Development and design of linear drive and methodology of linear drive testing

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Abstract

This article is devoted to developing and testing of belt-driven linear drive system. The main development target will be appointed and real-executed design will be explained. The linear axis configurations will be described and compared with commercially produced products. The main comparative parameters will be appointed and measuring methods will be outlined. The results of experiments will be discussed and competitiveness of designed drive system will be analysed.

Key words: Linear positioner, timing-belt, linear drive system, testing of linear drives, experimental verification

1. Design Part

1.1. Introduction

Developing process of the modular linear drive system was intended as a design exercise for attainment of technical design that could be compared with actual industrial competitive products. In next phase, the realized prototype was intended as a base for research of attributes and behaviour of simple linear drives at department of Design and Machine Parts at CTU. Produced prototype was tested by basic experimental methodics for verification of supposed operating parameters.

1.2 Choice of drive system type

Main load force – transmitting elements of linear drive is their own drive element and linear guide rail system, which is used for positioning of loaded carriage.

Linear actuators are made in wide product line-up of drive element systems. From light system driven by narrow timing belt, through wide timing belts with cord elements to heavy duty linear drives driven by trapezoidal and ball screw for biggest loading capacities. Choice of drive element is crucial on loading capacity of whole linear drive system (in direction of positioning), achievable dynamic parameters and two-sided stiffness of system.

Another main parameters of the linear drive are permissible force and torque loading capacities of linear guide system. Choice of this guide depends on the requested loading capacity, can be realized as a lightweight slider linear bearing, gothic and vedge cam roller systems for medium loadings and heavy duty complex roller guides and ball guide way systems.

For prototyping purposes, timing belt type AT (pitch 5 mm, width 16 mm) drive was chosen, according to its simplicity and low financial demands.

Conceptual design of linear positioner (hereinafter “LP”) is according to trends in linear positioning technology, it consists of aluminium strut profile middle beam and end-bearing housings from steel side plates and addition cover. In end-housings are placed shafts, which are connected with timing pulleys by shaft keys. Side plates are connected with middle profile by standard slider-nuts in T-grooves of profile and countersunk screw with conical head. Timing belt is connected with carriage with standard clamping plates.

Calculated durability of LP was determined as equivalent of two years of continuous operation with six hours a day shifts. This is usual value from industrial practice and can be determined as amount of 3200 hours.[1]

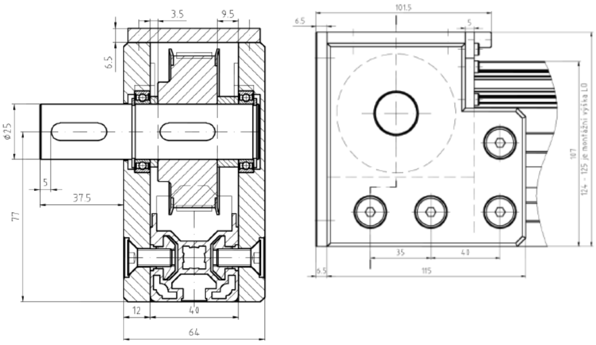

Fig. 1. *Detailed design of pulley housing.[1]*

1.2 Configuration of LP for chosen capacities

Three classes of LP were proposed for further detailing design. These classes were specified by usable loading capacities of used linear guides and size of timing pulleys and were designed as product which can be competitive to industrially manufactured linear drives. Loading capacities were based of market research.

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1.2.1 Lightweight class

Light class was based on linear slider guide IGUS DryLin Type N. Several sizes and types of sliders were considered. Carriage based on tandem of sliders N40x50 was further developed. Chosen slider system was analysed by DryLin Expert 2.0 design application. For many acceleration levels were calculated permissible loading values which fulfilled above mentioned durability.

