MODAL ANALYSIS FOR THE DRIVER STATION OF A BELT CONVEYOR

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ABSTRACT: The article deals with the use and applicability of modal analysis methods in ~~of~~ assessing the durability indicators and exploitation stability for the mechanical structures and systems in driver groups of a belt-conveyor. The methods of modal analysis gave the possibility to analyse the behaviour and parameters of a dynamic system through pure static research methods and techniques. Through this methodical treatment, some structural changes and reactive support are researched, along with their influence on the dynamic behavior of the support frames and driver groups from the driver stations of a belt conveyor. Dynamic models for the support frame of the driver group and station are developed estimating the stability of the frame towards bending vibrations. From that particular modelling, diagrams and result in a tabular form are presented.

KEYWORDS: belt conveyor, driver group frame, dynamic model, modal analysis.

МОДАЛЕН АНАЛИЗ НА ЗАДВИЖВАЩАТА СТАНЦИЯ ЗА ГУМЕНО-ЛЕНТОВ ТРАНСПОРТЬОР

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РЕЗЮМЕ: В работата се разглежда използването и приложимостта на методите на модалния анализ за оценка на показателите на дълготрайността и експлоатационната устойчивост за механичните структури и системи на задвижващите групи на гумено-лентов транспортьор. Методите на модалния анализ дават възможност да се анализират поведението и показателите на динамична система при използване на чисто статични изследователски методи и техники. При тази постановка са изследвани някои структурни взаимодействия, както и структурната реакция, и тяхното отражение върху поведението на динамичната система на рамите и задвижващите групи от задвижващата система на гумено-лентов транспортьор. Разработени са динамични модели на рамата на задвижващата станция, като подробно е изследван модел за оценка на устойчивостта на рамата спрямо огъващи трептения, като на база на този модел са изведени резултати в графичен и табличен вид.

КЛЮЧОВИ ДУМИ: гумено-лентов транспортьор, рама на задвижваща група, динамичен модел, модален анализ

INTRODUCTION

This study examines the dynamics of a rubber belt conveyor driver group, along with an evaluation of the driver frame performance. Rubber belt conveyors (RBCs) are experiencing a "renaissance" in contemporary technological schemes. The possibility of shortening the transport and conveyance distance of mining dump trucks by installing inclined conveyors through the non-working board of the mine or through inclined transport construction, can shorten the transportation path of dump trucks by up to 5-6 times (Shamsi, 2021). The frame of the driver group performs a supporting function and connection between the units forming the driver group, namely: reducer, clutch, turbo clutch, and electric motor. The configuration preferred in mining industry is using a bevel gear cylindrical reducer, couplings, and electric motor due to the longitudinal expansion and release of the frame by a rear reactive support. The reducer is suspended on the drive drum shaft by two couplings, as shown in Figure 2, and the load from the driver group is taken by the two driver drum bearings. The reactive support takes part of the load from the driver group and its frame and the force from the reactive moment when driving the conveyor.

Regime changes in conveyor loading are directly reflected on the frame and its reactive support. The stiffness of the frame corresponds to its dynamic deformations, which is a prerequisite for its modal study.

THEORY AND METHODOLOGY

The main purpose of the modal analysis is to evaluate the mutual influence of the driver group and the transport nodes in the frequency area or, in engineering terms, compare forced and natural frequencies. Belt conveyors have many installed parts and assemblies, and even in an established mode, they create a prerequisite for causing dynamic effects and phenomena. Four main causes can be considered together with their respective analysis methods when the dynamic effects occur:

– elastic wave propagation and theory of elastic sound wave propagation in different media (Lodewiks 1996, Daijie He 2016, Bortnowski 2021, Bortnowski 2022, Hou 2008);

– propagation of tension waves in the direction of the axis of the belt and the direction of movement of the conveyor (Harrison 1983, Harrison 1986);

– propagation of transverse waves along the axis of the conveyor (Harrison);

– propagation of longitudinal waves caused by the movement of the tension system and the elasticity of the belt as a discrete mechanical system (Harrison 1983, Damyanov 2008, Damyanov 2009a, Damyanov 2009b).

