6. Turbine Losses

Wake (후류)
<table>
<thead>
<tr>
<th></th>
<th>Turbine Losses and Performance Degradation</th>
<th>2</th>
</tr>
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<tbody>
<tr>
<td>2</td>
<td>Steam PathAudit</td>
<td>10</td>
</tr>
<tr>
<td>3</td>
<td>Profile Loss</td>
<td>19</td>
</tr>
<tr>
<td>4</td>
<td>Secondary Flow Loss</td>
<td>38</td>
</tr>
<tr>
<td>5</td>
<td>Leakage Loss</td>
<td>59</td>
</tr>
<tr>
<td>6</td>
<td>Supersaturation Loss</td>
<td>71</td>
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<tr>
<td>7</td>
<td>Moisture Loss</td>
<td>79</td>
</tr>
<tr>
<td>8</td>
<td>Exhaust Loss</td>
<td>90</td>
</tr>
<tr>
<td>9</td>
<td>Heat Transfer Loss</td>
<td>112</td>
</tr>
<tr>
<td>10</td>
<td>Mechanical and Electrical Loss</td>
<td>113</td>
</tr>
</tbody>
</table>
All of the available energy is not converted into work in the steam path because many losses are occurred in the steam path. Typical losses are profile loss, secondary loss, leakage loss, exhaust loss, and moisture loss etc.

The efficiency of the entire power plant is largely dependant on the efficiency of the energy conversion in the turbine.

It is important to minimize aerodynamic and steam leakage losses in the steam path.
Losses of Turbine Components

600 MW Reheat Steam Turbine

Steam Turbine Efficiency

<table>
<thead>
<tr>
<th>Year</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>1970</td>
<td>87%</td>
</tr>
<tr>
<td>1990</td>
<td>90%</td>
</tr>
</tbody>
</table>

Component Losses Related to Turbine Output

- Valve and admission
- Profile
- Shaft & inter-stage seal
- Extraction & exhaust
- Reheat pressure
- X-over pressure
- Moisture
- Mechanical
The nozzle and bucket profiles have been manufactured with a specified surface finish and with strict tolerance. When the steam turbine used for long time, the steam path is deteriorated, and both output and efficiency of the turbine will decrease.

Solid particle erosion, foreign object damage, water droplet erosion of LSB, fouling, seal wear are main causes of performance degradation.
Available energy is the difference in enthalpy from the stage inlet pressure to the enthalpy at the stage outlet at the same entropy.

\[ \eta_{\text{turbine}} = \frac{\text{Useful Energy}}{\text{Available Energy}} \]

- \( AB \): Isentropic expansion line
- \( AC \): Original expansion line
- \( AD \): New expansion line due to aging
- \( P_i \): Pressure at the inlet of turbine
- \( P_o \): Pressure at the outlet of turbine
- \( ds \): Increase of entropy due to the loss

Reduction in Useful Energy (Performance Degradation)

Increase in entropy due to aging

\( h \) vs. \( s \) diagram with points A, B, C, D, and P_i, P_o.
Performance Degradation

![Graph showing the performance degradation of HP and IP turbines between 1981 and 1985. The HP turbine efficiency decreases from 80% in 1981 to 70% in 1985, while the IP turbine efficiency decreases from 90% in 1981 to 85% in 1985. The graph indicates a clear trend of efficiency degradation over time.]

- **HP Turbine**: Efficiency decreases from 80% in 1981 to 70% in 1985.
- **IP Turbine**: Efficiency decreases from 90% in 1981 to 85% in 1985.
## Component Deterioration Potential

<table>
<thead>
<tr>
<th>Potential</th>
<th>Components</th>
<th>Causes</th>
</tr>
</thead>
<tbody>
<tr>
<td>high</td>
<td>LSB</td>
<td>• WDE</td>
</tr>
</tbody>
</table>
|           | HP-1 stage | • SPE - high temperature  
|           |            | • creep (bucket)  
|           |            | • high cycle fatigue - partial arc admission  |
|           | IP-1 stage | • SPE - high temperature  
|           |            | • creep (bucket)  |
| medium    | LSB & L–1 stage | • corrosion  |
|           | stages with drilled hole in the vane for lacing wires | • corrosion  |
|           | HP-2, 3 & IP-2, 3 | • SPE  |
|           | HP-1 & IP-1 diaphragm | • creep  |
|           | Nozzle box | • SPE  |
| low       | All other components and stages in the unit | |
Typical Turbine Location of Problems