Complete design of prototype slider linear drive was suited for this linear guide and therefore it's the most compact one.[1,4]

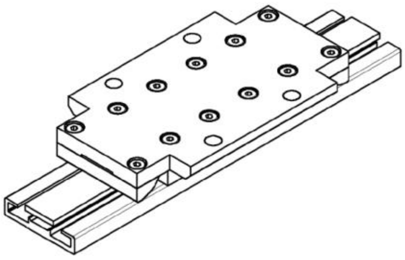

Fig. 2. *Lightweight carriage and slider rail.*

1.2.2 Medium class

Medium class was based on gothic cam rollers linear guide and carriage assembly of own design with parrarel positioning of rolls axis. Gothic cam rollers with steel rod guidelines made by TEA Technik AV06 and MATIS GD6 were considered at design variants. Design process was similar to slider class. Carriages were designed to be able to transfer maximal permissible force and torque loads specified by manufacturer of cam rollers.[1]

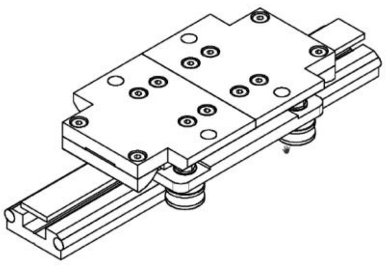

Fig. 3. *Cam roller carriage.[1]*

1.2.3 Heavy duty class

Heavy duty class LP was based on using of a compact high loadable linear guide and its inclusion to proposal. This variant is the most space-intensive, but offers the most loadable option in every force/torque direction. Franke FDA 25 linear guideline was chosen for this variant. Mechanical design was directed to as few as possible amount of mechanical modification of purchased prototype parts. Custom parts were added and fasteners were changed, the mounting points of Franke carriage were retained. The permissible loads calculations were made for many levels of dynamic loads and positioning velocities. Rollers inside the carriage are oriented to a cross arrangement and because of this, the force and torque loads are projected to calculations by unique way and calculations have to be repeated to specific applications. [1]

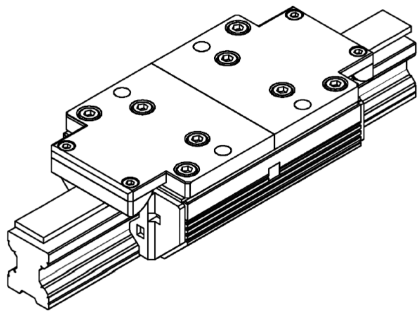

Fig. 4. *Modified Franke carriage.[1]*

1.3 Realized prototype drive

For further research was realized and manufactured heavy duty prototype of LP. Preloading mechanism timing belt drive was realized by a pushing bolt placed in a lower central threaded hole of middle strut profile. When this screw is loosed, its head is pushing to an end-housing body strut profile and preload force in belt is generated. On preloading side of linear drive were doubled joints between middle beam and side plates, in order to prevent tilting of side plates caused by forces effects during the preloading procedure.[1]

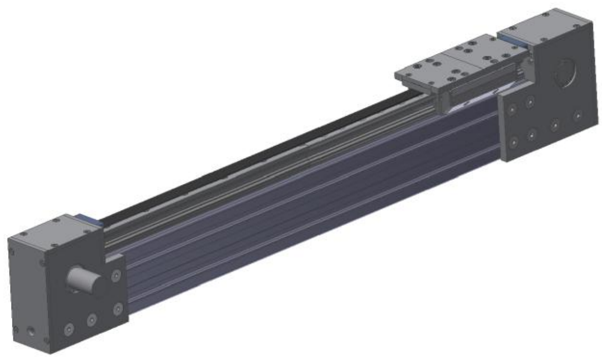

Fig. 5. *Realized prototype model.[1]*

2. Calculations Part

For complete prototype of LP layout was necessary to appoint and evaluate appropriate mathematical base of design. It was necessary to apply mathematical model of timing belt inner forces behaviour, which is considering dynamics of positioning and actual position of carriage and positioned loading. Simple computational application based on MS Excel was programmed. The result of calculation was represented by checks of maximal permissible loads according to acceleration of system and graph of velocity ramp of positioning.

