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Safety for Gravity Loaded Axis from Standardized Requirements to Real Design of Safety

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One of the design measures adopted for the risk reduction of machinery is the de-energization of all the components of the machine to ensure a "safe stop state". Therefore, for gravity loaded axes (GLA), the risk of unintended gravity descent in the de-energized state shall be considered in the risk assessment.
In case of power failure, the gravity loaded axes are held solely by the braking or counterweight systems, which are installed in the machine. For example, due to undetected failures or poor maintenance, an inadequate response of the system after power failure is possible. Moreover, in multimodal machine tools (e.g. automatic, setup, maintenance modes) the exposition of the persons to the GLA can vary depending on the machine and the specific use, thus requiring different system design for each case. In this paper the design and optimization of a real mechanical lock system for automatic stopping of GLAs is presented in full details. First, the evaluation of the power failure detection time is discussed, with measures able to assure a proper detection functionality: the mechanical lock shall be active before the GLA begins to move. Then, considering the damping devices of the full GLA mechanism, the real magnification factor due to the dynamical response of the system under gravitational loads is presented. This factor, when determined, is useful for designing the GLA system with a proper safety factor using a simple equivalent static design. Finally, some critical aspects related to standardized requirements for GLAs are discussed based on real systems design.

Keywords: Safety of Machinery, Energy failure of machinery, Machine Design.

1. Introduction

Machinery horizontal axes have no major safety issues when they are in the de-energized state, but vertical or slant axes in the same state are subjected to gravity.
Measures to reduce the risk related to gravity loaded axes (GLA) were first considered in DGUV (2012) and introduced in safety of machinery standards in ISO 16090-1 (ISO, 2017). Results and examples of possible risk reduction measures were presents by Landi and al. (2018) during the Esrel Conference in Norway.

The 2022 revision of ISO 16090-1, dealing with safety of machining centers, bring some un-essential changes to the GLA safety measures clarifying some important aspects for counterbalance systems and pure mechanical locking systems of GLA.
So, when the machine systems are in an energized state, it can be usually assumed that the risk assessment performed by the machine builders ensures the proper safety of the machine itself through design measures and, as example, by Safety Functions as prescribed in ISO 13849-1 (ISO, 2023).

In case of energy failure or energy removal (for example during maintenance), gravity loaded axes (weight-loaded, vertical, slant axes) are held solely by, as an example:

- brakes internal or external to the motors (clamping systems),
- counterbalancing device (e.g. oil pistons),
- mechanical locking device (MLD) as the one shown in figure 1.

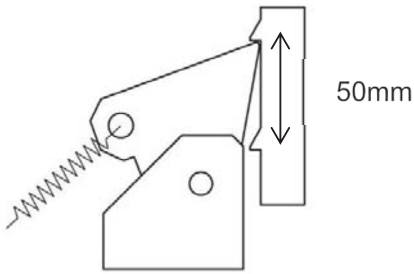

Fig. 1. Example of a MLD for GLA.

The locking device shown in Fig. 1, due to its pre-loaded spring system that is considered safe by design for ISO 13849-2 (ISO, 2012), assures a self-locking system when the energy is removed (the full description of the system will be done later in the article). During the de-energized state also the warning system could be off-line due to the complete de-energization of the machine, and no signal is possible to the operator.

In the following paragraphs first the full description of the GLA locking system designed by a European Machine Tools builder for risk reduction of GLA will be described. The requirements to be covered by design and testing of this system will be highlighted and discussed.

Then a testing procedure useful to discover the actuation time of the system in real machines will be presented. Finally, a reduced FEM model useful to recover the real dynamic loading of the GLA impacting on the MLD will be presented and discussed.

The results of the experimental and FE design stage will be discussed together with the standardized requirements at the end of the paper to highlight the possible improvement and clarification of upcoming standard based on risk reduction method of ISO 12100 (ISO, 2010).

