CRANE CALCULATION REPORT

Content

24/09/2016

1.0 General description

The lifting appliance considered in this calculation report is a jib crane designed to lift various material in off-shore condition.
This crane is foreseen for ARABIYAH HASBAH DEVELOPMENT PROJECT.
The crane is composed by a column, fixed to the platform deck with a slewing double girder boom of fixed lengh on wich travel a trolley.
The net capacity of the crane at the hook is **15 t**.
The mouvements of the jib crane hook are the classics : **hoisting**, **trolley travel** , **slewing**.

Hoisting
Realized with a winch fixed to the strucure at one end of the boom , rope deviation sheaves on trolley and return sheave on the other end (outreach) of the boom and hook block.

Cross travel: realized by means of four idle wheels at the corners of the trolley with :
N. 1 trolley translation winch located at one end of the boom
N. 1 return sheave at the other end of the boom

Slewing: realized with base bearing that is the connection element between column and boom and two motor-reductors with pinion engaged on the bearing crown.

2.0 CODES AND STANDARD

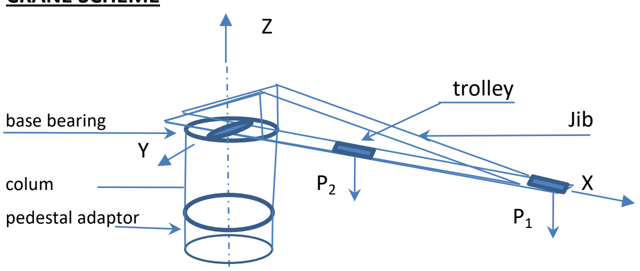
The supply and the design have to comply with the technical specification of the
SAUDI ARABIAN OIL COMPANY:
JIB CRANE - Tag No.G63-U-130
And for the calculation the followjngs code and standard will be adopted::
API 2C
F.E.M. 1.001 3rd edition(UNI7670)

3.0 LOAD COMBINATION

According to API 2C we take the following combination that is the haviest one because all the forces and factors are taken into account as:

Pay load	= 15,0 t	
dynamic factor Cv =	2	
off-lead angle	= 0,50°	
side-lead angle	= 0,50°	
operational wind	= 75,6 Km/h	
tangent acceler	= 0,2m/s2	
centrip. acceler.	= 0,03m/s2	(negligible)

3.1 CRANE SCHEME



3.2 LOAD CODITION

The conditions of load that will be considered in the calculation are two:
Trolley in **position 1** at the extreme (outreach at **10m** from slew centre) position of the boom
In this position we have the haviest condition for boom structure, base bearing, column and pedestal adaptor
Trolley in **postion 2** at the **middle** of the two suspension point of the boom girder
In this position we have to check the stresses of the boom girder were the trolley run.

4 MECHANICAL

In the following we start to calculate the various mechanismes.

4.1 Hoist

The force on the ropes "SWLH" is composed by the pay-load to lift "SWL" plus the hook block dead weight.

SWLH= 150KN+5KN= 155 KN

Where:

150 KN = the load to lift (net crane capacity)

5 KN = Hook block dead weight

4.1.1 rope

The minimum braking load "BL" required according to API rules is:

$BL = W*DF/N*E_{rs} =$ 366 KN

Where:

N = 2 reeving(is 2 lines of rope where 1 line go on the drum).

W = SWLH = 155 KN

DF = 2,25xCv= 4,5

$E_{rs} = (K_b^N-1)/[K_b^S xNx(K_b-1)] =$ 0,952 Reeving system efficiency

$K_b =$ 1,02 for roller bearing

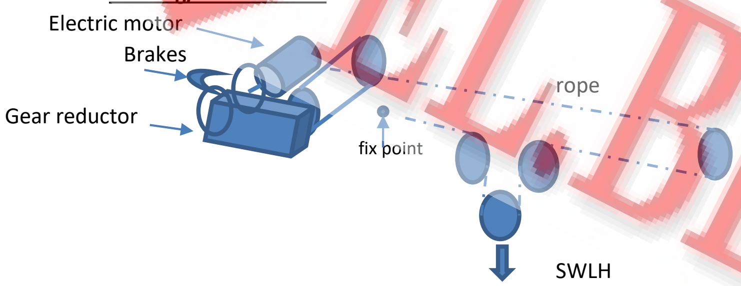
S = 3 total number of sheaves in reeving system

