ISTANBUL TECHNICAL UNIVERSITY
FACULTY OF AERONAUTICS AND ASTRONAUTICS

Design of Machine Elements Course, Term Project

TRANSMISSION SYSTEM DESIGN FOR COLIBRI

Engine
Turbomeca Arrius 2F
Aircraft
EUROCOPTER EC120B Colibri

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Introduction and Contents:

Design of a transmission system which is build on to reduce or increase of the torque and rotational speed is a main problem of power systems. Especially in aircrafts, transmission systems are the main elements between engine and other elements. This report includes the design and the explanation of the steps of design of a transmission system of a helicopter.

Shortly, system contains two basic axis' which are connector between engine and main rotor. Its mission is to reduce the rotational speed and increase the torque. At the same time, system provides transformation of direction of movements.

Report is three part, and the parts are named as below:

A) First Parameters and Prelimenary Design
B) Calculations of dimension and other parameters for pinion and gears
C) Calculations of dimension and other parameters for shafts

In the first part, basic parameters (engine type, helicopter type, rotational speed, material properties and specialities etc.) have been chosen or calculated.

In the second part, pinions and gears which are main part of system have been designed.

In the third part, the bears and shafts driving the gears and the faderss have been designed.

It is seen that the report is simple and easy to understand. You can find technical drawings at the end of report.

A) First Parameters and Preliminary Design: Engine and Helicopter

Motor has been selected as “Turbomeca Arrius 2F” which is powerful and enough and functionally for using in EC120B Colibri Civilian Helicopter produced by Eurocopter. It is assumed that helicopter rotor has a rotational speed 600 rpm. This asserts comes from that is an average rotor speed of light utility helicopters.

Appendix 1.

Engine rating: 6000 rpm exit shaft speed,
It’s wanted to dip in 600 rpm rotor speed.
So the reduction ratio is
i= 6000/600 = 10
Engine shaft Torque:

$$M_{d1} = 9550 \times \frac{P}{n} (N.mm)$$

Because of the usefulness of the helical gear and bevel gear on transmission, system is designed as two different mechanism. Above, reduction ratio has been found 10. To catch it, it is decided the reduction ratio of the first mechanism is 4 and the other is 2,5. It is seen that the multiply of this numbers equals to 10.

It must be remembered that, system changes the direction of turning axis. It means, second mechanism takes power from horizontal axis and give it to vertical axis.

When it is compared with other elements, it is seen that pinion takes the big majority of load and this situation makes the pinion critical element. So, pinion is the main parameter of the design and it is the reason why all calculations made, according to pinion.

Formulas for Damage Analysis and Design:

There are three possible type of damages in a gear system:

1. Fracture of a tooth
2. Pitting Failure
3. Scoring

For each of these damages, there are formulas to calculate the m_n coefficient, which is an important module for design.

According to fracture of a tooth:

$$m_n = 0{,}6 \cdot \sqrt[3]{\frac{K_A \cdot K_v \cdot M_{d1} \cdot \gamma \cdot \cos\beta_0}{z_1 \cdot \sigma_{em} \cdot \varepsilon_\alpha \cdot \psi}} \quad [mm] \tag{1.1}$$

According to pitting Failure

$$m_n = 0{,}9 \cdot \sqrt[3]{\frac{K_A \cdot K_v \cdot M_{d1} \cdot E(i+1) \cdot \cos^4\beta_0}{z_1^2 \cdot P_{em}^2 \cdot i \cdot \psi}} \quad [mm] \tag{1.2}$$

Diameter of addendum:

$$d_{Bp} = m_n\left(\frac{N}{\cos\beta} + 2\right) \tag{1.3}$$

Diameter of dedendum:

$$d_{Tp} = m_n\left(\frac{N}{\cos\beta} - 2{,}4\right) \tag{1.4}$$

Pitch diameter:

$$d_p = m_n\left(\frac{N}{\cos\beta}\right) \tag{1.5}$$

Width of helical gear:

$$b = \psi \times \pi \times m_n \tag{1.6}$$

B) Calculation of Dimensions and other parameters for pinion and gears.

