Active mooring line transducer for small scale physical model tests

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Proceedings of the 5th International Conference on The Application of Physical Modelling to Port and Coastal Protection, Coastlab14, Varna, Bulgaria, 29 September – 2 October 2014.

Abstract

Physical models of moored floating structures are used extensively when looking at complex wave-structure interactions such as those involved in vessel downtime & mooring analyses. In order to correctly represent the motion of a vessel at berth, the highly nonlinear mooring line characteristics need to be correctly re-created at scale. A novel active mooring line transducer (AMLT) is described, which reproduces the non-linear characteristic of mooring lines, without the restrictions of linear springs.

Keywords: mooring line; active control, active transducer, hysteresis.

1. Introduction

Physical modelling of moored floating structures is still used as a reliable method for representing the complex interactions between water, structure and moorings, despite advances in computing power and numerical modelling techniques. When correctly constructed, a physical model will give a good representation of the transfer of energy from waves to a floating structure and from the floating structure to a berth or mooring structure. The final transfer of energy from the floating structure to the berth or mooring structure takes place through mooring equipment, often fenders and mooring lines. It follows that to get an accurate representation of the energy flow then the energy absorption characteristics of the mooring equipment must be well replicated in the physical model and preferably the forces exerted upon the mooring equipment should be measurable in order to inform the outcome of the model.

It is common practice to represent the mooring lines or fenders using either cantilevers or coil springs with integrated strain gauges for measuring forces. Both cantilevers and coil springs provide an attractive solution thanks to their apparent economy, reliability, simplicity of use and the ability to tune the stiffness of each spring or cantilever to match the equipment it is being used to model.

Despite the success of these methods there are limitations and drawbacks to their use. When faced with modelling a highly nonlinear mooring line or fender response the use of coil springs and cantilevers becomes increasingly impractical. It is possible to connect multiple coil springs in series with pre-defined stops, or to add stops to a cantilever which result in a number of linear responses being formed to approximate the non-linear load extension curve- as in the example shown in Figure 1. Although acceptable for many purposes, the length of time taken to set up a series of springs increases as the series becomes more complicated. As a number of different mooring configurations will often be tested, the economy of using this system can soon be negated by the number of man hours required to effectively set up the different mooring lines.

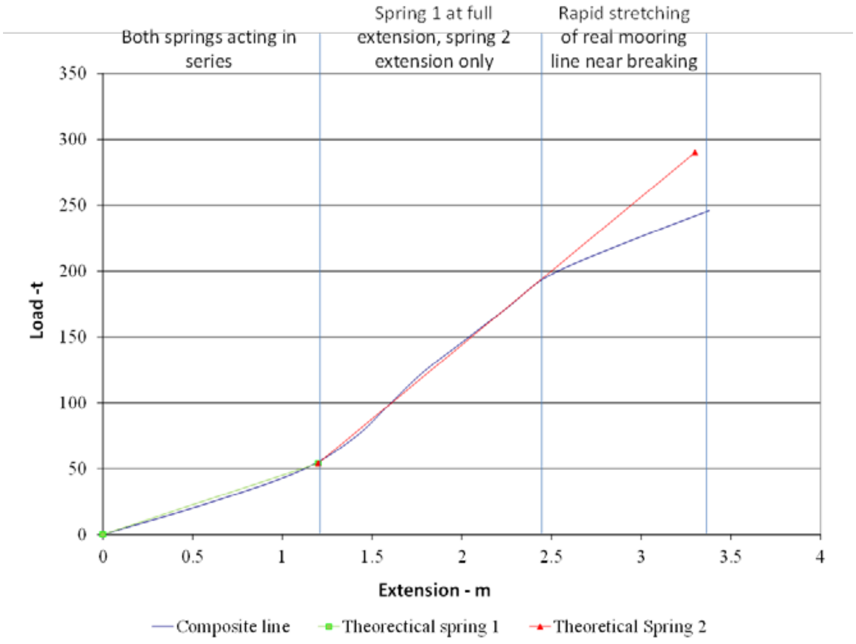

Figure 1. Errors associated with the use of linear springs to represent non-linear mooring lines.