2.1 Mathematical base of design

Elementary part of design process was complete force behaving calculation. Influences of all possible forces, properties of drive electromotor and up to 2° gearing system were assumed. Result of this calculation was force effects exerted on supporting structure of linear drive (shafts, ball bearings, preload bolt joints, buckling and bending of middle beam…). [1,2,3]

Behaving of inner forces of timing belt drive is shown in the below diagrams:

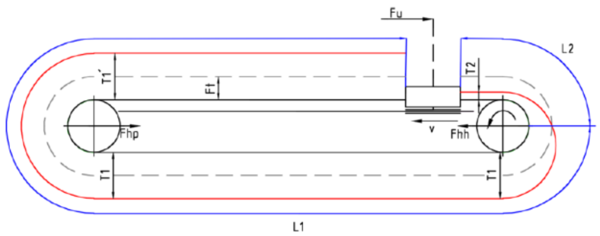

Fig. 6. *Positioning in direction from driven pulley.[1]*

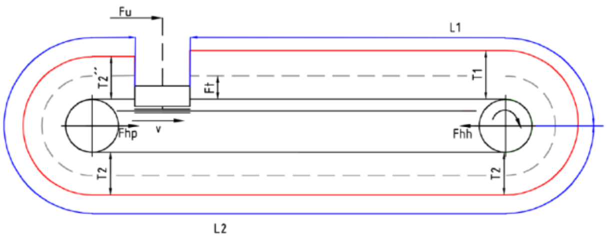

Fig. 7. *Positioning in direction to driven pulley.[1]*

Force effects exerted on the shafts are expressed as the sum of forces in upper and lower spans at particular pulley. The course of the force in belt span is linear on the stroke segment, and is defined by actual length of slack and tight spans. [1]

2.2 FEM Analysis of components

Loaded parts of prototype underwent basic FEM Analysis for calculated operational loads. This analysis can be divided to two main chapters. In the first computational instance was side plate sub-assembly checked, in the second instance was checked stiffness of prototype middle beam strut profile. FEM Analysis was made in Simulia Abaqus 6.12 Software. [1]

2.2.1 Side plates FEM Analysis

Simplified sub-assembly model was created. In this model is side plate mounted to middle beam with T-slider nuts and preloaded countersunk bolts. This sub-assembly was loaded by maximal calculated value of ball bearing reaction force. This force was calculated as a half of maximal driven shaft force (Approx. 1,26 of maximal drive shaft force).

Contact between components of subassembly was defined as Surface-Surface hard contact with friction coefficient (Anodised aluminium/Steel). Subassembly was mounted by fixing of reference surfaces defined on the inner side of simplified middle beam profile. Subassembly was loaded by ball-bearing loading force (2428N) distributed to half-cylindrical segment of surface.

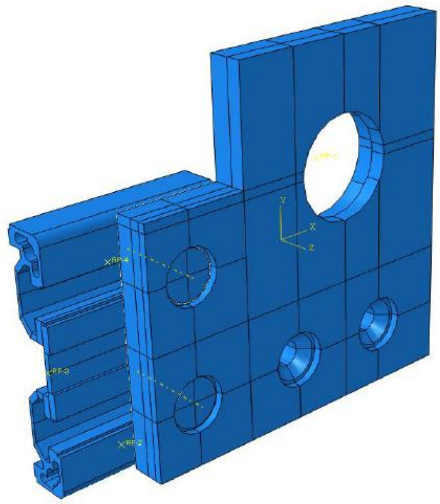

Fig. 8. *Side plate FEM Simplified sub-assembly.[1]*

Preload bolt joints were realized as a detailed geometry of bolts and T-nuts. Contact between bolt and nut was generated by the Tie constraint, bolt was preloaded by 4255 N in the middle of its body.

Every part of assembly was meshed properly, according to its geometrical complexity. Middle beam and screws were meshed by HEX Type elements, side plates and T-nuts were meshed by TET Type elements. Task was executed in two steps. In a first step were applied boundary conditions and bolts preloading forces, in a second step was applied loading force.

The value of the Von Mises stress didn't exceed the level of 80Mpa after the extreme values were eliminated from the results visualisation. At Illustration shown below was interval of displayed stress limited from 5 MPa to 80 MPa. Extreme value of contact press didn't exceed the 207 MPa. [1]