The rope adopted is ϕ 22 mm with BL = 441 KN > 366KN that means DF = 5,5>4,5

The sheave pitch diameter is 400 mm > 22x18 = 396 mm

The drum pitch diameter is 550 mm > 22x18 = 396 mm

Reeving/Hoist scheme



4.1.2 Drum

The drum diameter adopted in function of the geometry is $\phi_D =$ 550 mm

The calculation of the minimum thickness af the drum body, of the drum, shaft and its bearing are in **annexes 1 ,2,3**. This calculation is done with the UNI rules and not only for resistance but to have a life of 25 years that in this case correspond to 12500 hours of work.

4.1.3 Electric motor

The selection of the motors has been done taking in account the hoistig speed of the load "SWLH " and the efficiency due to the reeveng and gearing.

N=required power = SWLH x v /(61,2x0,98^{3}x0,94) = 54,39 Kw

Where:

v = hoisting speed = 19 m/min

The adopted motor has a power of 68 Kw > 54,39 Kw

4.1.4 Brake

The brake service adopted is a doble shoes drum brake with electohydrailic thrustor.
The max torque of the mechanism is:
M_t torque of the mechanism = N_m/ω = 68.000/188,4 = 361 Nm
Where :
N_m = motor power = 68 Kw = 68.000 W
ω = motor angolar speed= 188,4rad/sec (1780rpm) 188,4

The service brake adopted, has M_f = braking moment= 800 Nm= 2,2 x M_t > 1,5x M_t

4.1.5 Gear reductor

The reductor gear has been selected taking in account the max torque required.
The max torque required on the low speed(output) shaft is:

Mt = SWLH/N x ϕ_D /E_{RS} = 22393 Nm

The adopted one has a torque of 29500 Nm > 22393 Nm Type: RXP3-814
The theoretical ratio between the in/out shafts is r = 80,9
The adopted ratio between the in/out shafts is r = 80,6

4.2 TROLLEY TRAVEL

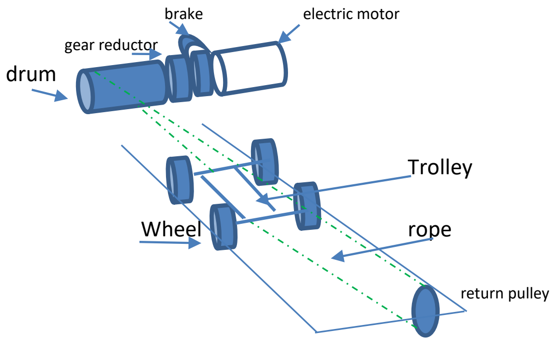
This mechanism is realized by a rope that pull the trolley in the two direction.
The motion is driven by a drum on wich the rope are winded on one side and un-winded on the other one.
The drum is coupled with a motoreductor with brake and the rope is fixed to the trolley.
The trolley run on four wheels over two beams that are the boom of the jib crane .

4.2.1 Wheel

This element transmit the actions to the boom via the rails.
The rails adopted are constitued of a steel plate 30x50mm b = 50 mm
The verification of the correct diameter will be done with the Fem rules for the class M5.
Two condition have to be verified: one for the average load and one for the max load.
Trolley weight Pc= 700 Kg
Wheel diameter D = 200 mm
According to Fem rules we have:
Coefficient C_1 = 1,04 depending from the rpm of the wheel (V= 10 m/min)
Coefficient C_2 = 0,8 depending from the classification
Depending from wheel material p = 5 Mpa
$P_{average}$= [(SWLH +Pc)x2+Pc] / 3/4 = 27,6 KN
P_{max} = (SWLH +Pc)/4 = 40,5 KN
The allowable pressure are:
P_{aveall}= bxDxpxc_1xc_2 = 41 KN > $P_{average}$= 27,6 KN
$P_{max\ all}$=1,4xbxDxp = 69 KN > P_{max} = 40,5 KN

The axle of the wheel and its bearing are verified on **annex 4and annex 5**

4.2.2 Trolley scheme



4.2.3 Rope

The rope required traction is the action enought to pull the trolley with the load (SWLH) under the influence of friction, acceleration, wind and offlead angle.

