OPTIMAL DESIGN OF HEAVY INDUSTRY CONVEYORS

Abstract: *In this paper a new approach in the design of the heavy industry conveyors is presented. In order to illustrate the optimal design with genetic algorithms of the heavy loaded conveyors chains are side by side presented the selection procedure and the example published by U. S. Tsubaki Chain Division in its brochures and the optimal design of the same problem performed by the authors. The two design solutions are very different but the optimal design offers the advantage of a strongly reduced input power (about 28% with respect of U. S. Tsubaki solution) at the same operating capacity.*

Key words: *optimal design, genetic algorithms, heavy industry conveyors.*

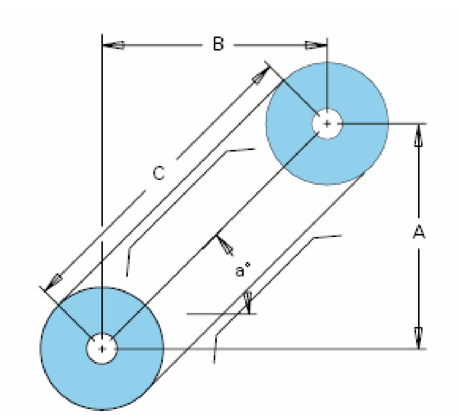
1. Introduction

The main goal of this paper is to emphases once again the advantages of the optimal design of all sorts of products. In this particular case we deal with the optimal design of an incline scraper conveyor. This design problem is extracted from the U.S. Tsubaki Chain Division brochure [1]. After we comment the Tsubaki solution we will perform the optimal design with genetic algorithms of the same product.

2. Input Data

The sketch of the conveyor is shown in Figure 1 and the main requirements of the incline scraper conveyor are listed below.

- Operating capacity: Q = 150 t/h;
- Operating speed: S = 0.508 m/s;
- Total lift: A = 9.144 m;
- Total horizontal run: B = 12.192 m;
- Sprocket centers: C = 15.24 m;
- Infrequent moderate shock;
- 24-hour operation – “Dirty” conditions;
- Scraper paddle: 7 × 305 × 584 mm, 10.402 kg each spaced every 304.8 mm (2 pitches).



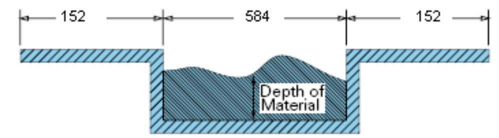


Fig.1

3. U.S. Tsubaki Selection/Check Procedure

The U.S. Tsubaki Selection/Cheek procedure encompasses 9 phases, as follows:

a. Determine the Conveyor's Basic Requirements

Weight of conveyed material on the conveyor:

$$M = \frac{Q}{S} \quad \text{[kg/m]} \qquad (1)$$

Estimated weight of chain:

$$CW = 0.002 \cdot M \cdot C = 7.441 \quad \text{[kg/m]} \qquad (2)$$

Weight of chain, attachments, and other moving parts of conveyor

$$W = CW \cdot N + W_S = 48.882 \quad \text{[kg/m]} \qquad (3)$$

N – number of chain strands = 2;
W_s – weight of slats per linear unit = 10.402 [kg].

Estimated Friction Coefficients:
- material/steel trough: $f_m = 0.55$
- chain/steel trough: $f_s = 0.33$

Sideboard Friction

$$J = 5 \cdot 10^{-5} \cdot \frac{C \cdot h^2}{R} = 569.569 \quad \text{[N]} \qquad (4)$$

h – height of slat = 305 [mm];
R – factor of material = 14 [mm^3/N]

Service Factor (takes into account the frequency of shock, character of conveyor loading, conditions of operation and daily operating period):

$$V = 1 \cdot 1.2 \cdot 1.4 \cdot 1.2 = 2.016 \qquad (5)$$

b. Calculate Conveyor Pull

$$\begin{aligned} P = 9.81 \cdot [(M \cdot f_m + W \cdot f_s) \cdot \cos a + \\ + (M + W) \cdot \sin a] \cdot C + J = 18310 \quad \text{[N]} \end{aligned} \qquad (6)$$

c. Select Sprocket Size

From [5] one obtains 12-tooth sprocket (z = 12) as best selection choice, that means a *Speed Correction Factor E* = 0.990.