2. Mechanical locking device main design characteristic

As anticipated in the introduction, this paper considers a case study an existing mechanical locking device. Specifically, the system consists of a rack mounted on the GLA, with teeth spaced at 50 mm, a pneumatic system connected to a safety tooth and a damping system.

In the event of axis de-energization and failure of the motor brake, the axis would descend under its own weight. Under these conditions, the system de-activates a pneumatic valve that retracts the safety tooth. Under the hypothesis of frictionless motion, the axis falls under gravity until the safety tooth engages with a rack tooth moved by a failsafe spring mechanism.

To mitigate high-impact loads and the resulting increase in the reaction force exchanged between the teeth, a damping system consisting mainly of four nylon dampers has been implemented, as shown in Figure 2.

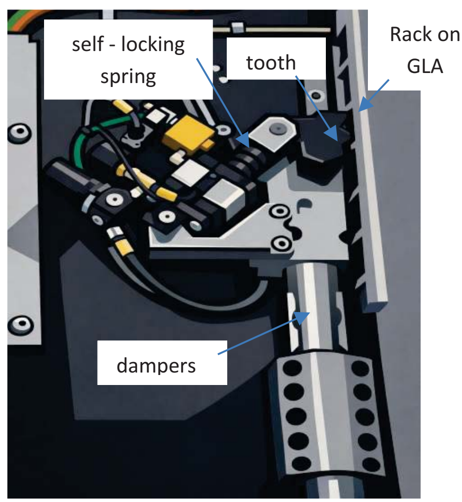

Fig. 2. Photorealistic 3D representation of the MLD of the GLA used in the case study, real design is not shown due to confidentiality

So, if the machine is de-energized in any case, the force due to pre-loaded spring system can automatically move the teeth towards the rack, locking the axis movement when a rack tooth is impacted. The GLA force is transmitted to the fixed part of the machine through the damping system that assures a proper damping of the impact force due to the descending GLA.

3. Main requirements that must be assured by design

Annex G of the ISO 16090:2022 (last revision of the standard containing requirements for GLA) states the requirements for risk reduction:

- R_1. the mechanical parts of power transmission shall be at least designed with double weight load to withstand the occurring static and dynamic stresses.
- R_2. for the self-locking device (e.g. pinhole device) it shall limit falling of the gravity loaded axis to 50 mm or less.

Those requirements are very difficult to satisfy without a deep knowledge of machine dynamic behavior and cannot be checked by simple calculation.

As an example, if we consider the free fall motion of an object, initially in quiet, due to gravity using the well-known "free falling objects formulas", it is simple to retrieve that the falling time needed for accomplish a displacement of 50 mm is 0.1 s. Also, the end velocity after this fall is approximately 1 m/s.

So, if the reaction time of the system from energy removal to full extraction of the MLD tooth is more than 0.1 s the requirement R_2 cannot be satisfied. Indeed, the only way to fully fulfill the requirement is that the safety tooth of the MLD engages the rack's tooth immediately above at the beginning of the falling.

The stopping time depends hardly on the machine initial state (GLA moving or not, energy eventually stored in the drive at energy removal, type of safety stopping devices, ….) and tests on machine have to be done to retrieve those "intervention time".

Although requirement R_1 may appear straightforward to satisfy at first glance, considering that the designer is required to adopt a safety factor of 2 to the weight of the falling axis. This force amplification factor is introduced to account for dynamic loading effects in an otherwise static design. However, according to standard mechanical engineering textbooks (e.g., Juvinall et al., 2005), the dynamic amplification factor may reach values as high as four times the static load. So, all the real machines use, as already shown on figure 2, dampers to assure a proper damping and dynamic attenuation of forces/displacements during falling of axes.

Also, this real dynamic factor should be retrieved for real safety of the GLA assured with a MLD even if the free-falling hypothesis of GLA is not a fully realistic worst-case scenario.