The component due to offlead: SWLH *sen(0,5°)	=	1352 N
The force due to a wind of 20 m/sec = 300*10	=	3000 N
The force due to the friction rail/wheel = (SWLH+Pc)*0,010	=	8550 N
The force due to acceleration of 0,1 m/sec^2 = 0,15x(SWLH+Pc)	=	2430 N
The total force result	**Ft** =	15332 N

The rope adopted is ϕ 10 mm with BL = 90 KN > 15,4KN that means DF = 5,8
The adopted sheave pitch diameter is = 200 mm > 18x10 = 180 mm
The adopted drum pitch diameter is d= 220 mm > 18x10 = 180 mm
The calculation of the minimum thickness af the drum body, of the drum, shaft and its bearing are in **annexes 6 ,7,8**. This calculation is done with the UNI rules .

4.2.4 Electric Motor

This motor have to be with a torque and than the force enought to react the total force with a power enought to work with the previous conditions .

V = trolley travel speed =	15 m/min		0,25 m/sec
n= motor rot. speed =	1800 r.p.m.		188,40 rad/sec
Mt = torque required=	Ft x V / n	=	20,3 Nm

The motor is 3,5 Kw at 1800 rpm (188 rad/sec) with a start torque of 1,5 times the rated one
Mt = start adopted motor torque = 27,9 Nm > 20 Nm

4.2.5 Brake

The brake adopted is a disc brake with electomagnetic thrustor.
The max torque of the mechanism is:
M_t max of the mechanism is = N/n = 19 Nm
Where :
N = motor power = 3,5 Kw = 3500 W
n = angolar speed= 183rad/sec (1750rpm)
The brake adopted, has M_f = braking moment= 40Nm=2,1 x M_t > 1,0x M_t

4.2.6 Gear reductor

The reductor gear has been selected taking in account the Fem rules for a class M6.
The max torque required on the low speed shaft of the reductors is:

Mt = Ft (pull force) x d/2 (drum radius) = 1687 Nm
The ratio speed of input / speed output shafts is r = 82,896
The adopted one has a nominal torque of 2196Nm with a life of 50.000 hours.
Type: EX253MD r=85,9

4.3 SLEWING

This mechanism is realized by means of a base bearing with external crown. The base bearing has been choise on the basis of the max moment due to the various load and the motion is driven with two motoreductor by pinions engaged with the crown.
The calculation is carried out for the worst condition that is with trolley with full load at 10m, operational wind of 20 m/s, acceleration , side-led angle and bearing efficiency.

SWLH	=	15500 Kg	
Trolley weigth	=	943 Kg	
Slew structure weigth	=	22500 Kg	2,15 m
swing circle	=	0,9 m	
Bearing friction factor	=	0,006	
swing speed	=	0,5rpm	
ramp time	=	6 s	
angolar acceleration	=	0,009 rad/s	
max SWLH radius	=	10,0 m	
polar inertia SWLH	=	1644300 Kgm2	
inertia dead weight	=	104006 Kgm2	

The moment required to swing is the sum of the following actions:

Moment due to the bearing friction	=	2063 Nm
Moment due to acceleration	=	15249 Nm
Moment due to service wind	=	55672 Nm
Moment due to side-lead	=	33144 Nm
The total moment required Mt	=	**106128 Nm**

4.3.1 Electric Motor

The power required during the start is:

N_{swing} = Mtx $\omega/(ca/cn)/\eta$= 3,70 KW

Were:

ca/cn= starting torque = 1,5

general efficiency η = 0,8

The motors are n.2 of 3,5 Kw at 1800 rpm (188 rad/sec) with a start torque of 1,5 times the rated one

N_{swing} = 2x3,5Kw= 7Kw > 3,7 KW

4.3.2 Brake

The brake adopted is a disc brake with electromagnetic thrustor.

The max torque of the mechanism is:

M_t max torque of the mechanism = N/n = 19 Nm

Where :

N = motor power = 3,5 Kw = 3500 W

n = angolar speed= 188rad/sec (1800rpm)

The brake adopted, has M_f = braking moment= 30 Nm=1,6 x M_t > 1,5x M_t

4.3.3 Gear reductors

The reductor gear has been selected taking in account the Fem rules for a class M6.