d. Calculate Design Conveyor Pull

$$DP = P \cdot V \cdot E = 36540 \ \text{[N]} \tag{7}$$

e. Calculate Chain Tensions

$$T = \frac{1.2 \cdot DP}{N} = 21920 \ \text{[N]} \tag{8}$$

f. Select Chain Size

Referring to [5], G-29 or G-19 attachments are convenient for bolting scraper flights. Since attachment spacing is every 12” (304.8 mm), one may choose either 4”, 6”, or 12” pitch chain. The chain 1131R with G-29 accessories every 2nd pitch was selected and its main features are presented in Table 1.

Table 1

Pitch	p	152.4	mm
Pin diameter	d_p	19.05	mm
Bushing diameter	d_b	28.575	mm
Roller diameter	d_r	76.2	mm
Roller length	l_r	36.575	mm
Bearing area	BA	1038.7	mm^2
Chain weight	W_{cl}	18.6	kg/m
Chain weight with accessories	W_{cla}	22.6	kg/m
Maximum work load	MWL	26253	N

g. Recalculate Actual Chain Tension

$$W = CW \cdot N + W_S = 75.22 \ \text{[kg/m]} \tag{9}$$

Since we chose a rolling chain (not a sliding one), we have to calculate the specific rolling friction coefficient with the equation:

$$f_l = \frac{d_b}{d_r} \cdot f_s = 0.124 \tag{10}$$

$$\begin{aligned} P = 9.81 \cdot [(M \cdot f_m + W \cdot f_s) \cdot \cos a + \\ + (M + W) \cdot \sin a] \cdot C + J = 24668 \ \text{[N]} \end{aligned} \tag{11}$$

$$DP = P \cdot V \cdot E = 47044 \ \text{[N]} \tag{12}$$

$$T = \frac{1.2 \cdot DP}{2} = 28226 \ \text{[N]} \tag{13}$$

h. Check Roller/Bushing Bearing Pressure

Load of a single roller:

$$T_r = \frac{2 \cdot T}{z \cdot K_n} = 6237 \quad [\text{N}] \tag{14}$$

K_n – factor of load uniformity = 0.75

Roller/bushing bearing pressure:

$$BP_r = \frac{T_r}{d_b \cdot l_r} = 6 \quad [\text{MPa}] \tag{15}$$

Note that the bearing pressure BP_r should be less than the allowable bearing pressure BP_a = 5 [MPa]. Obviously, the choosing of the chain 1131R with G-29 accessories was a wrong decision.

i. Calculate Design Input Power (Horsepower)

$$P_{kW} = 1.2 \cdot 10^{-3} \cdot K_\eta DP \cdot S = 32.2 \quad [\text{kW}] \tag{16}$$

K_η – factor that compensates for motor efficiency = 1.1

It is worthy to mention here that all the above equations are modified according to the metric system.
In Figure 2 the U.S. Tsubaki design solution of the problem is presented.

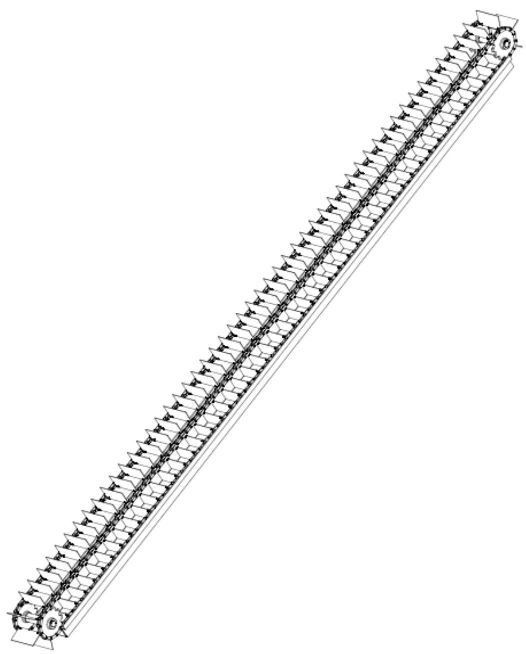


Fig. 2

4. Optimal design of the conveyor

a. Genes

After a rigorous study of the optimization problem we choose 7 genes as in Table 2.

