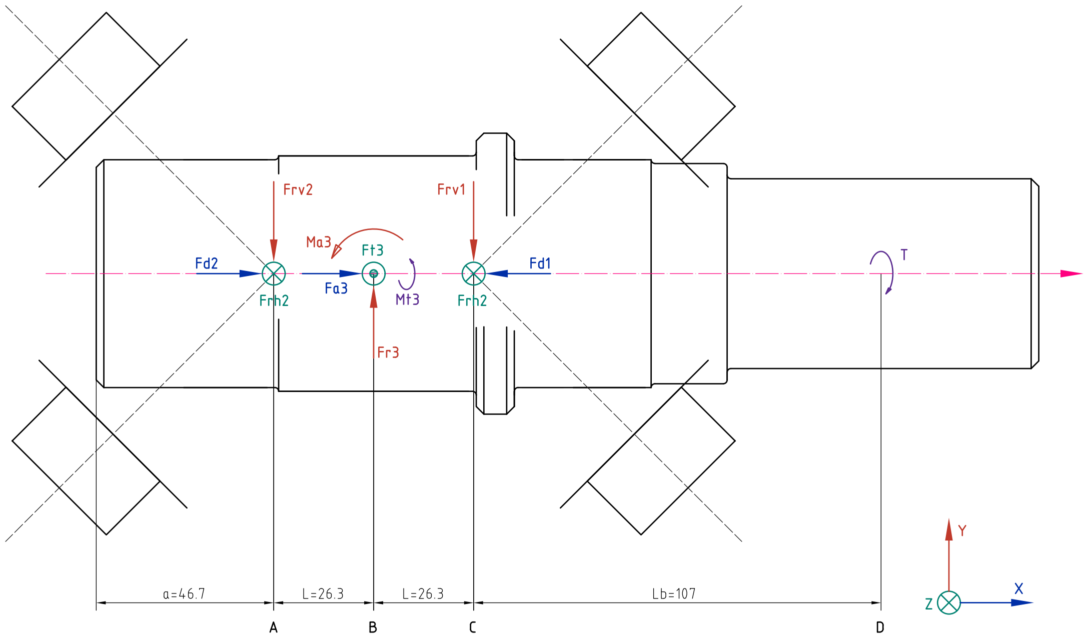


对于 oAxis 我们选用 7212B 轴承，查得轴承支点距背面的距离 $a = 46.7mm$，

并且安装方式为正装，对于 H 面有：

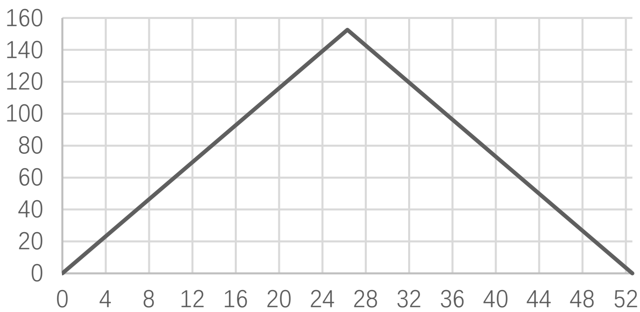
$$\begin{cases} \sum F_H = 0 \ \Rightarrow \ -F_{t3} + F_{rh1} + F_{rh2} = 0 \\ \sum M_{HA} = 0 \ \Rightarrow \ -F_{t3} * L + F_{rh1} * 2L = 0 \end{cases}$$

计算得 H 面的分支座反力：

$$F_{rh1} = F_{rh2} = 5.801400825414522KN$$

则 H 面轴的弯矩方程为：

$$\begin{cases} M(x) = F_{rh2} * x & (0 \le x \le L) & (KN * mm) \\ M(x) = F_{rh1} * x - F_{t3} * (x - L) & (L \le x \le 2L) & (KN * mm) \end{cases}$$



对于 V 面有：

$$\begin{cases}\sum F_V = 0 \Rightarrow F_{r3} - F_{rv2} - F_{rv1} = 0 \\ \sum M_{VA} = 0 \Rightarrow M_{a3} + F_{r3} * L - F_{rv1} * 2L = 0 \\ M_{a3} = F_{a3} * \dfrac{d}{2}\end{cases}$$