Coefficients which are used in design steps(The results are showed in table 1.1):

(Values are selected respecting to α= 20°)

K_A	: Contact load factor
KV	: Dynamic load factor. (its initial value is taken as 1 and after rotational velocity is calculated, it is compared the new Kv. Than less errored Kv is choosen with an iterational way)
N_1	: Teeth number of pinion
N_2	: i N_1
B	: Helix angle (It is choosen as 15°)
Zeq	: $N_1 / \cos^3(\beta)$:
γ = (form factoria)	: By table 2 respecting to Zeq .
$\varepsilon_{\alpha n}$	: It is readen by table 3 , respecting to β .

Material Choosen in table 5 is 13NiCr18;

σ_{em}	: 220N /mm²
P_{em}	: 1400N /mm²
E	: 200000 N/mm²
Ψ	: (Table 7)

1. Calculations for pinion: (Table 1.1)

Parameters for pinion	
K_a (Contact load factor)	1,0000
K_v (Dynamic factor)	1,1165
n_p (rpm)	6000,0000
T (Torque N.mm)	512516,6667
γ (Form factor)	8,6720
β (Helix angle)	15,0000

Cosβ	0,9659
N_1	15,0000
σ_{em}	220,0000
ε_α	1,6500
ψ	12,0000
P_{em}	1400,0000
I	4,0000

With the formulas (1.1),(1.2),(1.3),(1.4),(1.5),(1,6) and the tables in appendix 2, $M_{n\ \text{fracture of a tooth}}$, $M_{n\ \text{pitting failure}}$ are calculated:

m_n (According to pitting failure)	2,1826
m_n (According to fracture of a tooth)	2,5117
m_n (mm) - Choosen	3,0000

For the security, mn is choosen 3.
And dimensions of the pinion is calculated as below:

Dimensions of Pinion	
d_{Bp} (addendum)	52,5874
d_{Tp} (dedendum)	39,3874
d_p (pitch circle)	46,5874
b (width)	113,0973
V (mm/sn)	14635,8722
Distance between the axes	116,4686

2. Helical Gear Mechanism:

Material is same with pinion but some parameters change for gear according to pinion.
Different values with pinion is highlighted on the table 1.2 below:

Calculation of module for gear	
K_a (Contact load factor)	1,0000
K_v (Dynamic factor)	1,1165
n_g (rpm)	**1500,0000**

T (Torque N.mm)	2050066,6668
γ (Form factor)	8,6720
β (Helix angle)	15,0000
Cosβ	0,9659
N_2	**60,0000**
σ_{em}	220,0000
ε_α	1,6500
Ψ	12,0000
P_{em}	1400,0000
I	4,0000

Here is the dimensions for the helical gear:

Dimensions of Gear	
N_2	60,0000
N_2 (Number of teeth)	60,0000
i_{new}	4,0000
d_{Bg} (addendum)	192,3497
d_{Tp} (dedendum)	179,1497
d_g (pitch circle)	186,3497
b (width)	113,0973
V (mm/sn)	14635,8722

3. Bevel Gear Mechanism:

For Bevel Gear Pinion:

Calculation of module for pinion	
K_a (Contact load factor)	1,0000
K_v (Dynamic factor)	1,0188
n_p (rpm)	1500,0000
T (Torque N.mm)	2050066,6668
γ (Form factor)	8,6000
β (Helix angle)	0,0000
Cosβ	1,0000
N_1	17,0000

σ_{em}	220,0000
ε_α	1,7300
F_w	12,0000
P_{em}	1400,0000
I	2,5000

m_n (According to pitting failure)	3,3097
m_n (According to fracture of a tooth)	3,6833
m_n (mm)	4,0000