Table 2

No	Gene		Values
0	Chain index	i	0 … 31 (se Table 3)
1	Teeth number of the sprocket	z	9, 10, 11, 12, 14, 16, 18, 20, 24
2	Number of chain strands	N	2, 3
3	Slate width	g	5, 6, 8, 10, 12, 15, 18, 20
4	Slate height	h	100 … 500
5	Slate length	L_r	200 … 2000
6	Pitch ratio*	r_p	2, 4, 6, 8

Note: *) mounting slate pitch/chain pitch

b. Objective function

As objective function we chose the necessary power (horsepower) to act the conveyor.

$$P_{kW} = 1.2 \cdot 10^{-3} \cdot K_{\eta} DP \cdot S \quad [\text{kW}] \qquad (17)$$

c. Constraints

We attached to the optimization problem a set of 6 constraints. All value of these constrains has to be negative or zero.

C1. The chain tension (in one strand) has to be inferior or equal to the maximum work load (see Table 3):

$$g_1 = \frac{T}{MWL} - 1 \qquad (18)$$

In order to compute the actual value of constraint C1 we have to know some important parameters:

- Weight of a slate (ρ_r = 7300 kg/m):

$$m_r = g \cdot h \cdot L_r \cdot \rho_r \cdot 10^{-3} \quad [\text{kg}] \qquad (19)$$

- Mounting slate pitch:

$$p_r = r_p \cdot p \quad [\text{mm}] \qquad (20)$$

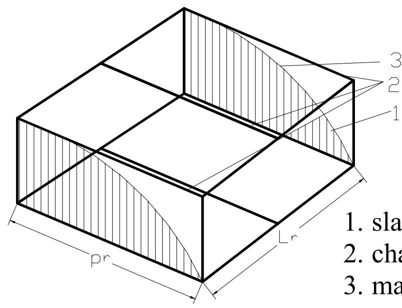
Table 3

No.	Chain	Pitch p [mm]	Pin diameter d_p [mm]	Bushing diameter d_b [mm]	Roller diameter d_r [mm]	Bearing length L [mm]	Chain weight		Max. work load MWL [N]
							with acc. W_{cacc} [kg/m]	without acc. W_c [kg/m]	
0	89R+G19	101.600	16.002	22.352	57.150	31.750	17.41	15.77	20024
1	925R+G19	228.600	16.002	21.504	76.200	41.402	14.58	12.20	18466
2	B-1263R+G19	304.800	19.050	40.195	88.900	49.276	20.68	16.37	32038
3	C-1263R+G19	304.800	19.050	40.195	88.900	49.276	21.13	16.37	32038
4	D-1263R+G19	304.800	19.050	40.195	88.900	49.276	19.94	16.37	32038
5	E-1263R+G19	304.800	19.050	40.195	88.900	49.276	22.77	17.86	32038
6	B-1264R+G19	304.800	22.352	30.735	101.600	55.626	27.08	22.32	40938
7	B-1266R+G19	304.800	19.050	29.145	82.550	39.624	17.86	14.14	28033
8	1273R+G19	304.800	25.400	36.016	127.000	65.024	41.07	31.99	56512
9	B-1863R+G19	457.200	19.050	27.102	88.900	49.276	16.37	14.14	32038
10	D-1863R+G19	457.200	19.050	27.102	88.900	49.276	17.11	14.14	32038
11	F-1863R+G19	457.200	19.050	27.102	101.600	49.276	17.71	14.88	32038
12	B-1864R+G19	457.200	22.352	30.735	101.600	55.626	21.13	17.86	40938
13	G-1864R+G19	457.200	22.352	30.735	101.600	55.626	20.83	16.37	40938
14	1867R+G19	457.200	38.100	55.206	152.400	74.676	53.42	46.87	99229
15	1871R+G19	457.200	31.750	44.285	127.000	68.326	36.75	31.25	72976
16	1873R+G19	457.200	25.400	36.016	127.000	65.024	31.55	25.30	56512
17	94R+G29	101.600	12.700	19.128	38.100	20.574	7.89	6.10	10679
18	1131R+G29	152.400	19.050	28.399	76.200	36.576	22.62	18.60	26253
19	WH-78+H1	66.269	12.700	19.05	–	–	12.35	5.95	15574
20	WH-82+H1	78.105	14.224	21.336	–	–	14.73	7.14	19579
21	WH-78+H2	66.269	12.700	19.05	–	–	11.16	5.95	15574
22	WH-82+H2	78.105	14.224	21.336	–	–	13.09	7.14	19579
23	WH-82+wing	78.105	14.224	21.336	–	–	12.80	7.14	19579
24	WH-124+wing	101.600	19.050	28.575	–	–	21.13	12.35	32706
25	WH-124H+wing	103.200	25.400	38.1	–	–	28.42	21.87	46722
26	WCH-132+wing	153.670	25.400	38.1	–	–	31.25	21.13	68081
27	WH-132+RF12	153.670	25.400	38.1	–	–	81.84	21.13	68081
28	WH-150+RF12	153.670	25.400	38.1	–	–	86.30	25.00	68081
29	WH-155+RF12	153.670	28.702	43.053	–	–	93.74	29.76	88995
30	X-458+S22	102.387	16.002	24.003	–	–	7.74	4.76	17799
31	X-678+S22	153.187	22.352	33.528	–	–	16.96	9.97	31593