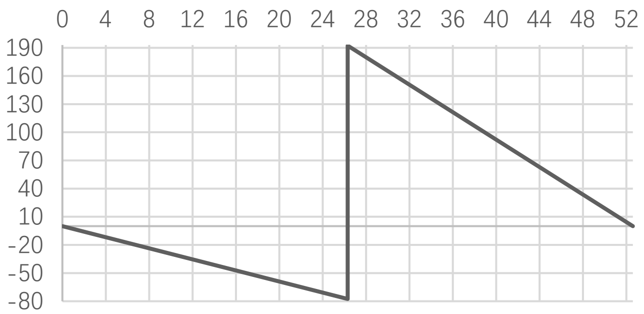
计算得 V 面的分支座反力:

$$F_{rv1} = 7.3132530478416236KN$$

$$F_{rv2} = -2.950341711024021KN$$

则 V 面的弯矩方程为:

$$\begin{cases}M(x) = F_{rv2} * x & (0 \le x \le L) & (KN * mm) \\ M(x) = F_{rv1} * (2 * L - x) & (L \le x \le 2L) & (KN * mm)\end{cases}$$

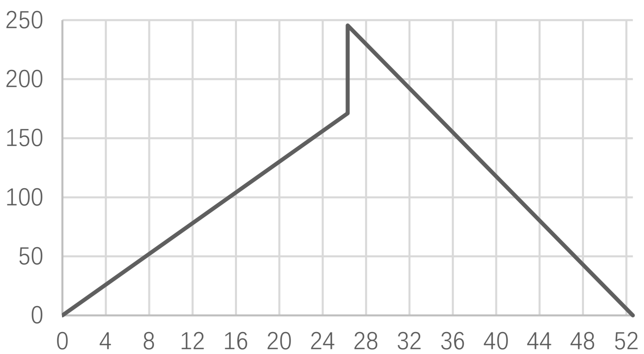


则轴承总支座反力为:

$$F_{r1} = \sqrt{F_{rv1}^2 + F_{rh1}^2} = 9.33487662901257KN$$

$$F_{r2} = \sqrt{F_{rv2}^2 + F_{rh2}^2} = 6.5085150187218925KN$$

输出轴总弯矩图(矢量合成)为:

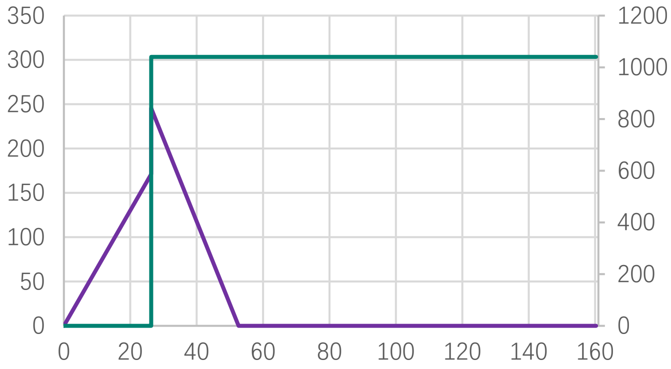


$$M_A = 0;\ M_{BL} = 74.982KN * mm;\ M_{BR} = 344.915KN * mm;\ M_C = 0$$

对于输出轴的 BD 段，扭矩为:

$$-T = M_{t3} = F_{t3} * \frac{d}{2} = 1040.346480238948KN * mm$$

输出轴的弯矩和扭矩图为:



根据弯扭图，输出轴(oAxis)危险截面为 B 截面(26.3mm 处，直径为 62mm)，由于 B 截面为双键槽([b, h, l, t] = [18, 11, 50, 4.4])180°均布，的轴截面因此抗弯截面系数为:

$$W_B = \frac{\pi d^3}{32} - \frac{bt(d-t)^2}{d} = 552640.6466053243mm^3$$

输出轴材料与热处理方式为 45 调质，查得许用弯曲应力 $[\sigma_{-1}] = 60MPa$，轴所受弯矩所产生的弯曲应力为对称循环变应力，而扭矩所产生的扭转切应力为脉动循环变应力，故$\alpha = 0.6$，则轴的弯扭合成强度条件为:

$$\sigma_{ca} = \frac{\sqrt{M^2 + (\alpha T)^2}}{W} = \frac{\sqrt{(M_{BR} * 10^3)^2 + (\alpha M_{t3} * 10^3)^2}}{W_B} = 1.2137MPa < [\sigma_{-1}]$$

输出轴第三强度理论非常安全.