The max torque required on the low speed shaft of each of the two reductors is:

M_{tr} = (Mt / 2)*(z_p/z_c)= 7939 Nm

where:

Z_p= pinion theet numb.= 19

Z_c= crown theet numb.= 127

The ratio between the input/output shafts is r = 523,6

The adopted one has a nominal torque of 8000Nm with a life of 50.000 hours.

4.3.4 **Base bearing**

The base bearing is a type with three range of roller and external crown of theet.
The overturning moment taking in account the various factor is:

Moment due to factored load (Cv=2)	=	3156 KNm	
Moment due to accelerations	=	30 KNm	
Moment due to Wind direction x	=	6 KNm	1983,75
Total overturning factored moment		**3192 KNm**	
Vertical load factored (Cv=2)	=	**512,0 KN**	

The base earing adopted "191.25.1800.990.41.1502 " see load curve at **annex 9** in this condition with the upmentioned vertical load allow the following moment:

Dynaminc allowable moment Mtd_{all}	=	**4200 KNm**	**>**	**3192 KNm**
Static allowable moment Mts_{all}	=	**4800 KNm**	**>**	**3192 KNm**

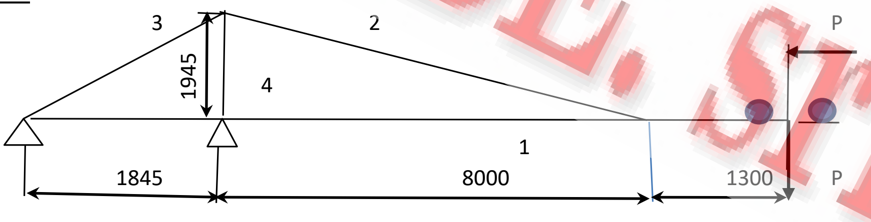
5.0.0 **STRUCTURE**

The calculation is carried out taking in account the forces of the mechanical actions and as told at the beginnig, in a conservative way (See Annex 12).
The material adopted for the structures is S355 J2 (sometimes called also Fe 510D) that, according to API and others codes, being σy/σu<0,7 the allowable stress are:

	operational	with wind	exceptional
σ all =	240 N/mm^2	270 N/mm^2	300 N/mm^2
τ all =	140 N/mm^2	156 N/mm^2	174 N/mm^2

5.1.0 <u>Jib</u>

5.1.1 <u>Jib Scheme</u>



5.1.2 <u>POSITION 1</u>

Girder 1 Two IPE 400 coupled

σ max = 197N/mm2	<	σ all = 240N/mm2
τ max = 53N/mm2	<	τ all = 140N/mm2
σ id = 217N/mm2	<	σ all = 240N/mm2

The stresses are inclusive of all the actions and factors (Cv=2) and the allowable stress is bigger than the max permissible because the actions <u>include also the wind but, to be more conservative, we take in account the stress allowable lower one.</u>

Tie rod 2

This element is only tensioned and is built with a pipe ϕ 152 mm thikness 12 mm

σ max = 157N/mm2 < σ all = 240N/mm2

Tie rod 3

This element is only tensioned and is built with a pipe ϕ 152 mm thikness 12 mm

σ max = 189N/mm2 < σ all = 240N/mm2

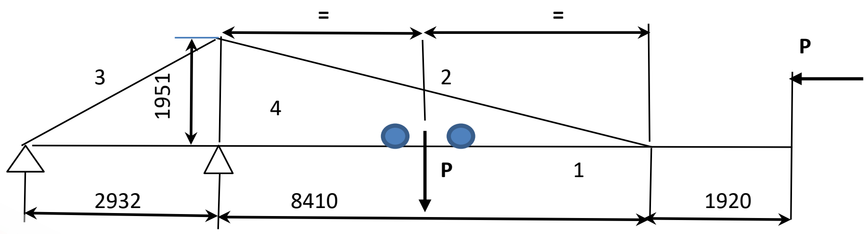
Compressed item 4

This element is only compressed and is built with a pipe ϕ 152 mm thikness 12 mm

σ max = 146N/mm2 < σ all = 240N/mm2

5.1.3 POSITION 2

With the pay load in the middle of the boom as shown we have the following stresses:



Girder 1 σ max = 210N/mm2 < σ all = 240N/mm2

τ max = 26N/mm2 < τ all = 140N/mm2

σ id = 215N/mm2 < σ all = 240N/mm2

The stresses are inclusive of all the actions and factors(Cv=2) and the allowable stress is bigger than the max permissible because the actions include also the wind, but to be more conservative we take the allowable lower one.