Notes:
1. Chains # 0 … 18 = roller chains ($t_l = 0$);
2. Chains # 19 … 31 = sliding chains ($t_l = 1$);
3. Chains # 0 … 18 and 30 … 31 = one side accessories ($t_a = 1$);
4. Chains # 19 … 21 and 28 … 29 = both side accessories ($t_a = 2$);
5. Chains # 23 … 27 = central accessories ($t_a = 0$).

- Theoretical coefficient of admission (see Figure 3 for details):

$$\phi_t = \begin{cases} \dfrac{h}{2 \cdot p_r \cdot \dfrac{A}{B}} & if \quad \dfrac{h \cdot B}{A} \le p_r \\ 1 - \dfrac{p_r \cdot \dfrac{A}{B}}{2 \cdot h} & if \quad \dfrac{h \cdot B}{A} > p_r \end{cases} \tag{21}$$



1. slate
2. chain
3. material

Fig. 3

- Actual coefficient of admission:

$$\phi = \phi_t \cdot K_b \tag{22}$$

- Chain length:

$$L_t = 2 \cdot C \cdot 10^3 + z \cdot p \quad \text{[mm]} \tag{23}$$

- Chain length (in number of pitches):

$$X_t = \text{ceil}\left(\frac{L_t}{p}\right) \quad \text{[pitches]} \tag{24}$$

- Actual sprocket centers:

$$A_t = 0.5 \cdot p \cdot \left[X_t - z\right] \quad \text{[mm]} \tag{25}$$

- Weight of chain with accessories mounted each r_p pitches (see Table 3 also):

$$CW = W_{cl} + W' \tag{26}$$

$$W' = \begin{cases} \dfrac{W_{cla} - W_{cl}}{r_p} \; if \; t_a \in \{0, 1\} \\ \dfrac{2(W_{cla} - W_{cl})}{3r_p} \; if \; t_a = 2 \end{cases} \quad \text{[kg/m]} \tag{27}$$

- Weight of slates mounted each r_p pitches (see Table 3 also):

$$W_s = \text{floor}\left(\frac{1000 \cdot m_r}{p_r}\right) \quad [\text{kg/m}] \qquad (28)$$