对于输出轴轴承，设计期望寿命为 3 年 2 班制工作($[h] = 5840h$). 对于 7212B 轴承查表得$F_d = eF_r(e = 1.14)$则:

$$F_{d1} = 1.14 * F_{r1} = 10641.759357074328N$$

$$F_{d2} = 1.14 * F_{r2} = 7419.707121342958N$$

根据最大值法，轴承派生轴向力为:

$$F_{d2} - F_{a3} = 4409.196868022452N < F_{d1} = 10641N$$

$$F_{a1} = F_{d1} = 10641.759357074328N$$

$$F_{d1} + F_{a3} = 13652.269610394835N > F_{d2} = 7419N$$

$$F_{a2} = F_{d1} + F_{a3} = 13652.269610394835N$$

对于轴承 1，径向动载系数 X 与轴向动载系数 Y 为:

$$\frac{F_{a1}}{F_{r1}} = 1.14 = e = 1.14$$

$$X_1 = 1; \quad Y_1 = 0;$$

对于轴承 2，径向动载系数 X 与轴向动载系数 Y 为:

$$\frac{F_{a2}}{F_{r2}} = 2.098 > e = 1.14$$

$$X_2 = 0.35; \quad Y_2 = 0.57;$$

根据工况条件，轴承运行中冲击较小，因此$f_d = 1.1$，则两轴承动载荷 P 为:

$$P_1 = f_d * (X_1 * F_{r1} + Y_1 * F_{a1}) = 10268.364291913827N$$

$$P_2 = f_d * (X_2 * F_{r2} + Y_2 * F_{a2}) = 11065.75132792549N$$

因为$P_1 < P_2$，按 2 轴承的受力情况计算(对于球轴承$\varepsilon = 3$)(C 取 56000N):

$$L_h = \frac{10^6}{60n_m}\left(\frac{C}{P_m}\right)^{\varepsilon} = \frac{10^6}{60 * oAxis.n}\left(\frac{C}{P_2}\right)^3 = 49481.948h > [h] = 17520h$$

输出轴轴承寿命符合设计预期.

输出轴与齿轮的连接使用 GB/T 1096-2003 键 B 18*11*50 (轴、齿轮、键的材料均为 45) ([b, h, l, t] = [18, 11, 50, 4.4])，轴径 d 为 62mm，两键 180°均布连接工作方式为静连接，受轻微冲击，故$[\sigma_{bs}] = 100\sim120\text{MPa}$，校核该键连接:

$$\sigma_{bs} = \frac{4000 * T(N * m)}{h * work.l * d} = \frac{4000 * M_{t3}(KN * mm)}{h * 1.5 * l * d} = 81.357 < [\sigma_{bs}]$$

齿轮与输出轴的键连接可靠.

输出轴与联轴器的连接使用 GB/T 1096-2003 键 A 14*9*70 (轴、联轴器 GYSJ1A50×84、键的材料均为 45) ([b, h, l] = [14, 9, 70])，轴径 d 为 50mm，两键 180°均布连接工作方式为静连接，受轻微冲击，故$[\sigma_{bs}] =$ 100~120MPa，校核该键连接:

$$\boldsymbol{\sigma_{bs} = \frac{4000 * T(N * m)}{h * work.l * d} = \frac{4000 * M_{t3}(KN * mm)}{h * 1.5 * (l - b) * d} = 110.0896} < [\boldsymbol{\sigma_{bs}}]$$

输出轴与联轴器的键连接可靠.