9

Power-Transmitting Turbomachines

Problem 1

The input power of a torque converter is 30 kW at 5000 rpm. Let the fluid flow from the turbine exit to the pump inlet be "axial flow," without any whirl component. Design the torque converter, so that the output torque is 1.5 times the input torque. (Hint: The data and solutions of all the solved and unsolved problems can be tabulated for an overall assessment or evaluation, so that the inlet and outlet diameters of the pumping unit can be assumed as a starting point for solution of this example. All other dimensions and angles can then be calculated.)

Solution: Data: P_p = 30 kW; N_p = 5000 rpm; $\alpha_{2t} = \alpha_{1p} = 90^o$, V_{u1p} = 0; V_{u2t} = 0, T_p/T_p = 1.5, η = 0.80.

To design all the dimensions and angles:

As a first step, it is possible to calculate the input torque, T_p by the equation for power,

$$P_p = \frac{2\pi N_p T_p}{60}$$

We have

$$\begin{aligned} T_p &= \frac{P_p \times 60}{2\ \pi\ N_p} \\ &= \frac{(30\times1000)\times60}{2\ \times\pi\ \times5000} \\ &= 57.3\,\text{N-m} \end{aligned}$$

Now, a table is prepared to record the complete design (data and solutions) of all the solved and exercise problems.

Table showing the detailed designs of earlier examples.

Example/Problem No.	9.3	9.4	9.5	9.2	9.3	9.4

D_{1p} (m)	0.18	0.18	0.3	0.075	0.24	0.2
D_{2p} (m)	0.25	0.25	0.4	0.11	0.32	0.3
B_{1p} (m)	0.02	0.02	0.02	0.01	0.02	0.02
B_{2p} (m)	0.015	0.015				
N_p (rpm)	1440	1440	1500	1000	3000	3000
T_p (N-m)	301	301	2298	1.875	1198	1049
P_p (kW)	45.44	45.44	361	0.195	376	329.6
β_{1p}	45°	45°	50°	60°	40°	40°
β_{2p}	90°	90°	80°	75°	50°	65°
V_{f1p} (m/s)	13.57	13.57		4.646	25	20
$\dot{m}$ (kg/s)	127.88	127.88	434	9.3	320	213.63
N_t (rpm)	1350	1036.9	1160	900	1280	4042

T_t (N-m)	301	376	2765	1.875	2386.6	662
P_t (kW)	42.6	40.9	335	0.1767	319.9	280
β_{1t}	85°	–	75°	81.85°		
β_{2t}	46.9°	54.25°	57°	63.81°		
T_p/T_p	1.0	1.25	1.2	1.0	1.99	0.63

From the table, the mean outlet diameter of pump may be assumed as 0.12 m. This results in blade velocity at pump outlet, U_{2p} , as given by

$$U_{2p} = \frac{\pi \times D_{1p} \times N_p}{60}$$
$$= \frac{\pi \times 0.12 \times 5000}{60}$$
$$= 31.416 \text{ m/s}$$

Corresponding to this blade velocity, the flow velocity, is assumed as $V_{f2p} = V_{f1p} = V_{f2t} = V_{f2t} = 15$ m/s.

Also, a blade outlet angle, β_{2p}, is assumed as 70° (backward bent).

Velocity triangle at blade outlet is drawn now as shown in Fig. 9.1.

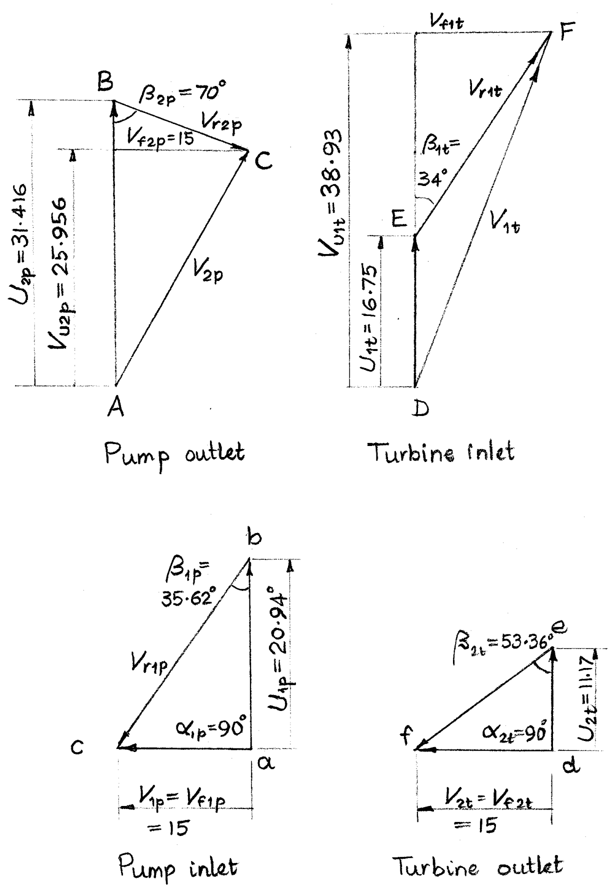

Figure 9.1 Velocity triangles.