The four examined methodologies have theoretical borrowings and in some aspects seem identical, but the results are different, which allows them to be separated for the purposes of comparative analysis. The parameters of the considered belt fabric are given in Table. 1.

Table 1. *Indicators of the belt and its carcass*

	belt properties		steel rope properties		
μ=(m'·B)/l'	51.7		0.138		kg/m
E	7.930E+09	7.930	1.313E+11	131.3	Pa | GPa
ρ = m/V	1422.9		6498.0		kg/m³
A	0.036320		2.124E-05		m²
J	1.560E-06	1.560	4.522E-09	0.005	m⁴ | mm⁴
EJ	1.237E+04		5.935E+02		N·m²
	v, m/s	T_{ld}, s	f_{ld}, Hz	n_{ld}, rpm	
v_b	2.9	1.39	0.72	43.3	
average	v, m/s		E_{avg},GPa	ρ, kg/m³	
v_{ST}	4340.0		122.4	6498.0	
$v_{LWS\text{-}avg}$	2360.0		7.925	1422.9	

The theory and methodology presented in the works of (Harrison 1983, Harrison 1986) is based on the transmission of tensile stress waves through the skeletal structure of the belt, where the wave speed is highest, as all other media through which mechanical stress is transmitted slow down or the wave speeds are lower. Characteristic velocities – the velocity calculated as a square root of the characteristic wave equation can be calculated from:

$$v_c = v_{CU} - \frac{\Delta Q}{Q_D}(v_{CU} - v_{CL}), m/s$$

$$v_{CL} = v_{ST}\sqrt{\frac{\mu_{ST}}{\mu_b + \mu_{rC}(x) + \mu_m(x)}}, m/s$$

$$v_{CU} = v_{ST}\sqrt{\frac{\mu_{ST}}{\mu_b + \mu_{rC}(x)}}, m/s$$

$$v_R = v_{ST}\sqrt{\frac{\mu_{ST}}{\mu_b + \mu_{rR}(x)}}, m/s,$$

where:

- v_{ST} =4300 m/s is the speed of the elastic steel cord (SC) of the belt fabric;
- v_{FW} = 500 to 1500 m/s is the speed of the elastic wave in the fabric cord (FW) of the belt;
- μ_{rC} is linearly distributed mass in the load branch of the conveyor from the roller supports;
- μ_m – linearly distributed mass of presence of material on the belt;
- μ_{rR} – linearly distributed mass from the roller supports in the return branch of the conveyor;
- μ_b – linearly distr. mass of the rubber elements of the strip;
- μ_{ST} – linearly distributed mass of the steel cord (steel ropes) of the belt skeleton;
- μ_{FW} – linearly distributed mass of the textile cord (textile weft) of the belt skeleton;

For belts with a textile cord in the formulae, the symbols with the index "SC" are replaced by those with "FW".

Table 2. *Indicators of wave speeds and their corresponding frequencies* according to Harrison (Harrison 1983)

	v	T = 2·L/v	f = 1/T	n = 60·f
1983,2020 H-n	m/s	s	Hz	rpm
v_{CL}	762.0	5.17	0.19	11.6
v_{CU}	1961.9	2.01	0.50	29.9
v_R	2069.2	1.90	0.53	31.5
v_C	1242.0	3.17	0.32	18.9
$v_{tkp\text{-}L}$	13770.4	0.29	3.5	209.7
$v_{tkp\text{-}U}$	18844.9	0.21	4.8	287.0

According to the methodology of Lodewijks, Hou and Nutall (Lodewijks 2022, Hou 2008, Nutall 2007), the transmission speeds of the axial stresses through the belt are defined as the speed of an elastic (sound) wave:

- longitudinal wave speed $v_{LWS} = \sqrt{\frac{E}{\rho}}$ or $v_{LWS} = \sqrt{\frac{E_b \cdot A}{\mu_b}}$;

- axial propagation speed of transverse waves

$$v_{APS} = \sqrt{\frac{\sigma}{\rho}} = \sqrt{\frac{F_t}{\rho_b \cdot A}} \text{ or } v_{APS} = \sqrt{\frac{F_t}{\mu_b}},$$

at an unloaded belt, as only the modulus of elasticity and the density of the medium affect the transmission of the elastic wave.