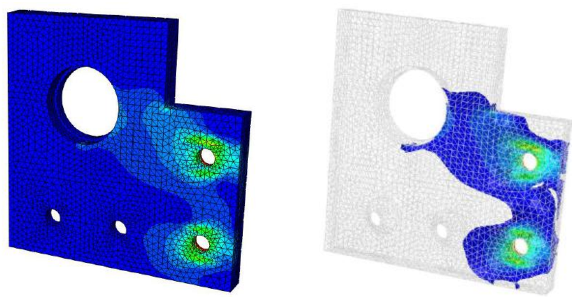

Fig. 9. *Results visualisation – Side-plates. [1]*

2.2.2 Middle profile FEM Analysis

Comparative FEM analysis of middle beam strut profile of bending was made. The influence on stiffness of sub-assembly by adding a linear guide rail was described. Sub-Assembly was modelled as a symmetrical task and detailed geometry was used. [1]

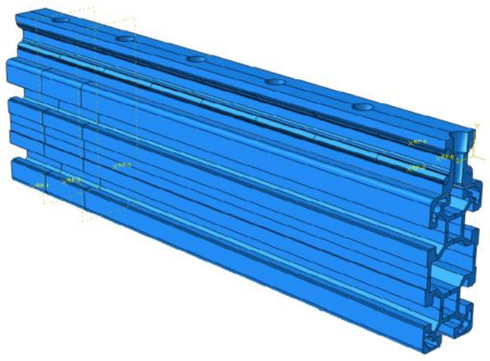

Fig. 10. *Simplified FEM sub-assembly of middle beam.[1]*

The rail was mounted to middle beam by Tie constraint. The boundary conditions were applied on the end bottom and sides of strut profile, where the reference surfaces were fixed. The loading force was distributed on contact surface of stainless steel rods, which are in contact with cam-rollers of the carriage. Sub-Assembly was loaded by equivalent of vertical force 1904N, which is maximal allowed dynamic loading, according to calculations of durability time. [1]

Tab 1. *Results comparison [1]*

Task Type	S bend max MISES (MPa)	Umax (mm)
Analytical approach	15,6	0,129
FEM without FDA Rail	20,3	0,165
FEM with FDA Rail	21,2	0,029

The main target of this analysis was to determine increase of stiffness after mounting of linear guideline. The computational case was divided into a two subcases. In first case, linear guideway is turned off and only the middle beam is loaded. This task was compared to simple calculation from strut profile manufacturer's handbook. The result of the analysis is shown in table above, the deformation of sub-assembly with rail decreased by more than 80% compared to deformation of alone middle beam. [1]

3. Experimental Part

Measuring methods for the comparative parameters were developed. These methods allow repeatable experiments in laboratory conditions. Methods required precise measurement of appointed characteristically values. These values were necessary for determination of real operational parameters of linear drive. These methods will be in this chapter described and explained individually.[1]

The main comparative parameters were appointed as:

a) Maximal load capacity of LP [N]
b) Two-sided stiffness of LP [N/mm]
c) Accuracy of positioning [mm]
d) Efficiency of linear drive [-]

3.1 Accuracy of positioning

Accuracy of linear positioning is a crucial attribute in real industrial service and application of the linear drive. The accuracy depends not only on the linear positioner itself, but on every component inserted into the drive system. In our case was necessary to considerate the influence of stiffness and deformation of every element in the drive system (clutches, torque sensor, drive shafts, etc.). Positioning error was thereafter calculated as the difference between theoretically expected and the experimentally measured real values of strokes.[1]

3.2 Efficiency of linear drive

The Efficiency of the linear positioner can be calculated as ratio of input and output mechanical work. These works were determined on the experimentally measured data. Input mechanical work was calculated as torque of the drive electromotor multiplied by its angular rotation. Output mechanical work was calculated similarly as the action force of platform multiplied by its velocity value. These values were divided and this quotient was the requested efficiency. [1]

$$\eta_{\ system} = \frac{P_{out}}{P_{in}} = \frac{W_{out}}{P_{in}} = \frac{F_u.s}{T_m.\alpha} = \frac{F_u.s}{T_m.\alpha.r_m.2.\pi} \quad (1)$$

3.3 Experimental Stand

The experimental stand was designed as energetically open system without energy recuperation. This solution is suitable for our relatively simple experiment. Individual sensors have been built into experimental system. The Screw-jack system ZIMM Z-5-SL with maximal stroke of 350 mm and maximal generated force of 5kN was chosen as the loading system. This screw-jack was used for experimental testing of load capacity and two-side stiffness of the linear drive. This screw-jack was driven by servo-motor FESTO EMMS-AS-70-S-RM. [1]