Tie rod 2

This element is only tensioned and is built with a pipe ϕ 152 mm thikness 12 mm

σ max = 77N/mm2 < σ all = 240N/mm2

Tie rod 3

This element is only tensioned and is built with a pipe ϕ 152 mm thikness 12 mm

σ max = 92N/mm2 < σ all = 240N/mm2

Compressed item 4

This element is only compressed and is built with a pipe ϕ 152 mm thikness 12 mm

σ max = 89N/mm2 < σ all = 240N/mm2

5.2.0 BASE bearing bolting

According to the API 2C rules,the ultimate strenght criteria of the bolts have to satisfied the following:

P= 4*M/Nb*D - H/Nb = 552,0 KN

where:

P = max allowable force in one bolt

M= 3,75 x total overtur factored mom. = 11970 KNm

Nb = number of bolts = 48

D = bolts circle diam.= 1,685 m

H = 3,75xvertical factored force = 1920,0 KN

A= resistance section of bolt M30 = 561mm2

σ_{bolt}= 984N/mm2

σ_{all} = 1000N/mm2 > σ_{bolt}

So we can say that the **bolts class 10.9** is acceptable.

Flange plate edge column calculation see annex 10

5.3.0 COLUMN

The verification will be carried out for the section at the base were the column is welded to the deck.

Dc = column outer diameter = 1800 mm
th = column thikness = 20 mm
J x = inertia moment = 4427764cm4
Wx = resistent mom. = 49197cm3
Ac=section area = 111784mm2
The actions on this element is the same of the base bearing increased by the moment of transport generated from the wind , acceleration and its the vertical dead load.
Furthermore the API rules foreseen for the crane support structure an additional factor PF that applies to the factored load.

PH = 1,56- SWLH/900000 = 1,56
Were value of PH must be not less than 1,2 **but not greater than 1,5** so in our case :
PH = 1,5
The vertical factored action and moment included the additional factor PH result:

Hc = 937 KN
Mc = 4788 KNm

σ_b max = 97N/mm2 < σ all = 240N/mm2
σ_c max = 8N/mm2 < σ all = 240N/mm2
τ max = negligible

Buckling verification

σ_{bb}= $\sigma_y (1-\sigma_y/(4\ K_b\ E\ (th/Dc)^2)$ = 299,3 N/mm^2
σ_{cb}= $K_c\ E\ (th/Dc)^2$ = 129,6 N/mm^2
τ_b= $0,58\ \sigma_y (1-0,58\ \sigma_y/(4\ K_s\ E\ (t/b)^2)$ = 165,0 N/mm^2

F = 0,67 (stress factor)

σ_b max =	97N/mm2 N/mm^2	<	σ_{bb} * F =	200,5 N/mm^2
σ_c max =	8N/mm2 N/mm^2	<	σ_{cb} * F =	86,9 N/mm^2
τ max =	0 N/mm^2	<	τ* F =	110,6 N/mm^2

$(\sigma_{maxb}/\sigma_{bb})^2 + (\sigma_{maxc}/\sigma_{cb})^2 + (\tau_{max}/\tau_b)^2$ = 0,11 < F

5.3.1 COLUMN BOLTING

For this verification **as conservative action** we use the same metod of the base bearing bolting.
According to the API 2C rules,the ultimate strenght criteria of the bolts have to satisfied the following:

P= 4*M/Nb*D - H/Nb = 535,5 KN
where:
P = max allowable force in one bolt
M= 3,75 x total overtur factored mom. = 11970 KNm
Nb = number of bolts = 48
D=bolts circle diameter = 1,823 m
H = 3,75xvertical factored force = 2343 KN
A= resistance section of M30 = 561mm2
σ_{bolt}= 955N/mm2
σ_{all} = 1000N/mm2 > σ_{bolt}

So we can say that the bolts **class 10.9** is acceptable.
Flange plate edge column calculation not necessary because is the same of base bearing .

5.4.0 SEISMIC

According to API rules we can assume that due to a very low probability of simultaneus occurence of a design siemic event at the time of the crane being used for a maximum rated lift, a reduced crane load may be considered simultaneous with the design seismic event.