From the velocity triangle,

$$\begin{aligned} V_{u2p} &= U_{2p} - \frac{V_{f2p}}{\tan 70} \\ &= 31.416 - \frac{15}{\tan 70} \\ &= 31.416 - 5.46 \\ &= 25.956\,\text{m/s} \end{aligned}$$

The torque is given by

$$T_p = (r_{1p}V_{u1p} - r_{2p}V_{u2p})$$

That is,

57.3 = $\dot{m}$ (– 0.06 × 25.956)   (because V_{u2p}= 0)

Solving for $\dot{m}$, we get $\dot{m}$ = 36.8 kg/s

We also have expression for $\dot{m}$ = $\pi D_{2p} B_{2p} V_{f2p} \rho$

Substituting the values,

$$36.8 = \pi \times 0.12 \times B_{2p} \times 15 \times 850$$

Therefore, blade width at outlet of pump,

$$B_{2p} = \frac{36.8}{\pi \times 0.12 \times 15 \times 850}$$
$$= 0.00766\,\text{m}\,(= 0.766\,\text{cm})$$

Also, let the mean inlet diameter of pump be 0.08 m (8 cm).

Writing expression for mass flow rate,

$$\dot{m} = D_{1p} B_{1p} V_{f1} \rho$$

Blade width at inlet,

$$B_{1p} = \frac{36.8}{\pi \times 0.08 \times 15 \times 850}$$
$$= 0.0115\,\text{m} = 1.15\,\text{cm}$$

At the inlet of the pump, the blade velocity, U_{1p}, is given by

$$U_{1p} = \frac{\pi \times 0.08 \times 5000}{60}$$
$$= 20.94\,\text{m/s}$$

The inlet velocity triangle is shown in Fig. 9.2. Because the entry of fluid is without whirl component, α_{1p} is 90°, V_{u1p} = 0 and blade angle is given by

$$\begin{aligned}\beta_{2p} &= \tan^{-1}\left(\frac{V_{f1p}}{U_{1p}}\right)\\ &= \tan^{-1}\left(\frac{15}{20.94}\right)\\ &= 35.62^\circ\end{aligned}$$

Now assume that efficiency of the device is 80%.

The power at the output shaft is

$$P_t = P_p \times 0.8 = 30 \times 0.8 = 24\,\text{kW}$$

Output torque is required to be 1.5 times the input torque; that is,

$$\begin{aligned}T_t &= T_p \times 1.5 = 57.3 \times 1.5\\ &= 85.95\,\text{N-m}\end{aligned}$$

We have

$$P_t = \frac{2\pi N_t T_t}{60}$$

Therefore output speed,

$$\begin{aligned}N_t &= \frac{P_t \times 60}{2\ \pi\ T_t} = \frac{24 \times 1000 \times 60}{2\ \times \pi\ \times 85.95}\\ &= 2666\,\text{rpm}\end{aligned}$$

Blade velocity at the inlet to the turbine is

$$U_{1t} = \frac{\pi D_{1t} N_t}{60}$$
$$= \frac{\pi \times 0.12 \times 2666}{60}$$
$$= 16.75 \text{ m/s}$$

We have the expression for torque as

$$T_t = \dot{m}(r_1 V_{u1t} - r_2 V_{u2t})$$
$$\dot{m} r_1 V_{u1t} \quad (\text{because } V_{u2t} = 0)$$

That is, $85.95 = 36.8 \times 0.06 \times V_{u1t}$

or, $$V_{u1t} = \frac{85.95}{36.8 \times 0.06}$$
$$= 38.93$$

The velocity triangle at both inlet and outlet of turbine is also drawn in Fig. 9.2.

From the inlet triangle,

$$\beta_{1t} = \tan^{-1}\left[\frac{V_{f1t}}{V_{u1t} - U_{1t}}\right]$$
$$= \tan^{-1}\left[\frac{15}{38.93 - 16.75}\right]$$
$$= 34°$$

From the output triangle,

$$\beta_{2t} = \tan^{-1}\left[\frac{V_{f2t}}{U_{2t}}\right]$$
$$= \tan^{-1}\left[\frac{15}{11.17}\right]$$
$$= 53.36°$$

The final solution is:

Pump inlet diameter, D_{1p} = 8 cm

Pump inlet angle, β_{1p} = 35.62°

Pump inlet blade width, B_{1p} = 1.15 cm

Pump outlet diameter, D_{2p} = 12 cm

Pump outlet angle, β_{2p} = 70°

Pump outlet blade width, B_{2p} = 0.766 cm

Turbine inlet blade angle, β_{1t} = 34°

Turbine outlet blade angle, β_{2t} = 53.36°

Output speed, N_t = 2666 rpm

Output torque, T_t = 85.95 N-m

Fluid flow rate, $\dot{m}$ = 36.8 kg/s

Comments on solution:

1. Since this is an overall design problem, more trials can be done for different solutions, assuming that the problem can have such solutions. This is toward optimization of the design.
2. The input power (30 kW) and speed (5000 rpm) suggest that the torque is on the lower side, for the given speed.
3. Totally five assumptions are made, starting with D_{2p} = 12 cm. If higher diameter is assumed, the blade velocity would also be higher; and for the given torque, the flow velocity may be too less. This can be ruled out by the shape of the velocity triangle at pump outlet, where the blade velocity (U_{2p} = 31.416 m/s) may be still higher and flow velocity (V_{f2p} = 15) may be still lower, resulting in very small fluid angle (α_{2p}).
4. Blade outlet angle, β_{2p}, which is assumed as 70°, is reasonable. Lower values reduce the torque, higher values may affect the stability of the operation.
5. Efficiency is assumed as 80%, whereas in earlier examples, it was about 85%. This is because of very high speed (5000 rpm) at the input side; the losses may be comparatively more.
6. Trials can, however, be repeated with slight modification. But, it should be towards increasing of the fluid angles like α_{2p}, α_{2t}.
7. The torque ratio is 1.5. Again, this cannot be more, due to high speed at inlet side.
8. It should be very good exercise to conduct trials on the design; towards optimization.