In the following two paragraphs, the measurement campaigns and the simulations used to verify the requirements are presented. A relatively simple finite element (FE) model is then introduced to validate the locking device, both from a standards-compliance perspective and from a mechanical design point of view.

3.1. *Intervention time of the spring system*

Before performing any modelling activity aimed at verifying the adequacy of the design requirement R_1 applied to the locking device, the responsiveness of the evaluated safety system must be assessed. In particular, in order to comply with requirement R_2, it is necessary to ensure that the safety tooth reaches its safe position before the GLA initiates the free-fall motion. To perform this kind of verification, it is necessary to take into account the entire control logic of both the motor brake and the MLD system, an overview of which is shown in Fig. 3.

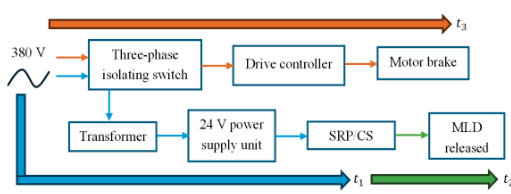

Fig. 3. Motor brake and MLD system control diagram.

When the 380 V supply voltage is lost/removed, the drive controller continues to supply the eventually stored current to the motor brake for a time interval t_3, thereby delaying the onset of free fall. At the same time, the voltage drop causes the signal commutation to the SRP/CS, triggering the extraction of the safety tooth. To ensure that the safety tooth is located in the safe position before fall begins, the following inequality must be satisfied:

$$t_1 + t_2 \leq t_3 \tag{1}$$

where:

- t_1 is the time between the 380 V voltage drop and the switching of the SRP/CS input signal;
- t_2 is the full safety tooth extraction time;
- t_3 is the time between the 380 V voltage drop and the beginning of the axis fall.

First, it should be noted that times t_1 and t_3, measured as described in the following, are subject to significant variability depending on the machine state and on the electronic components involved in the control circuit. For this reason, in accordance with Annex B of ISO 13855 (ISO, 2024), the tests were repeated ten times, and the results were evaluated from a statistical point of view. Measurements were carried out using the oscilloscope shown on Fig. 4. On the screen, the 380 V (yellow) and 24 V (green) voltage waveforms are reported; time t_1 is defined as the interval between the falling edges of the two signals.

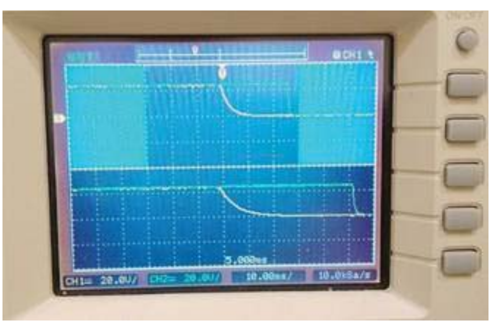

Fig. 4. Voltage waveforms: 380 V (yellow) and 24 V (green), recorded with an oscilloscope

The measurement of time t_3 was performed using a VEO-E 310L high-speed camera, with the trigger signal given by the opening of the three-phase isolating switch, capturing the instant at which the GLA starts to fall. The fall is delayed by means of a device referred to as a pulse generator, which, by exploiting the energy stored in the drive controller, is able to keep the motor brake energized and thus oppose gravity for few seconds after de-energization.

Finally, the safety tooth extraction time t_2 must be measured. Fig. 5 shows the setup of the measures conducted into a UNIPG laboratory; the camera was set to very high frame rate acquisition, 10000 Hz.

Fig. 5. Setup and equipment adopted for timing t_2 measurement.

The MLD system is actuated by a double-acting cylinder with spring releasing action. When, following the 24 V voltage drop, the solenoid valve switches, the combined action of the spring elastic reaction and the pneumatic system pushes the safety tooth into the safe position, as shown in Fig. 6(b), where the tooth moved left.