In response to these limitations and drawbacks HR Wallingford has developed a technique that uses servomotors to replicate the stiffness characteristics of the mooring line. Servomotors are motors in which the position of the motors shaft is fed back to a control unit allowing control of not only the motors speed but also the shaft position to be set. When combined with a high precision driver and control logic it is possible to provide precise control and monitoring of the motors actions. This provides a versatile tool for accurately representing mooring lines whilst providing the user with greater overall capabilities and being able to accurately model a wider range of mooring lines and scenarios.

2. Concept

The concept of using a servo motor to replicate a mooring line is based upon having precise knowledge of the pay-out of a mooring line connected to a floating structure, relative to a set position, and from this pay-out being able to apply an appropriate load in order to reproduce the required load extension curve for the mooring line. In order for this system to work then there is an assumption that the relationship between the current delivered to the servomotor by the driver and the resultant load delivered to the line is linear and free from hysteresis for the working range of the system. Even with high quality servo motors this cannot be guaranteed and it was identified that there would be a requirement for a feedback control system capable of monitoring the actual load in the line and modifying the current supplied to the motor in order to achieve the load.

The key advantage of using servomotors to represent mooring lines in a physical model is the ability to reproduce any load-extension curve, given that the maximum required load does not exceed the maximum load the motor is capable of delivering. However it is important to identify and measure possible limitations and sources of error which could be associated with using an active system to model a passive piece of equipment, principle amongst these are:

- Time lag between reading position and applying force;
- Error in the measurement of force being applied;
- Instability of torque being applied by the motor.

Feasibility studies investigating the use of the system were undertaken by HR Wallingford in order to identify whether these limitations could be overcome, before considering the use of active mooring line transducers on a physical model.

3. System configuration

3.1. Hardware

The design of the active mooring line transducer was to attach the mooring line to the servomotor using a pulley, the mooring line then passed through an eyelet providing a set position to act as a fixed attachment position e.g. the bollard position on a quay. After passing through the eyelet the mooring line was attached to an inline strain gauge used to measure line loading.

A first set of physical model tests revealed the importance of having a fast response time. The system was then re-built using state of the art industrial servomotors produced by Beckhoff Automation GmbH and controlled through their proprietary TwinCAT system (Table 1). The Beckhoff system was chosen as it is modular (and so flexible) and capable of taking readings, processing data and running control logic and transmitting commands at up to 1000 Hz.

The strain gauges used during testing were C-spring type with Wheatstone bridges used to measure the force. These C-Springs weighed approximately 10g and were monitored using the Beckhoff system.

Table 1. Specification of Beckhoff AM3111 Synchronous servomotor.

Quantity	**Value**
Standstill torque	0.21Nm
Standstill current	3.22A
Rated speed	3000rpm
Supply voltage	24V
Pulley diameter	55mm

3.2. Software

Two levels of software were required to enable the system to be used in an efficient manner. The flow of information between the two pieces of software is shown in Figure 2. The first level was the logic control software, which was written using the TwinCAT programmable logic control (PLC) language and was responsible for the basic communication with strain gauges and the motor drives as well as performing the basic calculations used to set the demand torque for each motor. A feedback control routine was implemented using a proportional-integral-differential (PID) control function. The control function corrects for errors between the target and the measured line loads by adjusting the demand torque to the servomotor.

The second level of software required was a graphical user interface to enable the user to configure the motor settings, record information and control tests using the system. This level of the programme is important for both efficiency and quality assurance point of view since all the test parameters are logged and can be reviewed at a later stage.