Table 3. *Indicators of wave speeds and their corresponding frequencies* according to Lodewijks (Lodewijks, 1996)

1996 Lws	v, m/s	T, s	f, Hz	n, rpm
v_{LWS}	2360.7	1.67	0.60	35.9
v_{APS}	61.4	64.12	0.02	0.9
$v_{LWS\text{-}R}$	1751.5	2.25	0.44	26.7
$v_{LWS\text{-}L}$	703.8	5.60	0.18	10.7
$v_{APS\text{-}R}$	45.6	86.42	0.01	0.7
$v_{APS\text{-}U}$	89.2	44.18	0.02	1.4
$v_{APS\text{-}L}$	37.4	105.23	0.01	0.6

The elastic wave speeds under the different loading are calculated by:

$$v_{LWS-R} = \sqrt{\frac{E_b \cdot A}{\mu_b + \mu_{rR}}} \quad v_{LWS-L} = \sqrt{\frac{E_b \cdot A}{\mu_b + \mu_{rU} + \mu_m}}$$

$$v_{APS-R} = \sqrt{\frac{S_1}{\mu_b + \mu_{rR}}} \quad v_{APS-L} = \sqrt{\frac{F_t}{\mu_b + \mu_{rC} + \mu_m}}$$

$$v_{APS-U} = \sqrt{\frac{F_t}{\mu_b + \mu_{rC}}},$$

and here the tensile force in the belt fabric is included as a determining factor.

According to Bortnowski's methodology (Bortnowski 2021): the belt is considered as a stretched string, where the wavelength and the length of the sector of the string are related: $L = n_m \cdot (\lambda / 2)$, and for the frequency *f = v/l* of the string, the influencing factors are two: tensile force and linearly distributed mass of the load on the string:

$$f_{str} = \frac{n_m \cdot \pi}{2 \cdot L}\sqrt{\frac{F_t}{\mu_i}}, Hz \quad \text{and} \quad v_{str} = f \cdot \lambda = f \cdot \frac{2 \cdot L}{n_m} = \pi\sqrt{\frac{F_t}{\mu_i}}, m/s,$$

where n_m is the number of the mode being investigated. If the dependence of the influence of the speed of movement and filling of the belt on its natural frequencies, it could be written:

$$f_{str}\left(L,F_t,v_b\right)=\frac{n_m\cdot\pi}{2\cdot L}\sqrt{\frac{F_t-B\cdot\mu_{Ai}\cdot v_b^{\,2}}{B\cdot\mu_{Ai}}},Hz$$

Table 4. *Indicators of wave speeds and their respective frequencies* according to Bortnowski (Bortnowski 2021)

2021 Bw	n_m =	1		
	v, m/s	T, s	f, Hz	n, rpm
$v_{str\text{-}R}$	207.6	18.97	0.053	3.2
$v_{str\text{-}CL}$	117.6	33.50	0.030	1.8
$v_{str\text{-}CU}$	241.4	16.32	0.061	3.7
$v_{sb\text{-}R}$	0.0	210275.3	0.000	0.0
$v_{sb\text{-}CL}$	0.0	523289.3	0.000	0.0

The Bortnowski development of the above analytical methodology for string behaviour with consideration of the bending stiffness of the belt and the bending of the belt under the action of weight and the influence of the speed of the belt is given as follows (Bortnowski 2021):

$$f_b\left(L,F_t\right)=\frac{n_m\cdot\pi}{2\cdot L^2}\sqrt{\frac{EJ}{B\cdot m_t}}\cdot\sqrt{1+\frac{F_t\cdot L^2}{\pi^2\cdot EJ}},Hz$$

$$f_b\left(L,F_t,v_b\right)=\frac{n_m\cdot\pi}{2\cdot L^2}\sqrt{\frac{EJ}{B\cdot m_t}}\sqrt{1+\frac{\left(F_t-B\cdot m_t\cdot v_b^{\,2}\right)\cdot L^2}{\pi^2\cdot EJ}},Hz\ ,$$

where EJ is the bending stiffness of the belt (Lazov, 2018, Lazov 2014).