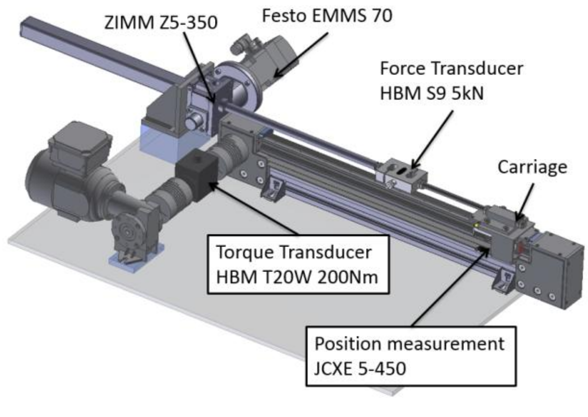

Fig. 11. *Experimental stand visualisation [1]*

Tab 2. *Used sensors [1]*

Measured Value	**Sensor**
Torque at drive shaft of LP	HBM T20WN
Action force of carriage	HBM S9
Position of carriage	JCXE 1 – 450mm
Safety end-switches	SAIP-CLS-111

3.4 Experiment

3.4.1 Loading capacity of LP

At first phase of experiment was necessary to find the accurate value of loading force which can be transmitted by the platform of the linear drive. The external loading force was generated by screw-jack while the drive shaft of linear positioner was fully locked. The size of the force was increased from zero to maximal transmittable load in steps of 50 N. After skipping the belt through locked pulley was the experimental phase finished and the maximal stable transmissible force was recorded and plotted to the graph. [1]

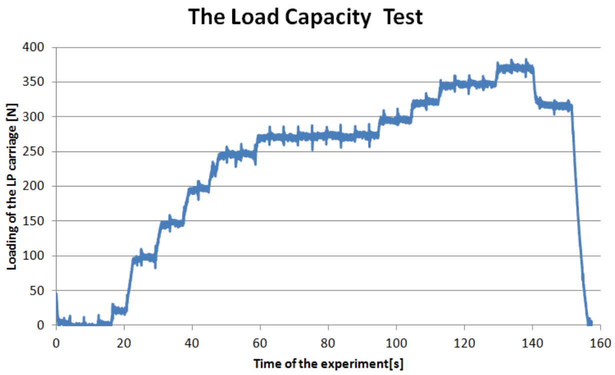

Fig. 11. *Loading capacity test [1]*

According to the graph (Fig.11) we could say the maximal loading capacity of tested LP in the direction of positioning is equal to 350 N. When the external load reaches the level of 375 N, the deformation of position significantly increases. In this point the linear positioner wasn't able to positioning under loading and the reaction force had been reduced to zero. [1]

Tab 3. *Load capacities and prices comparison[1]*

	Fu max	**Price**
Prototype LP	350 N	600 €
Commercial LP A	360 N	1778 €
Commercial LP B	350 N	1548 €

3.4.1 Two-sided stiffness of LP

In previous part was determined the maximal loading force to value of 300 N. Twelve suitable stroke positions were chosen for measuring the two-sided stiffness. This stiffness calculation was based on the immediate change of platform position under the influence of the loading force.

In the graph on the Fig.12 can be seen the influence of the preloading force to the stiffness of LP during the first measuring cycle. On the hysteresis curves area of decreasing of stiffness is in the loading level about 180 N. This area indicates the compensation of the backslash in the timing belt mechanism (backslash between the teeth of belt and the drive pulley). This phenomenon does not occur in the first measurement cycle after pre-loading of the positioner. The difference between position of the platform at start of the loading sequence and the position after relief from loading sequence is obtained positioning error.[1]