- Total chain weight:

$$W = CW \cdot N + W_S \quad [\text{kg/m}] \qquad (29)$$

- Weight of conveyed material on the conveyor:

$$M = 10^{-6} \cdot L_r \cdot h \cdot \rho_m \cdot \phi \quad [\text{kg/m}] \qquad (30)$$

- Sideboard Friction:

$$J = 5.085 \cdot 10^{-6} \cdot 9.81 \cdot \frac{A_t \cdot h^2 \cdot \phi}{R} \quad [\text{N}] \qquad (31)$$

- Calculate Conveyor Pull:

$$\begin{aligned} P = 9.81 \cdot [(M \cdot f_m + W \cdot f_l) \cdot \cos a + \\ + (M + W) \cdot \sin a] \cdot C + J \quad [\text{N}] \end{aligned} \qquad (32)$$

- Speed Correction Factor (the authors found a very simple and linear expression for this important factor):

$$E = \begin{cases} 0.720 \cdot S + 0.7606 \text{ } if \text{ } z = 9 \\ 0.540 \cdot S + 0.7756 \text{ } if \text{ } z = 10 \\ 0.468 \cdot S + 0.7635 \text{ } if \text{ } z = 11 \\ 0.414 \cdot \text{S} + 0.7448 \text{ } if \text{ } z = 12 \\ 0.324 \cdot S + 0.7379 \text{ } if \text{ } z = 14 \\ 0.288 \cdot \text{S} + 0.7304 \text{ } if \text{ } z = 16 \\ 0.252 \cdot \text{S} + 0.7230 \text{ } if \text{ } z = 18 \\ 0.234 \cdot S + 0.7243 \text{ } if \text{ } z = 20 \end{cases} \qquad (33)$$

- Calculate Design Conveyor Pull:

$$DP = P \cdot V \cdot E \quad [\text{N}] \qquad (34)$$

- Chain Tensions:

$$T = \frac{1.2 \cdot DP}{N} \quad [\text{N}] \qquad (35)$$

Now, the equation (18) is completely calculable and some parameters will be used in the next equations of the remaining constrains.

C2. The bearing pressure between the roller and busing must be less than the allowable bearing pressure:

$$g_2 = \begin{cases} \dfrac{BP_{rola}}{BP_a} - 1 & if \quad t_1 = 0 \\ -1 & if \quad t_1 = 1 \end{cases} \tag{36}$$

Note that in the case of sliding chains this constraint has not to be check. It is necessary to compute the following parameters, also:

- Load of a single roller:

$$T_r = \frac{2 \cdot T}{z \cdot K_n} \quad [\mathrm{N}] \tag{37}$$

- Roller/bushing bearing pressure:

$$BP_r = \frac{T_r}{d_b \cdot l_r} \quad [\mathrm{MPa}] \tag{38}$$

C3. Chain speed has to be greater than a minimal imposed value (v_{min} = 0.3):

$$g_3 = \frac{v_{\min}}{S} - 1 \tag{39}$$

The chain speed will be given by the following equation:

$$S = \frac{10^6 \cdot Q}{3.6 \cdot L_r \cdot h \cdot \phi \cdot \rho_m} \quad [\mathrm{m/s}] \tag{40}$$

C4. Chain speed must be less or equal to a maximal allowed value (v_{max} = 0.6 m/s):

$$g_4 = \frac{S}{v_{\max}} - 1 \tag{41}$$

C5. The bending stress in the critical cross section of the slate has to be less or equal the allowable bending stress (σ_{ia} = 90 MPa):

$$g_5 = \frac{\sigma_i}{\sigma_{ia}} - 1 \tag{42}$$

In the equation (42), the bending stress strongly depends on the number of the chain strands. Since the load that acts on a single slate is:

$$F = 9.81 \cdot \left(\frac{M \cdot p_r}{1000} + m_r \right) \quad [\mathrm{N}] \tag{43}$$

the bending stress in the critical cross section of the slate is given by the equation:

$$\sigma_i = \begin{cases} \dfrac{3 \cdot F \cdot L_r}{g^2 \cdot h} & if \quad N = 2 \\ \dfrac{3 \cdot F \cdot L_r}{4 \cdot g^2 \cdot h} & if \quad N = 3 \end{cases} \quad [\text{MPa}] \qquad (44)$$