Table 5. *Indicators of wave speeds and their corresponding frequencies at active length variation*

L_i	5	m	k_L =	5
	v, m/s	T, s	f, Hz	n, rpm
$v_{sb\text{-}R}(L)$	2908.7	1.355	0.74	44.3
$v_{sb\text{-}CL}(L,F_t)$	14798.4	0.266	3.76	225.4
$v_{sb\text{-}CL}(L,v,F_t)$	14754.2	0.267	3.74	224.7

The methodology for considering the influence of the bending stiffness of the belt on the transverse waves gives results indistinguishable from zero, as shown in the last two rows of Table. 5. In this case, this is due to the priority influence of the length, which prevails over the other factors.

If the length is changed, the results presented in Table. 5 show that it is possible for band-bending phenomena to introduce significant effects. Such a change of the terminal length (the length between the axles of the driving and turning drum) that affects the performance of the string in the belt conveyor is the entry into a different roller system or the transitions between: driving and tensioning drum; the two driving drums, etc.

In the methodology for studying a system with concentrated masses, the main factors are the stiffness of the belt and the number and size of the masses set in motion. The linear deformation modulus of the belt could be determined experimentally according to:

$$E_Y=\frac{\sigma}{\varepsilon}=\frac{F/A}{\Delta L/L}=\frac{F\cdot L}{A\cdot\Delta L}=\frac{F\cdot L}{B\cdot s_\Sigma\cdot\Delta L},MPa\ ,$$

or if given the catalogue value of the belt modulus (ContiTech 2018) to convert, to the following dependencies:

$$BM\left(N/mm\right)=\frac{E\left(MPa\right)}{B\left(mm\right)}\cdot A\left(mm^2\right)$$

$$BM\left[\left(N/mm\right)/Ply\right]=\frac{E\left(MPa\right)}{z_{ply}\cdot B\left(mm\right)}\cdot A\left(mm^2\right)$$

The reduced mass and the number and rates of freedom involved in the oscillation can be determined by various methods. Harrison proposes (Harrison 2018) to analyse a single mass system where the mass of the tension weight moves together with one third of the total mass of the conveyor, including the reduced masses of the rotating parts, too. The swing frequency of the tension weight and the influence of the tension weight is calculated from:

$$\left(m_{tkp}+1/3\cdot m_{eq}\right)\cdot\ddot{x}+b\cdot\dot{x}+c\cdot x=F\left(t\right)$$

$$f_{tkp\,i}=\frac{1}{2\cdot\pi}\sqrt{\frac{c_i}{m_{tkp}+1/3\cdot m_{eq\,i}}},Hz$$

Results for the considered conveyor are given in Table. 2. This setting is not applicable to tensioning devices that are rope-polyspaston, spring, and hydraulic without modification and addition. For them, the methods discussed in previous publications could be applied, more specifically (Damyanov 2009a, 2009b, 2008). It should be foreseen that when the systems and methods are complicated, there is a question of identifying and evaluating some (Gladysiewicz 2019, Grincova 2016) of the indicators of the dynamic process in order to specify and refine the calculations.