Dimensions of Pinion	
d_{Bp} (addendum)	76,0000
d_{Tp} (dedendum)	58,4000
d_p (pitch circle)	68,0000
b (width)	150,7964
V (mm/sn)	5340,7075
Distance between the axes	120,0000

For Bevel Gear:

Calculation of module for gear	
K_a (Contact load factor)	1,0000
K_v (Dynamic factor)	1,0188
n_g (rpm)	593,0233
T (Torque N.mm)	5185462,7454
γ (Form factor)	8,6000
β (Helix angle)	0,0000
Cosβ	1,0000
N_2	43,0000
σ_{em}	220,0000
ε_α	1,7300
F_w	12,0000
P_{em}	1400,0000
I	2,5000

m_n (According to pitting failure)	2,4291
m_n (According to fracture of a tooth)	3,6833

Dimensions of Gear	
N_2	42,5000
N_2 (Number of teeth)	43,0000
i_{new}	2,5294
d_{Bg} (addendum)	180,0000
d_{Tp} (dedendum)	162,4000
d_g (pitch circle)	172,0000
b (width)	150,7964
V (mm/sn)	5340,7075

B) Calculation of Dimensions and other parameters for shafts

Shaft 1(Engine Shaft):

T (N.mm): 512516,6667
d (mm): 46,5870
β: 15,0000
cosβ: 0,9659
sinβ: 0,2588
α: 20,0000
sinα: 0,3420
cosα: 0,9397

Formulas for Helical Gears:

$$F_t = 2\left(\frac{M_d}{d}\right)(N)$$

$$F_n = \frac{F_t}{\cos\alpha \times \cos\beta} \text{ (N)}$$

$$F_r = F_t\left(\frac{\sin\alpha}{\cos\alpha \times \cos\beta}\right) \text{(N)}$$

$$F_a = \frac{F_t \times \cos\alpha \times \sin\beta}{\cos\alpha \times \cos\beta} \text{ (N)}$$

According to this formulas:

F_t (N):	22002,56152
$F_{N:}$	24240,61586
$F_{r:}$	8290,778911
$F_{a:}$	5895,56859

Calculation of forces and moments:

Chosen Shaft Length: 110 mm

By	2896,9492
Bz	11001,2808
Bx	0,0000
Ax	5895,5686
Ay	5393,8297
Az	11001,2808

Mz: The moment coming to gear from A	
M	296660,6335
Mz: The moment coming to gear from B	
M	159332,2066

My: The moment coming to gear from A	
M	605070,4417
My: The moment coming to gear from B	
M	605070,4417

Critical Radius Calculation:

$$\sigma_{eg} = \frac{M_e}{\frac{\pi d^3}{32}} = \frac{\sqrt{M_z^2 + M_y^2}}{\frac{\pi d^3}{32}}$$

Σem	240,0000
Σeg	240,0000
dmil(mm)>	30,5814
dmil(mm)=	32,0000

Bear:

Bear Calculation:

d:	30,0000
D:	90,0000
B:	23,0000
C:	33500,0000
Co:	24000,0000
Fa/Co:	0,2456

According to Fa/Co, e is chosen as 0,37 with interpolation method from table 8.X *("Makine Elemanları Bağlama Elemanları Konstrüksiyon ve Hesap",* Gediktaş M. 1976)

Ftotal:	12252,4111
Fa/Ftot:	0,4812

Fa/Ftot > e,

According to this criteria, X and Y is chosen from table 8.X with interpolation method:

x:	0,5600
y:	1,0150
Fson:	6861,5995

Fson < Co and this result sharply asserts that this bear is usable.

Calculation of Fiather 1

We found above dmil = 32 mm. Through this information, from the *"Makine Elemanları Bağlama Elemanları Konstrüksiyon ve Hesap",* Gediktaş M. 1976, we get table below.