C6. The maximal deflection of the slate must be inferior or equal to the allowable deflection (f_{ia} = 5 mm):

$$g_6 = \frac{f_i}{f_{ia}} - 1 \qquad (45)$$

As the bending stress, the value of the maximal deflection of the slate is based on the number of chain strands:

$$f_i = \begin{cases} \dfrac{F \cdot L_r^{\,3}}{E_r \cdot g^3 \cdot h} & if \quad N = 2 \\ \dfrac{F \cdot L_r^{\,3}}{16 \cdot E_r \cdot g^3 \cdot h} & if \quad N = 3 \end{cases} \quad [\text{mm}] \ (46)$$

d. Software

We used *Kreator v.1.0* software in order to perform the optimization and to finalize on this basis the new design of the incline scraper conveyor.

The main specification and the settings of the *Genetikos* software are presented in Table 4.

Table 4

Specification	Type or Value
Representation of genes	Real
Method of fitness assignment	Dynamic
Method of selection	Roulette wheel
Number of individuals	100
Crossover rate	0.75
Mutation rate	0.05

e. Results

The values of all considered genes, after optimization, are given in Table 3.

Table 5

No	Gene		Values
0	Chain index	i	7
1	Teeth number of the sprocket	z	12
2	Number of chain strands	N	3
3	Slate width	g	8 mm
4	Slate height	h	465 [mm]
5	Slate length	L_r	1525 [mm]
6	Pitch ratio*	r_p	4

Note: *) mounting slate pitch/chain pitch

The chain with index 7 (see Table 3) is B-1266R and the accessories are G19. The main features of this chain are presented in Table 6 and a sketch of the chain is shown in Figure 4.

Table 6

Pitch	p	304.8	mm
Pin diameter	d_p	19.05	mm
Bushing diameter	d_b	28.575	mm
Roller diameter	d_r	82.55	mm
Roller length	l_r	39.624	mm
Bearing area	BA	1154.83	mm^2
Chain weight	W_{cl}	14.14	kg/m
Chain weight with accessories	W_{cla}	17.86	kg/m
Maximum work load	MWL	28033	N

d_r

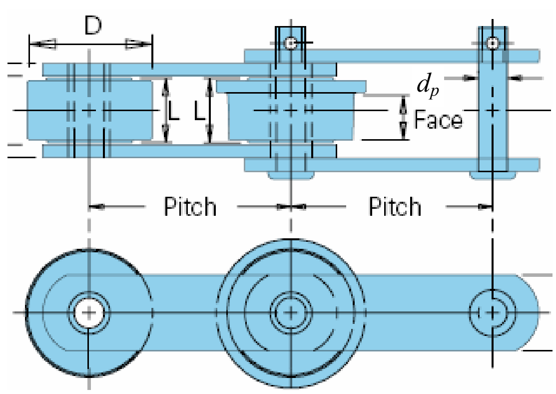


Fig. 4

A picture of the G19 accessory is shown in Figure 5. Note that in the case of obtaining solution there are 3 chain strands, that means that the middle strand will have the accessories mounted on the both sides, and the lateral strands will have the accessories mounted only one side.

Fig. 5

In Figure 6 the optimal design solution of the problem is presented. The real proportions of the constructions were respected.