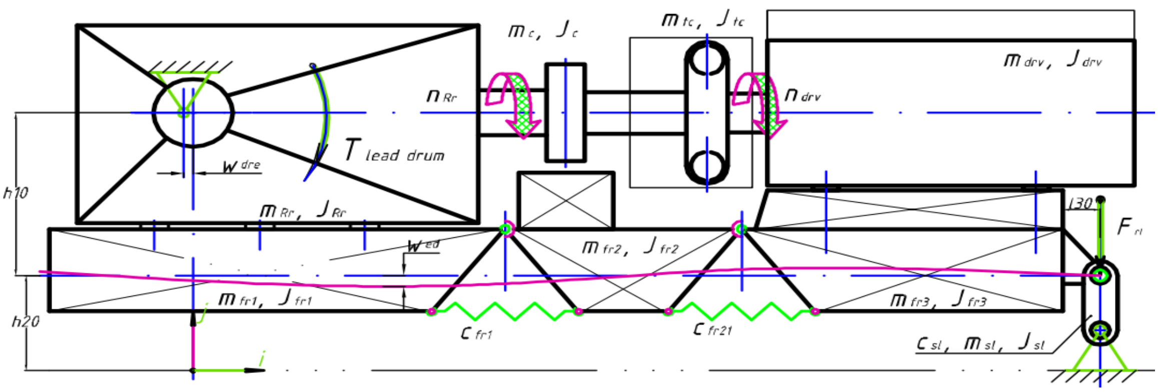

Fig. 1. Scheme towards calculation of the driver group frame

Results and analysis

The present study examines the problem of the occurrence and propagation of oscillating processes and phenomena from the position of the driver group of the conveyor belt. A dynamic model and research methodology was synthesised and investigated, focusing on a detailed study of the behaviour of the driver group frame as a mechanism, the bending of the driver group frame, as well as the dependent or related variations of the motor torque transmitted to the belt driver drum.

The methodology could be summarised as follows:
- generalised analysis of dynamic indicators for calculation and evaluation of periodic and discrete phenomena;
- dynamic model of the driver system and conveyor;
- assessment of periodic and harmonic processes and indicators;
- modal analysis of the driver group frame;
- analyses and conclusions.

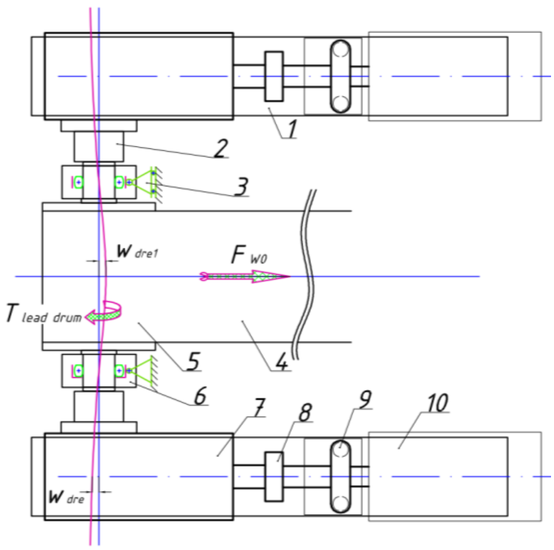

Fig. 2. Scheme of a station with a double-sided drive of a driving RBC drum

The dynamic model of the conveyor is made in a simulation (OSMC 2023, SimX 2023) environment, Figure 4, and the purpose of this model is to be able to introduce the factors affecting and determining the transmission of various elastic waves between the conveyor and the driver group. The model is built as linearized, where all elastic-damping elements are linear functions of the deformation.

The driver station may consist of one or two motor-reducer groups connected to the driver drum shaft, as shown in Figure 2. The driver groups are configured as shown in Figure 1. The movement of the group around the centre of rotation could be seen as a hinge four-bar mechanism around the centre of the driver drum bearings. The movement of the main unit (the frame) is dictated by the reactive moment in the driver drum shaft and is transmitted through the frame structure to the support reaction rod, and the dynamic components of the reactive moment are determined by the conveyor.

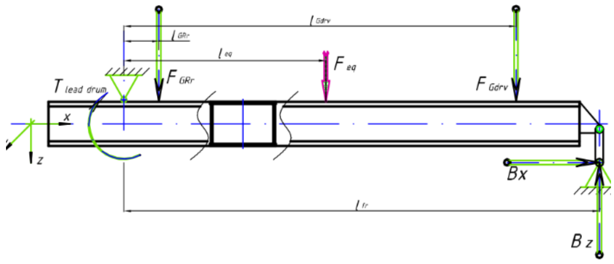

Fig. 3. Scheme for the calculation of frame bending

As to a first approximation, the driver group frame is modelled as a deformable beam according to the loading scheme of the driver station shown in Figure 3. In the driver group frame model considered in this way, priority is given to bending, since the other types of stress are partially or completely relieved by the configuration of the bearing supports. The study of bending (Lazov 2014, Lazov 2018) of a spatial beam is not a complex process and could be performed using a classical approach or the finite element method. In the idealised scheme, thus adopted, the bending moment on the frame of the group is composed of the weight of the nodes and the driving torque acting on the conveyor drum.