In absence of such a study, a load producing 2/3 0f the rated crane overturning moment capacity shall be cosidered.

The max acceleration of 0,125 g of the site condition is lower than the load reduction that we have taking in account the upmentioned consideration .

Furthermore the stress in the various structure members has a value with a sigificant margin in front of the allowable stress that in this case is the exceptional one.

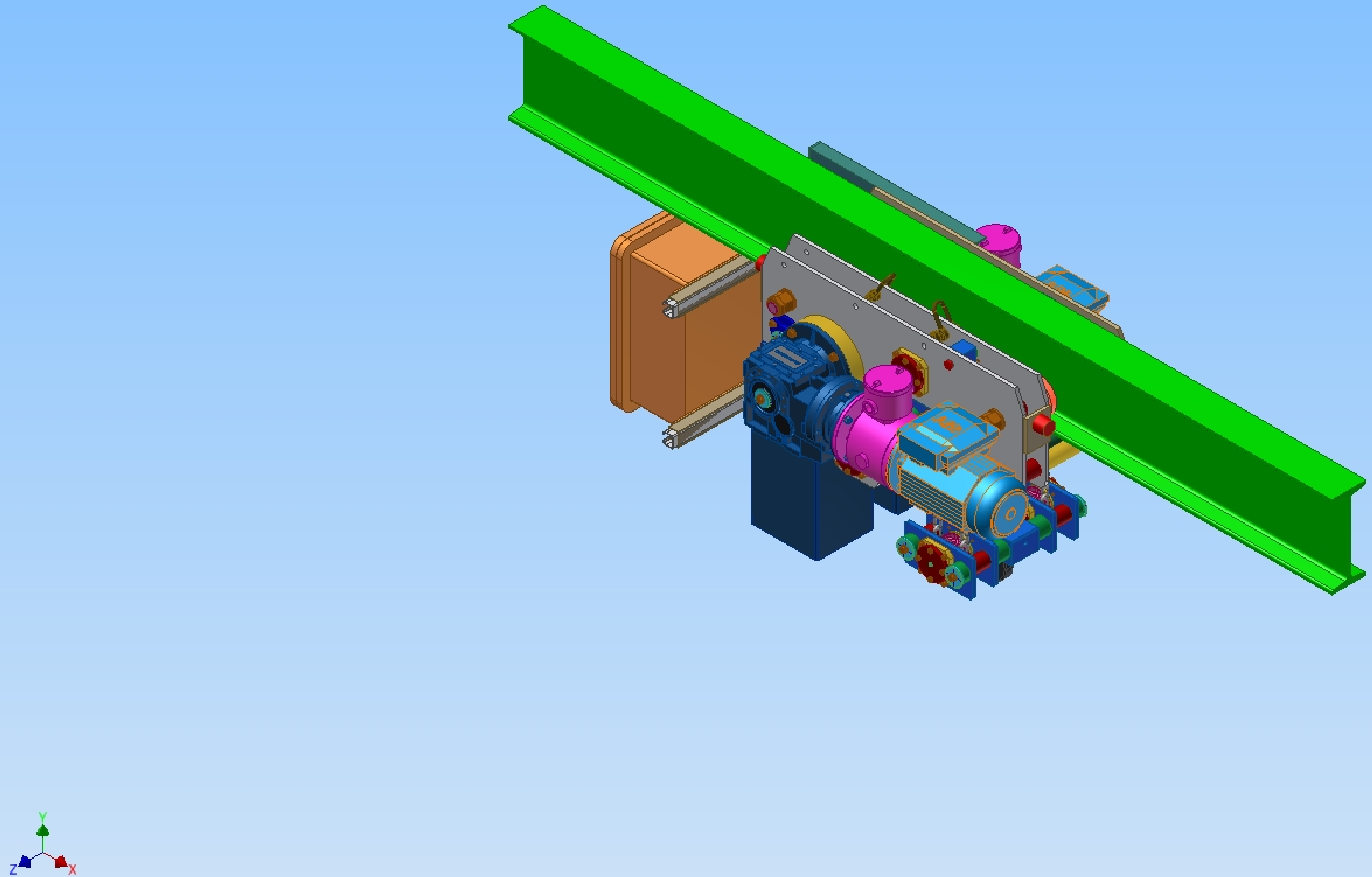
So, with this consideration we can assume satisfied this verification.

