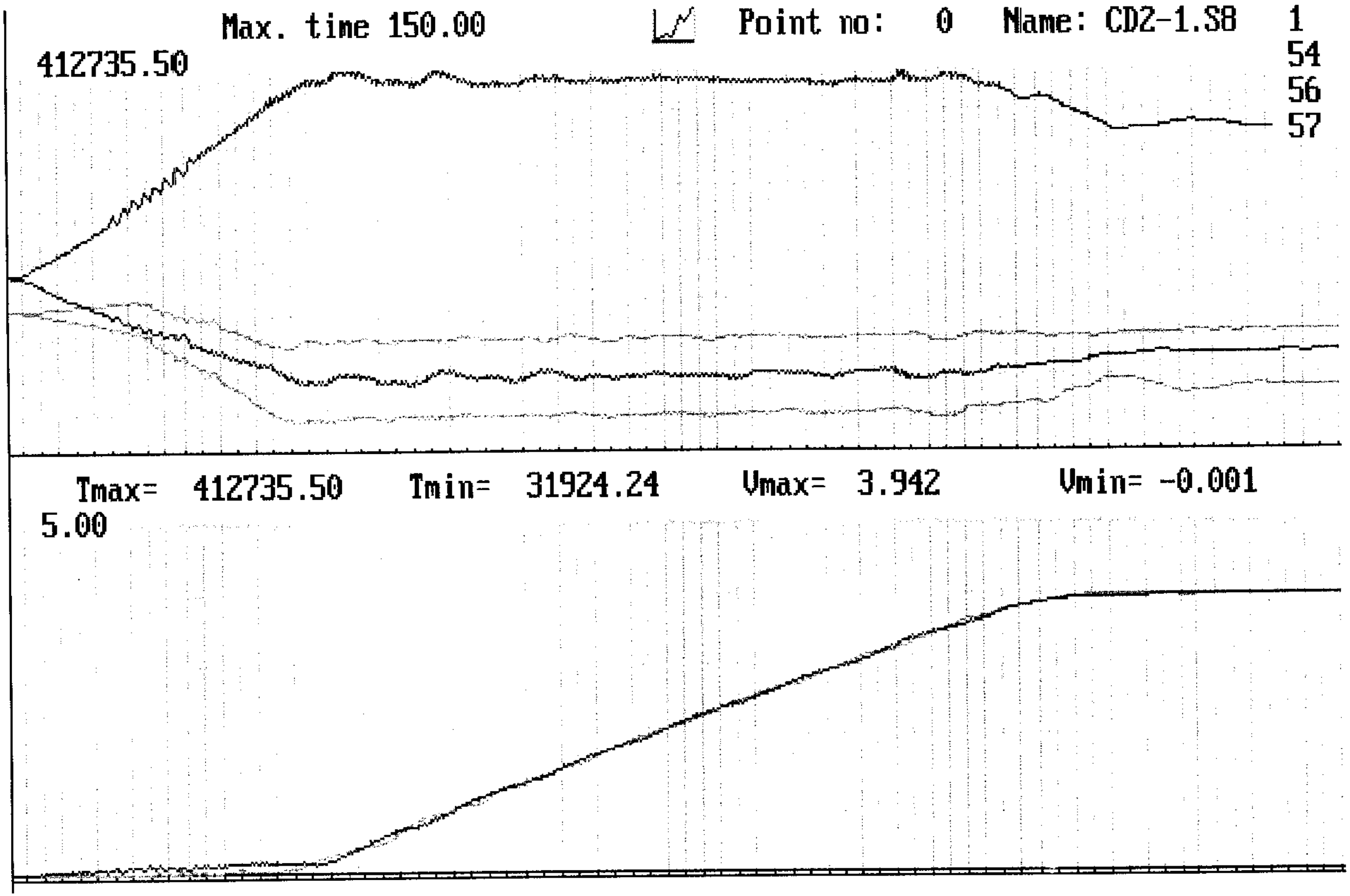

Fig.13 Loaded conveyor (8.5 km long, head and tail drives) start up. Modified valve operation. Graphs of tension [N] (upper) and velocity [m/s] (lower) at the head (black) and tail ends (T_1 and T_2).