The results of analysis with the finite element method are presented in Figures 5, 6, 7, and 8, according to the loading scheme, Figure 1, as having in mind that in case of malfunction of the supports, the idealisation described above is not valid and another model is needed for research. The modal analysis of the frame with finite element method (Figures 5, 6, 7, and 8) in bending modes for the different variants shows frequencies from 12 Hz to 43 Hz. These results differ several times and more (up to 100 times) from the results presented in Tables 2, 3, and 4, but compared to the results presented in Table 5 the difference is 2 or 3 times, as the main influencing factor is the terminal length. It should not be missed that these values (12-43 Hz) are within the range of the motor, especially when starting and stopping the conveyor. Also, this range is within the range of influence of tension weight.

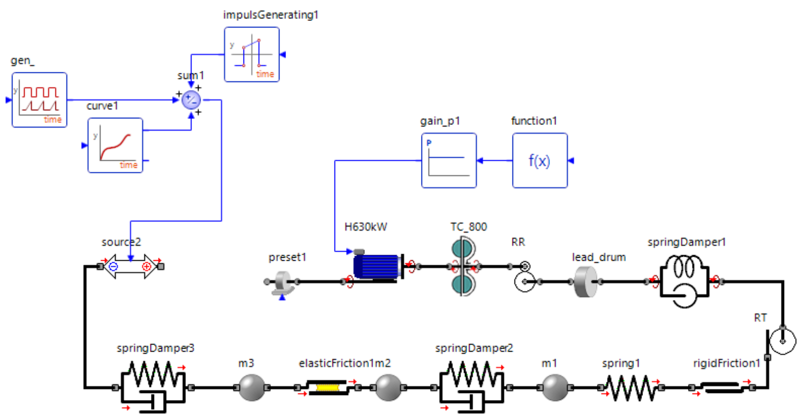

Fig. 4. Dynamic model of the driver group

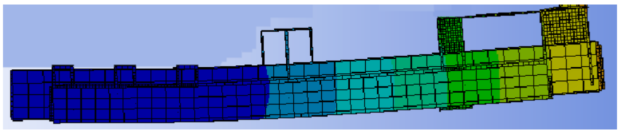

Fig. 5. Own shape of bending oscillations at frequency 12.2 Hz

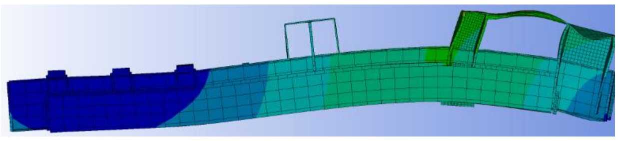

Fig. 6. Own shape of bending oscillations at frequency 31.9 Hz

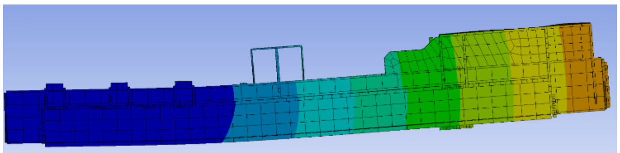

Fig. 7. Own shape of bending oscillations at frequency 17.3 Hz for a stiffening variant

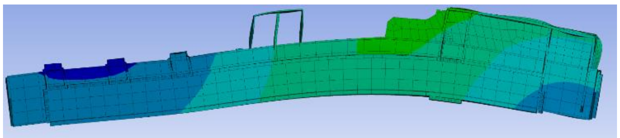

Fig. 8. Own shape of bending oscillations at frequency 42.6 Hz for a stiffening variant

As the phenomena and the dependencies between them become more complex, the models with which we evaluate them also become more complex. By comparing the forced frequencies at the start-up of the conveyor, the areas of intersection between the various components could be estimated.