Fiather Calculation (1st shaft)	
b	10
h	8
t1	4,7
t2	2,8
l	4,745525
l chosen	5

Shaft 2(Engine – Rotor):

With using same methods and formulas with Shaft 1 calculation:

T (N.mm)	2050066,6668
d (mm)	68,0000
B	15,0000
Cosβ	0,9659
Sinβ	0,2588
A	20,0000
Sinα	0,3420
Cosα	0,9397

F_t (N)	60296,07844
F_N	66429,26889

F_r	22720,14807
F_a	16156,28552

Length of shaft:	480,0000
By:	10215,6705
Bz:	30148,0392
Bx:	0,0000
Ax:	16156,2855
Ay:	12504,4776
Az:	30148,0392

Mz: The moment coming to gear from A

M 3001074,6217

Mz: The moment coming to gear from B

M 2451760,9139

My: The moment coming to gear from A

M 7235529,4122

My: The moment coming to gear from B

M 7235529,4122

Σem	240,0000
Σeg	240,0000
dmil(mm)>	69,2750
dmil(mm)=	70,0000

Calculation of Bear

d	65,0000
D	120,0000
B	29,0000
C	58500,0000
Co	45500,0000
Fa/Co	0,3551

With Interpolation: e = 0,3994

Ftotal	32638,4165
Fa/Ftot	0,4950

X	0,5600
Y	1,1160
Fson	18277,9095

Fson>Co shows that bear is usable.

For Bevel Gears, calculations may show differences then helical gears in some way.

$$F_t = 2\left(\frac{M_d}{dav}\right)(\mathrm{N})$$

$$F_r = F_t(\tan\alpha \times \cos\gamma)(\mathrm{N})$$

$$Fa = F_t(\tan\alpha \times \sin\gamma)(\mathrm{N})$$

Calculation of Fiather 2

We found above dmil = 70 mm. Through this information, from the *"Makine Elemanları Bağlama Elemanları Konstrüksiyon ve Hesap",* Gediktaş M. 1976, we get table below.

Fiather Calculation (2nd shaft)	
b	20
h	12
t1	7,40
t2	3,9
l	5,759423
l chosen	6

Rotor Shaft:

dp(mm)	102,0000
dg(mm)	306,0000
T(Nmm)	5185462,7464
I	2,5000
tanΦ	0,3640
Γ	68,1986
Γ	21,8014
Sinγ	0,3714
Cosγ	0,9285
sinΓ	0,9285
cosΓ	0,3714

Force analysis for bevel system	
wt(N)	33891,9134
wr(N)	4581,3445
wa(N)	11453,3613

Length of shaft : 400 mm

l : 300 mm

By	267,2451
Bz	45189,2178
Bx	11453,3613
Ax	0,0000
Ay	4314,0994
Az	-11297,3045

Mz: The moment coming to gear from A	
M	1294229,8281
Mz: The moment coming to gear from B	
M	1294229,8281

My: The moment coming to gear from A	
M	-3389191,3375
My: The moment coming to gear from B	
M	-3389191,3375

Critical Radius:

Σem	200,0000
Σeg	200,0000
dmil(mm)>	56,9563
dmil(mm)=	60,0000

D	55,0000
D	140,00
B	33,0000
C	76500,0000
Co	63000,0000
Fa/Co	0,1818

With interpolation:

e:	0,3359
Fr:	46618,0748
Fa/Fr:	0,2457
X	1,0000
Y	0,0000
Fes	46618,0748

Fes < Co drives the system usable.

Calculation of Fiather 3

We found above dmil = 60 mm. Through this information, from the *"Makine Elemanları Bağlama Elemanları Konstrüksiyon ve Hesap",* Gediktaş M. 1976, we get table below.

Fiather Calculation (3rd shaft)	
b	18
h	11
t1	6,8
t2	3,5
l	18,64604
l chosen	19

Web sitesi için sadeleştirildiğinden ekler kaldırılmıştır