5.5.0 TRANSPORTATION CONDITION

The transportation loads taken in account are:

Barge moyion imput:	Accelerations		
	X(G)	Y(G)	Z(G)
Roll = 12,5 deg			
Pitch = 5,725 deg	+/- 0,23	+/- 0,49	+/- 0,23
Heave = 0,255g = 2,5 m /sec^2			
Wind = 145 Km/h			

The stresses in the structues af the cranes, with the actions above mentioned are max 68N/mm^2
Also the base bearing and column bolting are understessed in front of the working verification.
The max stress in the bolt is 180 N/mm2 according to API 2C rules.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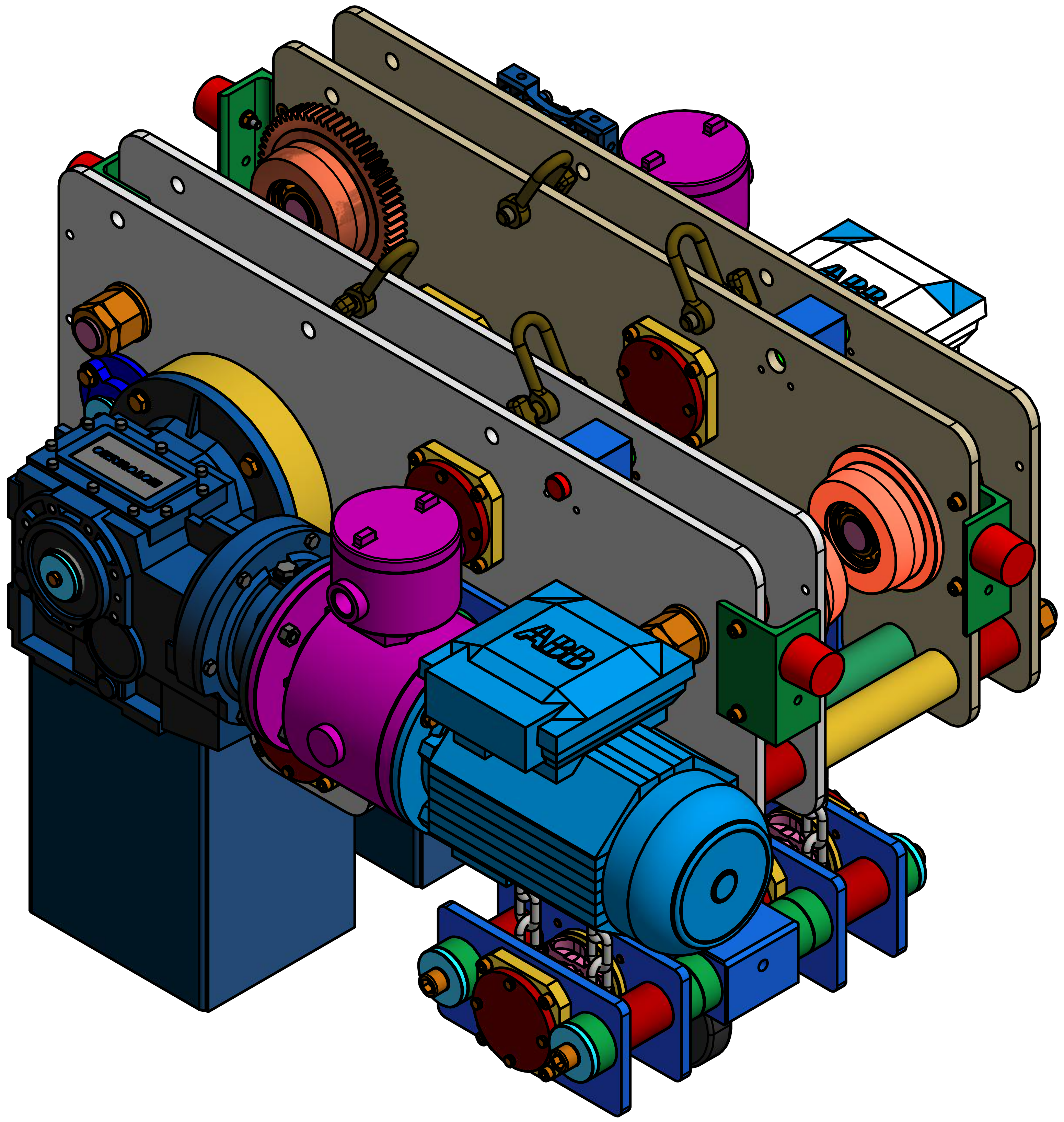
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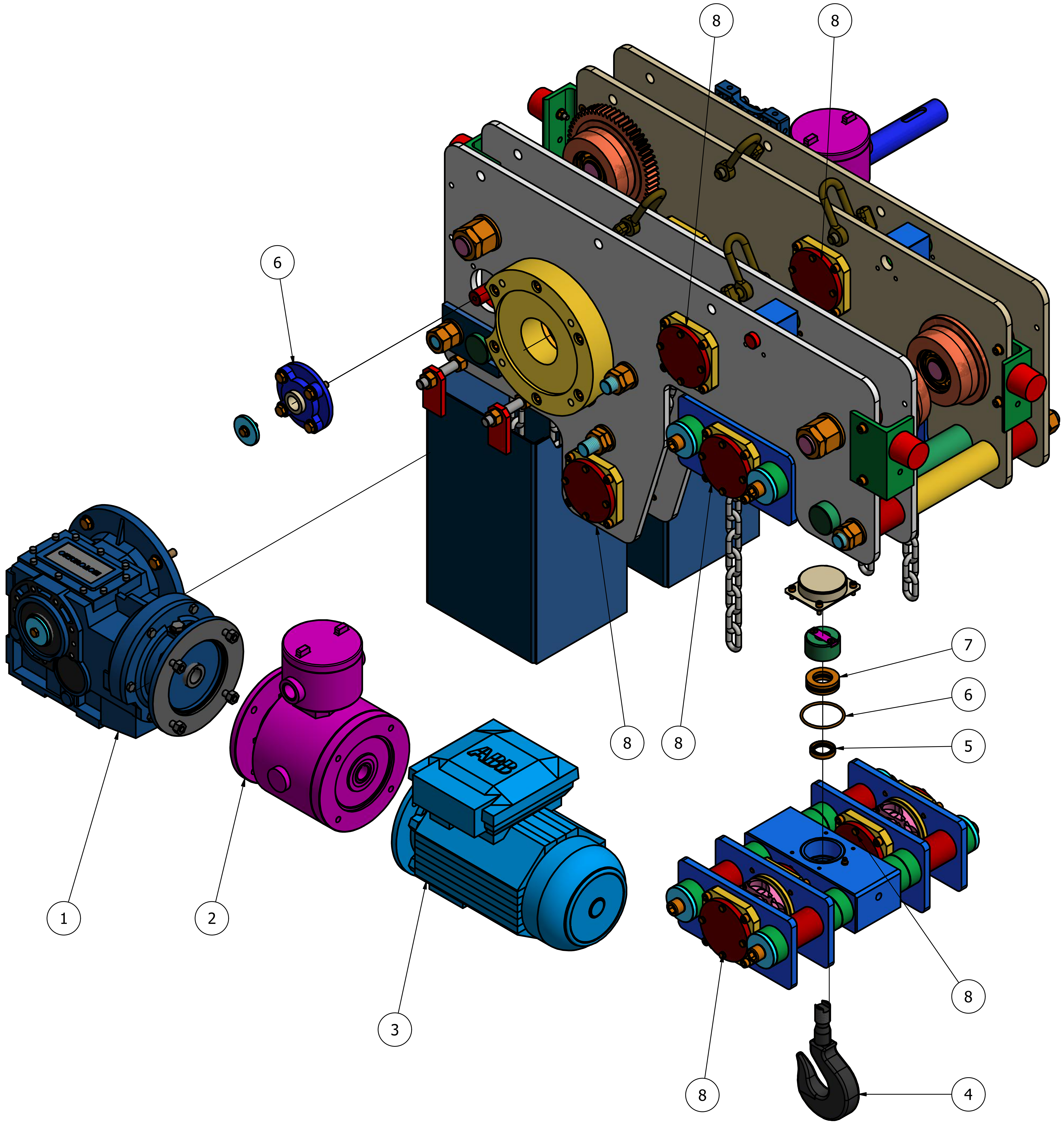
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