Tables 6 and 7 show the results at the initial and final iterations for changing the frame stiffness of the driver group spr_FR and in the loaded and unloaded case ($m_1 = m_2 = m_3$, $m_{1L} = 314.5$ t, $m_{1U} = 128.94$ t) - when studying the natural frequencies and shapes through the model shown at Figure 4.

Table 6. *Calculated natural frequencies from the model at parameters of the original driver group frame*

c_{1_FR}	3.345E+08	Nm/rad			
	f_L, Hz	f_U, Hz	Δf, %	n_L, rpm	n_U, rpm
f_{0-1}	0.123	0.189	35.1	7.4	11.4
f_{0-2}	2.38	3.64	34.6	142.7	218.4
f_{0-3}	4.15	6.44	35.5	249.1	386.2
f_{0-4}	16.02	16.67	3.9	960.9	1000.0

Table 7. *Calculated natural frequencies from the model when changing the parameters of the driver group frame*

c_{2_FR}	1.199E+09	Nm/rad			
	f_L, Hz	f_U, Hz	Δf, %	n_L, rpm	n_U, rpm
f_{0-1}	0.123	0.185	33.4	7.4	11.1
f_{0-2}	2.38	3.46	31.3	142.7	207.7
f_{0-3}	4.15	6.44	35.5	249.0	386.3
f_{0-4}	17.37	18.07	3.9	1042.4	1084.4

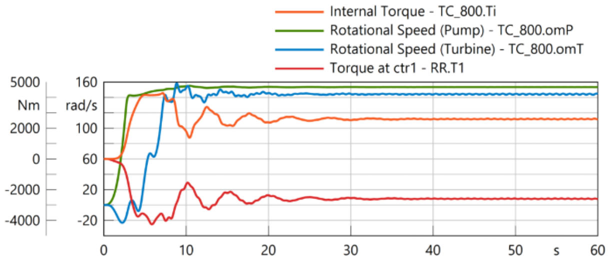

Fig. 9. Simulation results for the behaviour of the driver group - indicators of the turbo clutch

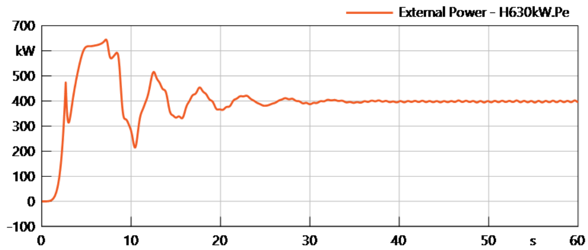

Fig. 10. Simulation results for powertrain behaviour - engine power

Conclusions

The precise calculation of the modal characteristics given in Tables 6 and 7 shows that the stiffening of the frame as shown in Figures 5 to 8 and set for several iterations respectively 3.6 times increase in frame stiffness, shows a result of 7.8% increase on the fourth natural frequency of the system and negligible influence on the first three. In this particular case study, such a change is not sufficient because the fourth natural frequency is 33% of the motor rotation frequency.

The main requirement for calculating the modal characteristics of the system to the presented model is fulfilled and the model allows setting stiffness and mass parameters by calculating the numerical values of the modal indicators of the system. Several parametric nodes representing elastic and dry friction in the system have been added to the model, but their precise parameterisation requires the processing of experimental data, which is not the subject of this study.

The model allows changing several parameters and monitoring dozens of system indicators. The main set indicators are the stiffness of the belt (Belt modulus), the division of the total mass of the moving parts of the conveyor between the moving masses m_1-m_3, the characteristic and components of the resistance force F_{res2}, and the indicators of the driver nodes, respectively, electric motor and turbo coupling.

The main indicators for analysis and tracking can be purely mechanical, as shown in Figures 9 and 10 - the mechanical characteristics of the driver group. Dynamic and kinematic characteristics could also be monitored.

The refinement of the model results and the computational methodology requires the development of an experimental set-up and processing of its results, which is a goal and a challenge for future development.

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