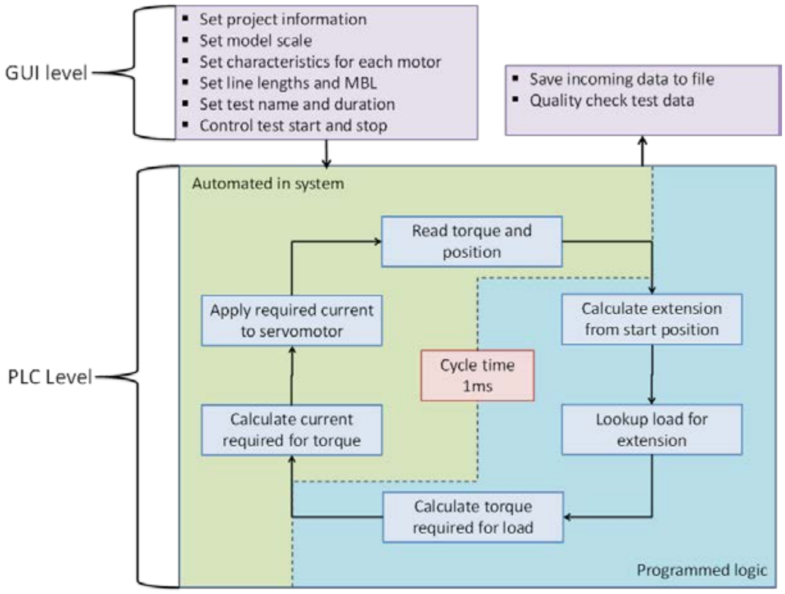

Figure 2. Information flow between control software and user interface.

4. Testing

In order to investigate whether the system is viable for use in physical models then it is important to quantify the errors within the system and consider whether these errors can be minimised or mitigated against. The results from the testing are presented in the following sections. For the purposes of discussion the smallest model scale considered in this report is 1:150, as undesirable scale effects are likely to make testing impractical at smaller scales.

4.1. Static testing

A summary of the errors identified during static testing is shown in Table 2, the following sections detail how these errors were derived.

Table 2. Static errors.

Error	Typical magnitude
Strain gauge	0.01g
Torque ripple	250g
Hysteresis	100g
PID control (static)	0.1g

4.1.1. Strain gauge noise

The use of a strain gauge in the control of the servomotors introduces noise into the control system. In order to quantify the noise in the strain gauge a zero load reading was taken. The noise in the signal is of the order of 0.15g, which equates to a load of approximately 0.5 tonnes at a scale of 1:150. This level of noise is considered appropriate for the testing to be undertaken.

4.1.2. Static servomotor load error

The mechanics of the motor and the stability of the current delivered by the motor drive lead to variations in the delivered torque, referred to as torque ripple. This torque ripple results in dynamic errors when a constant current is delivered to the motor and the shaft is rotated. Figure 3 shows this dynamic error at 5 different demand loads when no feedback correction is applied. The variation from the static load while the shaft is rotated is in excess of 200g, with the magnitude of the error not being notably more sensitive to the direction in which the shaft is turning than to the initial demand on the motor. It is expected that this sensitivity is associated with the hysteresis errors discussed below.

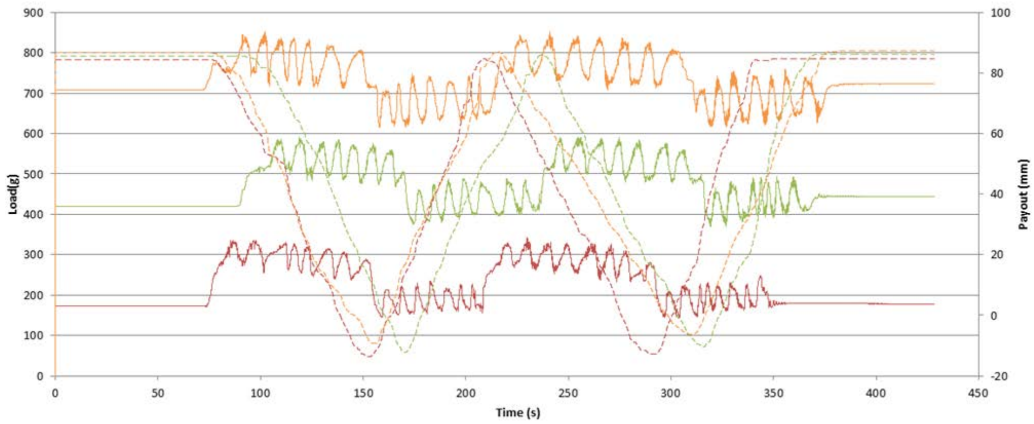

Figure 3. Line loads (solid line) and pay out (dashed line) at 25% (red), 50%(Green) and 75% (orange) demand torque.

Hysteresis in the servomotor was taken as the difference between the load up and the load down value of the actual torque developed by the motor at a given demand input. In testing this was measured using the in-line strain gauge with the line attached to a fixed point. The results of the test are shown in Figure 4. The figure shows that the motors used in testing display significant hysteresis with an average 90g difference between the load up and the load down values of the motor across the range tested. There was also a notable non linearity in the load delivered by the motor when the demand torque was being reduced, compared to when it was being increased, suggesting a poor response to direct changes in the demand to the motor.