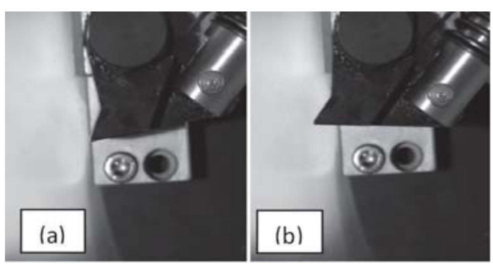

Fig. 6. Safety tooth in rest position (a), safe position (b).

Since the machine operates within a supply pressure range from 4 to 8 bar, the system was tested, in a controlled environment, starting from the minimum pressure up to the maximum one, with 1-bar increments. Once again, ten tests were carried out for each condition. A second set of measurements was then performed under the assumption that, after the loss of electrical power, the pneumatic system is no longer able to supply pressure to the cylinder, thus leaving the spring as the only source driving the safety tooth extraction. Tables 1 and 2 report the main results of the measurements performed.

Table 1. Main results of t_1 measures.

Mean μ	Standard Deviation θ	$\mu + 3\theta$
(ms)	(ms)	(ms)
86	46	224

Table 2. Main results of t_2 measures.

Pressure	Spring&Pneumatic $\mu + 3\theta$	Only Spring $\mu + 3\theta$
(bar)	(ms)	(ms)
4	54.6	69.1
5	56.7	72.1
6	58.7	80.4
7	60.1	84.7
8	60.1	89.7

Measures of time t_3 is not simple to be done and the results depend mainly on machine state at energy removal. A lot of different tests were done considering different conditions.

During those tests, whose can be done only if the machine PLc is modified to prevent the automatic removal of pulse generation of drives and de-activating all others safety devices such as the motor brake automatic braking, the minimum time t_3 retrieved during those tests is around 870ms. So even when the most conservative values are adopted (224 ms + 89.7 ms), inequality (1) is satisfied with a safety margin greater than 2.5. *We can state that the tooth is in safe position before the GLA begins to move.*

As can be observed, the standard deviation values in Table 2 have been omitted, since they never exceed 1 ms and they are negligible. This omission is therefore not critical, as it indicates that the measurements of t_2 are highly repeatable and the results are reliable.

In contrast, the experimental results for t_1 and t_3 exhibit significantly higher variability and instability. The standard deviation of t_1 is 46 ms, while the value of t_3 strongly depends on the operating conditions (e.g., axis motion, direction, speed, and type of machining process), which affect the electrical demand and make a reliable and unique characterization impractical. So, we decided to take the minimum value retrieved for all the tests because the single test cannot be evaluated as a single test anyway.

For these reasons, a safety factor greater than 2.5, although it may appear excessive at first glance, is in fact necessary to ensure that inequality (1) is always satisfied.

3.2. *FE model used for force retrieval*

As previously discussed, the most common method for the design of the MLD is carried out by introducing a dynamic amplification factor to the GLA weight, penalizing the static design of the MLD elements. Although requirement R_1 specifies a minimum factor of 2, it only refers to power transmission parts, excluding structural components which must be properly analyzed for the specific case study. Therefore, the actual stresses acting on the components may be significantly underestimated, since the impact event is governed by complex phenomena that are difficult to address through simplified analytical approaches. Indeed, the dynamic loads generated during the engagement of the MLD depend on several factors, including the initial motion state of the GLA, the residual energy stored in the transmission system at power removal, the stiffness and damping of the mechanical components which amortize the impact, and the design characteristics of the mass loading the mechanism. In this context, the dynamic amplification factor adopted during the design phase should implicitly account for the combined effects of all these contributions.

It is therefore evident that a safety component such as the one under investigation needs an appropriate dynamic analysis to ensure the real amplification factor necessary for the design. Since dynamic amplification strongly depends on the specific operating conditions of the machine, it is advisable to adopt numerical methods capable of representing several scenarios of abrupt impact of the GLA rack against the safety tooth that arrests the free fall.