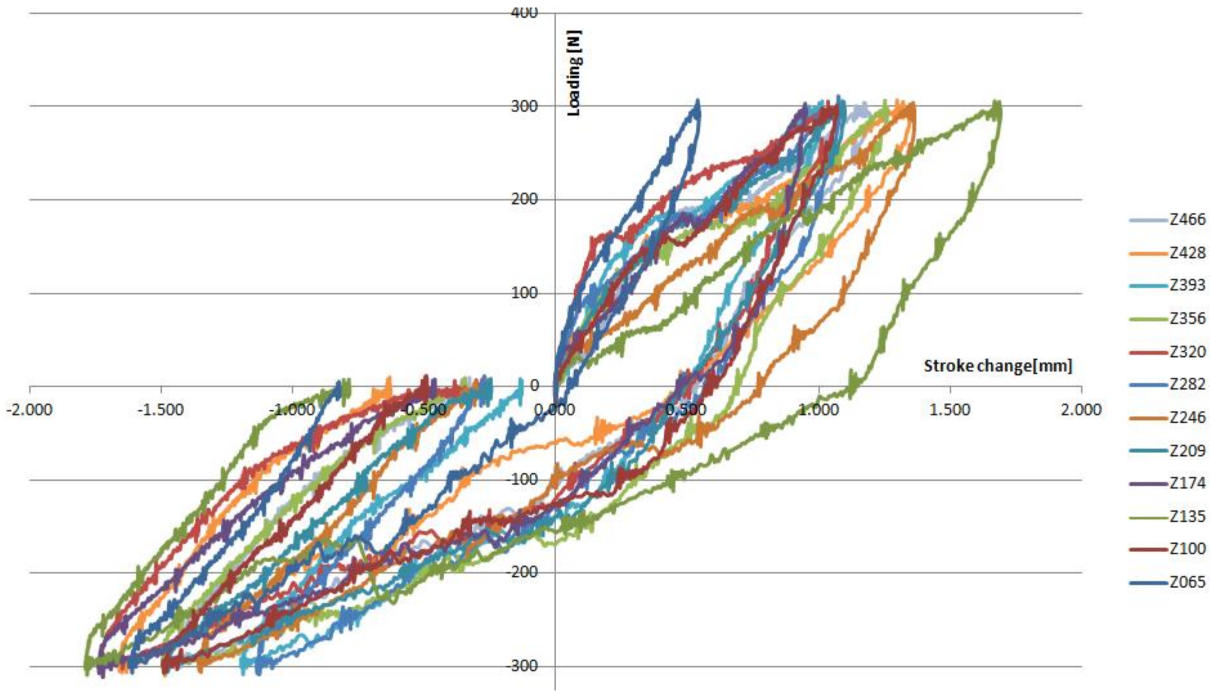

Fig. 12. *Graph of stroke position chase under the loading force [1]*

4. Conclusion

The design process of linear positioner prototype was described and basic used calculations and FEM analyses were outlined.

The experiment methodology was shown on real executed prototype experiment and the results were evaluated.

The achieved maximal load capacity of prototype was compared at the Tab.3 with commercially manufactured Hi-end linear positioners of comparable type.

In despite of the malfunction of preloading mechanism of the LP, when the thread in aluminium middle beam wasn't able to bring calculated preloading force, which was planned to meet the target of loading capacity of LP around 600N, the LP met the common loading capacity level of commercial products.

The weak points of structural design were appointed and areas of further development in order to improve operational parameters were recommended.

Nomenclature

LP	Linear Positioner	
T_1	Tight span belt force	[N]
T_2	Slack span belt force	[N]
$T_{1'}$	T1 reduced by inertia force of pulley	[N]
$T_{2''}$	T2 elevated by inertia force of pulley	[N]
F_w	Active positioning force of LP	[N]
F_{hh}	Force effects applied on drive pulley shaft	[N]
F_{hp}	Force effects applied on driven pulley shaft	[N]
L_1	Belt tight span lenght	[N]
L_2	Belt slack span lenght	[N]
$S_{bend\ max}$	Maximal bending stress	[MPa]
U_{max}	Maximal deformation	[mm]
η_{system}	Efficiency of linear positioner	[-]
P_{out}	Power at the system output	[W]
P_{in}	Power at the system input	[W]
W_{ut}	Mechanical work at the system output	[J]
W_{in}	Mechanical work at the system input	[J]
Fu	Force generated by LP in the direction of positioning	[N]
s	Stroke of LP	[mm]
T_m	Torque at the drive shaft of LP	[N.mm]
α	Revolution angle of drive shaft	[rad]
r_m	- Revolutions of drive shaft	[rev]

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