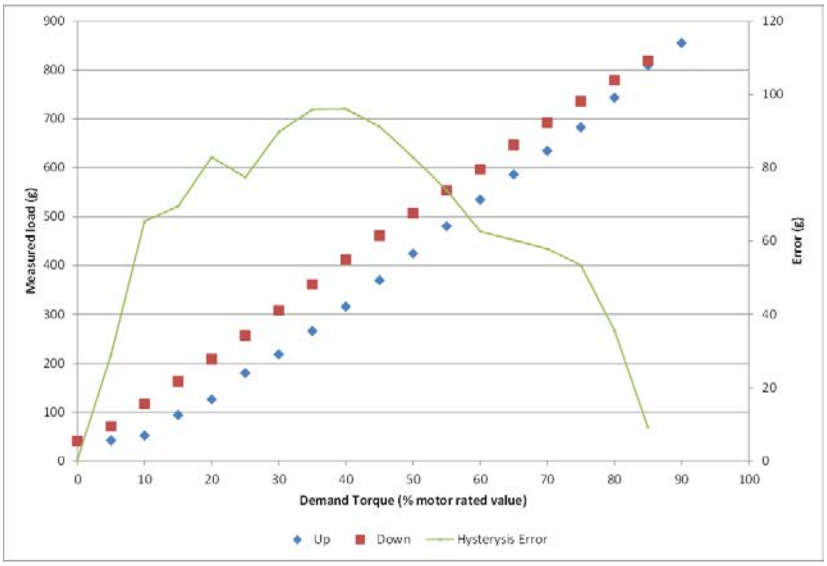

Figure 4. Change in line load with increasing and decreasing demand torque.

The magnitude of the errors identified in the static testing emphasised the need for a feedback mechanism to ensure that the correct load was being applied. The use of a PID control loop to set the current to the motor results in a lag between the measurement of an error in a target load and the correction being calculated and applied. The performance of the PID loop is dependent upon the coefficients used to condition it, if the PID loop is poorly conditioned then the motor will tend to “search” for the correct load and not behave in a stable manner. The measured loads when the system is left to settle under the control of the PID loop showed that the static load error is of the magnitude of 0.1g.

4.2. Dynamic testing

To test the dynamic response of the motor a random force was applied to three mooring line configurations and the test repeated 5 times, while the response of the system was measured. The load extension curves of the three configurations are shown in Figure 5, with one being linear, one representing a non-linear relatively elastic line and the last representing a non-linear relatively stiff line.

The load extension curve recorded during testing using the stiff spring is shown in Figure 6. The vertical section of the load extension curve, not shown the Figure 5, represents the pretension in the mooring line assuming a dynamic winch is used. The use of a dynamic winch capable of maintaining a minimum tension is an option available in the software which can be turned off. It is encouraging that the data show the measured load-extension curve forming a close cluster around the target and responding to changes in the gradient of the curve.

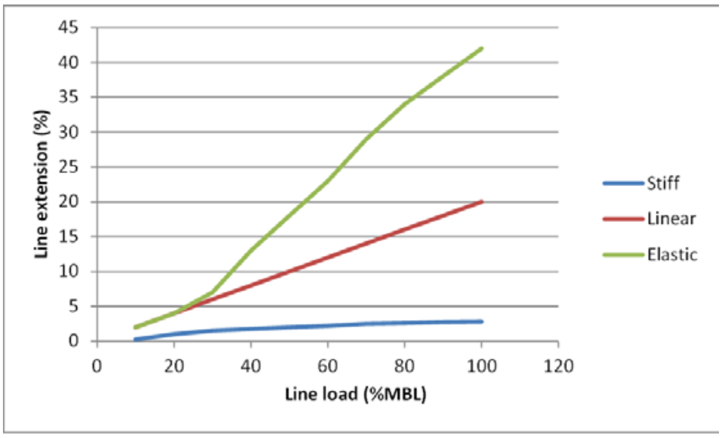

Figure 5. Load extension curves used during dynamic testing.