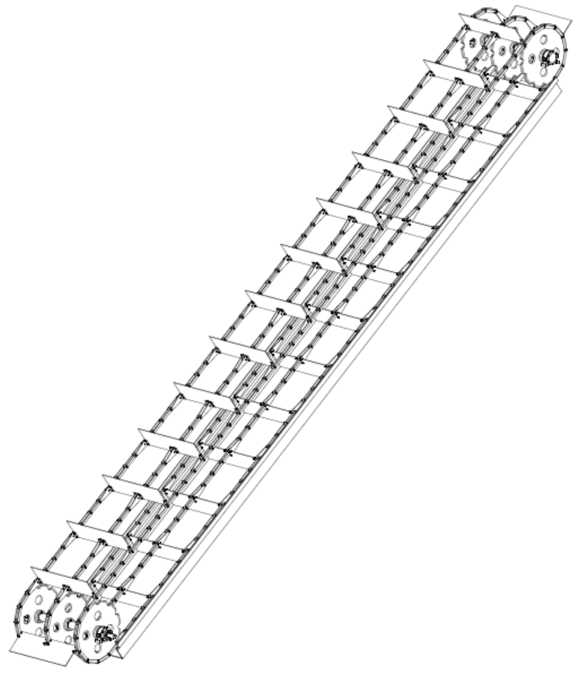


Fig. 6

In Table 4 the specifications of the U.S. Tsubaki solution and the optimal solution are shown side-by-side.

Table 7

U.S. TSUBAKI SOLUTION	OPTIMAL DESIGN
MAIN FEATURES OF THE CHAIN	
TYPE: Rolling chain	
NUMBER OF CHAIN STRANDS *N*	
2	3
CHAIN	
1131R	B-1266R
CHAIN PITCH *p* [mm]	
152.4	304.8
BEARING AREA *BA* [mm^2]	
1038.708	1154.836
ACCESSORY TYPE	
G-29	G-19
CHAIN SPEED *S* [m/s]	
0.508	0.3

<table>
<tr><td colspan="4">NO. OF TEETH OF THE SPROCKETS z</td></tr>
<tr><td colspan="2">12</td><td colspan="2">12</td></tr>
<tr><td colspan="4">CHAIN WEIGHT</td></tr>
<tr><td>with acc. W_{cla} [kg/m]</td><td>without acc W_{cl} [kg/m]</td><td>with acc. W_{cla} [kg/m]</td><td>without acc. W_{cl} [kg/m]</td></tr>
<tr><td>22.62</td><td>18.6</td><td>17.86</td><td>14.14</td></tr>
<tr><td colspan="4">MAXIMUL WORK LOAD MWL [N]</td></tr>
<tr><td colspan="2">26253</td><td colspan="2">28033</td></tr>
<tr><td colspan="4">SLATE WIDTH g [mm]</td></tr>
<tr><td colspan="2">8</td><td colspan="2">8</td></tr>
<tr><td colspan="4">SLATE LENGTH L_r [mm]</td></tr>
<tr><td colspan="2">584</td><td colspan="2">1525</td></tr>
<tr><td colspan="4">SLATE HEIGHT h [mm]</td></tr>
<tr><td colspan="2">305</td><td colspan="2">864</td></tr>
<tr><td colspan="4">SLATE WEIGHT m_r [kg]</td></tr>
<tr><td colspan="2">10.402</td><td colspan="2">41.413</td></tr>
<tr><td colspan="4">MOUNTING SLATE PITCH/ CHAIN PITCH</td></tr>
<tr><td colspan="2">2</td><td colspan="2">4</td></tr>
<tr><td colspan="4">NUMBER OF SLATES</td></tr>
<tr><td colspan="2">106</td><td colspan="2">28</td></tr>
<tr><td colspan="4">IMPUT POWER P_{kW} [kW]</td></tr>
<tr><td colspan="2">32.224</td><td colspan="2">23.365</td></tr>
</table>

Conclusion

The most important result of the optimal design of the incline scraper conveyor is the huge reduction of the input power (horsepower) with respect of the solution adopted by U.S. Tsubaki Chain Division (in the conditions of the same operating capacity). Since the needed power in the optimal design is about 23 kW, and the U.S. Tsubaki solution request a power about 32 kW, that means an economy of 28%.

This massive reduction of the input power (horsepower) was possible because within the frame of the optimal design we let all the variable to vary along their range of values, and we chose as variable amounts that Tsubaki considered them as constants (for example the steel trough width, the height of the slate, the ratio between the mounting slate pitch and the chain pith etc.).

References

1. *** *US Tsubaki Union Engineering Chain General Catalog. Engineering Class Chain*, http://www.ustsubaki.com/eng_chain_cd/sectionA.html, 2006.