The analysis was carried out using a well-known FE code (Ansys), in which a simplified equivalent model of the entire system was developed. When contact between the teeth is established, the load is transferred to the damping system (see Fig. 2). This component, as shown in Fig. 7b, consists of four dampers and two spacers, aligned along the vertical direction and mounted in series. A metallic enclosure is used to house the elements and to fix the damper assembly to the machine frame. When a load is applied to the top of the damping system, the elements undergo compressive deformation along their axis, depending on their material properties and geometry. The characteristic force–displacement curves is provided by compression tests conducted by the manufacturer of the machine and is reported in Fig. 8 a). The spacer curve in Fig. 8 b) is retrieved by a non-linear compression finite element analysis, using the real geometry of the spacer and the material that is equal to the damper one. As can be observed, the nonlinear behavior of these elements cannot be properly captured using a conventional linear spring. Therefore, six elastoplastic elements were introduced into the model as COMBIN39 spring elements (see Fig. 7a), each defined by its specific force–displacement characteristic.

The GLA was modeled as a mass body with the same mass as the vertical axis of the machine under investigation (as an example 800 kg) and was applied at the top of the damping system.

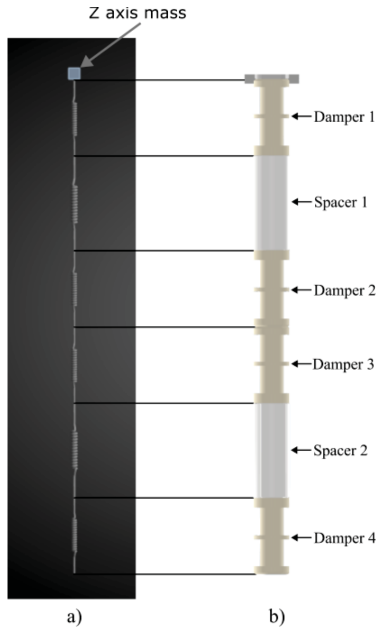

Fig. 7. Damping system schematic: a) ANSYS Mechanical simulation; b) real configuration.

Simulations were then performed considering different initial velocity values, ranging from 0 m/s up to approximately 1 m/s, which represents the maximum achievable impact velocity, as discussed previously in Section 3.

The finite element analyses made it possible to obtain, for each simulated impact velocity, the maximum reaction force in the falling direction (machine z-axis), the total deformation of the damping system up to complete deceleration, and the corresponding dynamic amplification factor.

The latter was computed as the ratio between the maximum reaction force along the vertical direction and the static weight force of the machine z-axis.

As reported in Table 3, in this specific example the values of the dynamic amplification factor largely exceed the limit prescribed by requirement R_1, reaching a maximum value of 3.78, while the total deformation of the damping system reaches 31.98 mm. This value shall be added to the free-fall descent in order to compute the total displacement of the GLA before stopping.

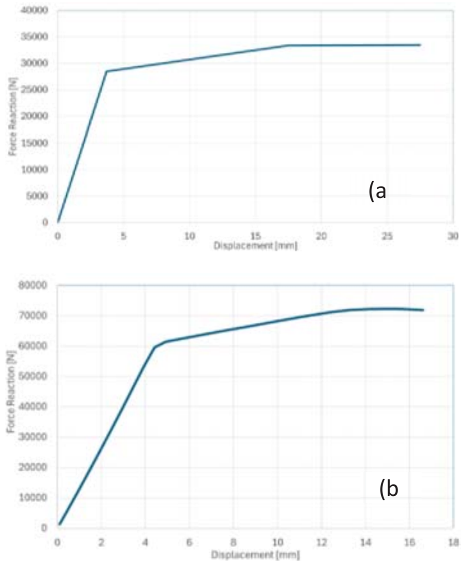

Fig. 8. Force–displacement curve: a) dampers; b) spacers.

Table 3. Simulation results of the damping system with m = 800 kg.