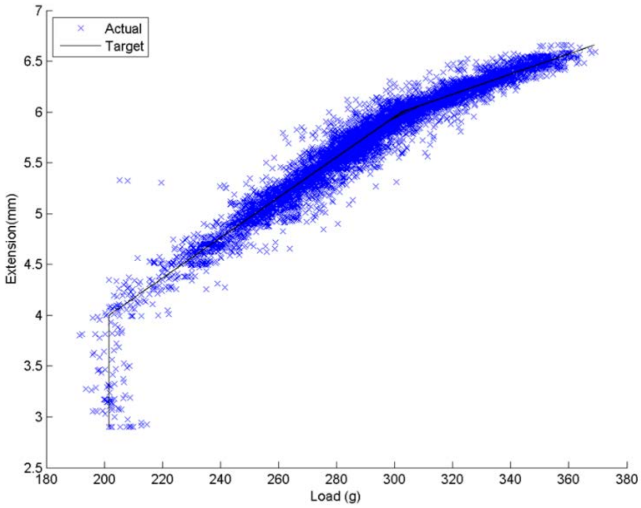

Figure 6. Target and actual load extension curve for stiff mooring line.

The root mean squared error was used to assess how well the system matched the measured and target line loads. A summary of the errors calculated using all the test results is presented in Table 3, which shows that the highest mean RMSE from the unsmoothed signal was recorded with the elastic mooring line. This may partly be the result of the torque ripple identified in the static testing, as with a very elastic mooring line the working range of the mooring line will cover a greater rotation of the servomotor shaft, meaning more corrections for the effect of torque ripple will be required compared to the stiffer lines.

Table 3. Root-mean-squared errors recorded during testing.

	Test type	All tests	Linear line	Elastic line	Stiff line
Unsmoothed	RMSE Max (g)	8.61	8.61	7.89	7.93
	RMSE Min (g)	5.81	5.81	6.61	6.41
	RMSE Mean (g)	7.13	6.93	7.76	7.08
Smoothed	RMSE Max (g)	5.12	4.35	4.2	5.12
	RMSE Min (g)	3.25	3.41	3.25	3.98
	RMSE Mean (g)	3.97	3.79	3.67	4.45

The root mean squared errors presented in Table 3 give a good indication of the overall performance of the system in replicating the required load, but in order to improve the system it is important to try and identify the source of these errors. Using the load time series, such as that presented in Figure 7, it is clear that the signal is subject to high frequency fluctuations in the line load with magnitudes up to 30g and a period of approximately 10 Hz.

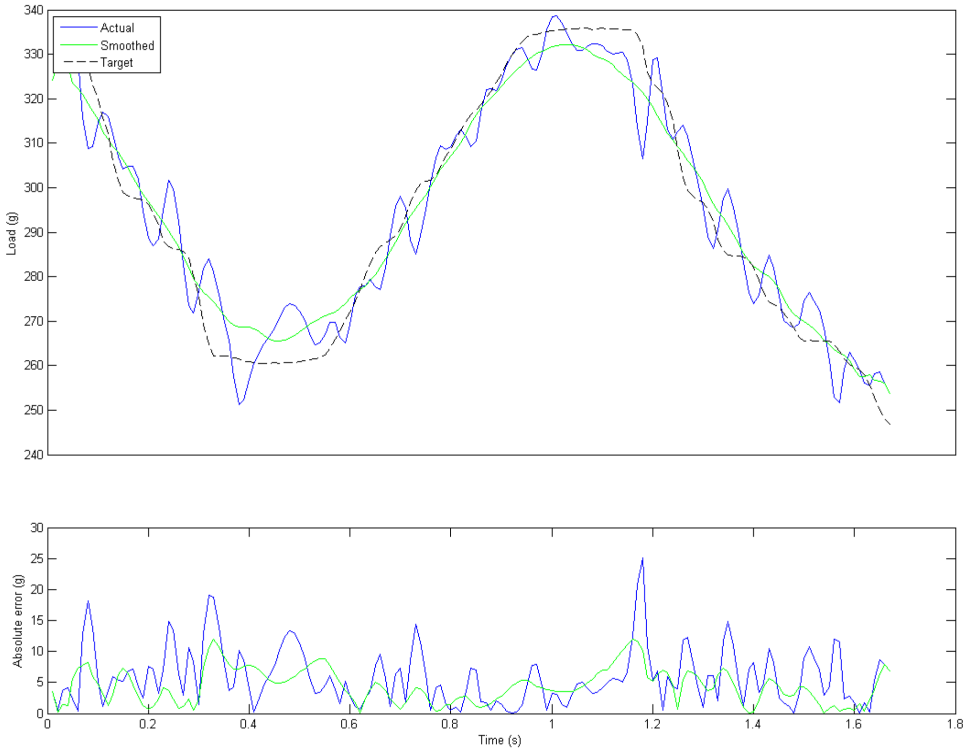

Figure 7. Time series showing target, actual and smoothed loads (top) and absolute error (bottom) during dynamic test.

These fluctuations are likely to result from both the sources of error seen in the static testing, and errors associated with the dynamic response of the feedback control. Whilst the static errors in the system are to a large extent inherent in the hardware used, there should be methods available to improve the feedback control to reduce any errors resulting from its use. Optimisation of the coefficients used in the PID control loop is a time consuming task if not automated. Therefore we intend to develop an automated optimisation routine to assist in minimising this error.

It is important to consider these high frequency fluctuations in the context of the application for which the system is intended to be used. When modelling floating structures it would be expected that the response of the structure to sudden and short lived changes in force would be somewhat dampened, with perhaps the best analogy being found in the response of structures to gusting winds. A gusting wind will apply a force across the whole structure but the overall force can be resolved to a point force acting upon the structure in much the same way as a mooring line will act. With this in mind it is possible to equate the minimum wind gust period that a structure will respond to, to the period which the fluctuations in the mooring lines are likely to have an effect in a model test. It is common practice in port and harbour design to consider, and as such in physical modelling, to use 30 or 60 second wind gusting values for models of ships as representative of the winds to which a model will respond (Thorensen 2003, OCIMF 2008). As a rough representation of this response dampening, a 0.15 second (equivalent to approximately 2 seconds at a scale of 1:150) rolling average window was passed over the measured load data and the results plotted in the time series shown in Figure 7.

As might be expected smoothing of the measured data resulted in a significant reduction in the calculated error- with the average RMS error from all the tests reducing from 7.12g to 3.97g by significantly reducing the load fluctuations observed in the time series. However in any further testing it will be important to consider whether this high frequency loading could result in a second order low frequency response from the structure, as is often identified in the motion vessels in short period waves.

5. Key findings

The key finding of this work to develop an active mooring line transducer are:

- The motors used exhibited large hysteresis error and torque ripple, therefore a feedback control loop was introduced to improve their performance;
- The system was able to correctly reproduce a load-extension curve of a mooring line with a mean error of less than 10g;
- High frequency fluctuations in the load represent the major source of error, which are expected to be significantly dampened by a floating structure, however there is a possibility that second order motions might be developed over time; and
- It is believed that errors in control feedback of motor can be largely reduced by optimising the control system.

6. Future work

The results have been sensitive to the conditioning of the PID control loop used in the control logic. By using high precision hardware it is the speed of response, accuracy and stability of this control loop which determines the ability of the system to reproduce an accurate load extension curve and as such the success of modelling a mooring line. The established methods used to set the coefficients of the PID loop gave limited success in this application. In the next stage of development it is suggested that an automated, iterative system is developed in order to optimise the PID coefficients for any given motor-pulley combination.

An important development for the next stage of testing will be the design of a system suitable for use on a physical model. It is important that if this system is to be implemented that the controls and motors can be housed in a unit small enough to use on a floating structure. The design should be waterproof and avoid excessive cabling if the servomotors are to be mounted upon a structure.

Once the model ready unit has been designed and built the system can be deployed on a physical model and comparative tests can be run against traditional coil spring mooring lines. Special consideration will be given to any longer period motions which are developed that could be the result of high frequency load fluctuations. It will also be important to assess the performance of the system over a prolonged period in order to identify any errors which may not have developed during the bench testing. If the comparative tests prove successful then the system will be ready for use on physical model studies.

If proven to work effectively for mooring lines it is possible that active force transducers could be used in other scale model applications such as the representation of non-linear fenders and in reproducing gusting wind loading.

Acknowledgments

The work described in this publication was supported by the European Community's 7th Framework Programme through the grant to the budget of the Integrating Activity HYDRALAB IV, Contract no. 261520.

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