Initial velocity [mm/s]	Maximum constraint reaction in Z [N]	Maximum total deformation [mm]	Dynamic amplification factor
1000	29632,03	31,98	3,78
750	28920,38	23,84	3,69
557	28496,05	19,04	3,63
541.45	28000,69	18,71	3,57
500	26687,05	17,83	3,40
250	19396,47	12,96	2,47
0	15688,67	10,48	2,00

This simple elastic-plastic model is able to retrieve the damped oscillation of the GLA and to take into account the plastic behavior of, as an example, the dampers if a reaction force of 28500 N is exceeded as one can see from curve Fig. 8 a).

Indeed, imposing a maximum distance between two consecutive teeth in the rank lower than 50 mm, as clearly specified by the standard, does not guarantee the R_2 requirement. Falling distance greatly depends on characteristics of the damping system adopted to reduce the dynamic effect, depending on the compression of the dampers and the spacers. Analyzing the dynamic amplification factors listed in Table 3 some important considerations are possible. First, the values are always equal or greater of 2, this means that the requirement suggested by the standard underestimates the real loads which could be much superior. Secondly, the amplification factor is not proportional to the velocity of the GLA, and it depends also on the mechanical characteristics of the damper. So, the design calculation should be based on maximum force that can be sustained by the damper and dynamical deformation of it. Furthermore, the use of a damping system is essential. Indeed, without the damping effect generated by these elements, the reaction force would reach values that could easily compromise the mechanical resistance of the locking system parts. Table 3 highlights how the damping system contributes should be carefully analyzed for both reducing the dynamic effect of the impact while maintaining a limited fall of the GLA. In this way a conscious design of the MLD can be performed and the risks for the operators can be mitigated.

4. Conclusion

This paper analyzes a gravity-loaded axis of a machine tool according to the criteria defined in ISO 16090-1:2022. Since de-energization of the machine may lead to the uncontrolled falling of the axis, a locking system is introduced and investigated with respect to both its response time and the dynamic overload experienced during engagement.

Although it is not possible to inspect the effectiveness of the solution for all possible machines configurations, it has been shown that the MLD reaches its safe state before the onset of free fall. Finite element analyses further show that, even when a damping system is adopted, the dynamic amplification factor exceeds two in the worst-case condition of free falling without frictional effects. This result suggests the requirement R_1 should be followed, for transmission components, by a detailed dynamic analysis, required for the proper design of the structural components of the safety device.

The maximum force sustained by the damping system can be used as a guide for the design of components of transmission system if velocity of GLA is not neglectable. The analysis presented in this article is a worst-case analysis: free falling of GLA. The realistic falling velocity of the system will be lower than expected, a significant amount of energy will be adsorbed by elastic-plastic deformation of damping system.

Moreover, it is shown that the requirement R_2 and its validation in real design is not properly defined by the ISO standard: for any stopping device (locking but also usual clamping device), the validation of the 50mm of maximum axis travel is not possible to be guaranteed by any manufacturer. As an example, useful dumpers deformation has not to be considered in requirement because without deformation the probability of mechanical breaking components due to very high impact forces is expected. In machine design of a GLA a lot of unknowns and conditions not fully possible to be defined (damping due to components in the transmission system, amount of friction, …). S0, it is shown that the damping system represents a critical component for the GLA, as it strongly affects both requirements prescribed by the standard. Achieving a limited falling distance of the axis requires accepting higher dynamic amplification factors, and vice versa.

Authors of the paper encourage a revision of the R_2 to a more comprehensible requirement as expressed in other standards, such as the packaging machinery harmonized standard. If the requirement for maximum falling distance will be changed to a requirement on maximum spacing between rack teeth (as an example 50 mm as shown in Fig. 1) it will be fully understandable and clearer also for surveillance of the market and end user of the machine. The requirement of the theorical maximum falling distance less than 50mm is obtainable only by removing the dumping mechanism of GLA and in this case the impact forces will rise to an undesired theoretically unlimited value.

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