- **SPE of Valves**
- **SPE of Blades**
- **Bearing Rubbing**
- **Seal Rubbing**
- **Fouling**
- **Stress Corrosion Cracking**
- **Rotor Bow** due to rubbing in transient operation such as during startup
- **WDE of LSB**
| 1 | Turbine Losses and Performance Degradation |
| 2 | Steam Path Audit |
| 3 | Profile Loss |
| 4 | Secondary Flow Loss |
| 5 | Leakage Loss |
| 6 | Supersaturation Loss |
| 7 | Moisture Loss |
| 8 | Exhaust Loss |
| 9 | Heat Transfer Loss |
| 10 | Mechanical and Electrical Loss |
Steam Path Audit

### Definition and Purpose
- Steam path audit is to assess the condition of the steam path to identify performance degradation of the unit and to point out the causes and locations of power and efficiency losses.
- The results of the audit identify specific problem areas and quantify the impact of the problems in order to assist the owner in making decisions whether to repair or replace steam path components.

### Benefits of Steam Path Audit
- Provides detailed inspection of steam path.
- Quantify heat rate, power and efficiency impact on component-by-component basis.
- Aid economic decisions during the repair outage.
- Prioritize maintenance decisions on a benefit-to-cost ratio.
- Quantify the quality of turbine repairs by performing a closing steam path audit.
- Provides excellent record/history of equipment conditions for future reference.
Areas Addressed in Steam Path Audit

1) Leakage
- nozzles
- buckets
- shaft end packings where rotors emerge from casing
- poorly fitting joints
- other miscellaneous leakages

2) Surface Roughness
- deposits
- corrosion
- solid particle erosion
- mechanical damage

3) Flow Blockage
- deposits
- foreign objects
- mechanical damage

4) Flow Path Modification
- solid particle erosion
- water droplet erosion
- mechanical damage

Pre-Outage Test  Opening Steam Path Audit  Upgrade and Maintenance  Closing Steam Path Audit  Post-Outage Test

Shutdown Disassembly

Startup Reassembly
Inspection

[ Clearance Measurement ]

[ Spill Strip ]

[ Solid Particle Erosion ]

[ Surface Roughness ]
Sample – 200 MW, 3000 rpm, 14710 kPa/535°C/535°C, 2 Flow LP

Power Loss Distribution

Opening Audit

- Interstage Packings: 1447.4 kW
- Tip Spill Strips: 4795.3 kW
- End Packings: 2740.4 kW
- Miscellaneous: 2506.6 kW
- Flow Path Damage: 4177.8 kW
- Flow Change Impact: 3477.5 kW
- Surface Roughness: 260.2 kW
- Cover: 282.4 kW
- Deposits: 1975.3 kW
- Hand calculations: 2077.8 kW
- Trailing Edge Thickness: 3241.1 kW
- Recovery: 6738.2 kW

Closing Audit

- Interstage Packings: 4570 kW
- Tip Spill Strips: 3241.1 kW
- End Packings: 3241.1 kW
- Miscellaneous: 1975.3 kW
- Flow Path Damage: 2077.8 kW
- Flow Change Impact: 2077.8 kW
- Surface Roughness: 2077.8 kW
- Cover: 2077.8 kW
- Deposits: 2077.8 kW
- Hand calculations: 2077.8 kW
- Trailing Edge Thickness: 2077.8 kW
- Recovery: 6738.2 kW
Sample –
200 MW, 3000 rpm, 14710 kPa/535°C/535°C, 2 Flow LP

Power Loss by Casing

Opening Audit

Closing Audit
Steam Turbine

6. Turbine Losses

Sample –
200 MW, 3000 rpm, 14710 kPa/535°C/535°C, 2 Flow LP

Power Loss Distribution – HP Turbine

Opening Audit

Closing Audit

Interstage Packings: 669.3 kW
Tip Spill Strips: 4270.6 kW
End Packings: 2627.5 kW
Miscellaneous Leakages: 1300.6 kW
Flow Path Damage: 1630.7 kW
Flow Change Impact: 262.1 kW
Surface Roughness: 1037.0 kW
Cover Deposits: 1083.8 kW
Hand Calculations: 1083.8 kW
Trailing Edge Thickness: 120.5 kW
Recovery: 4063.7 kW
Sample –
200 MW, 3000 rpm, 14710 kPa/535°C/535°C, 2 Flow LP

Heat Rate Loss Distribution

Opening Audit

Closing Audit
Sample –
200 MW, 3000 rpm, 14710 kPa/535°C/535°C, 2 Flow LP

Heat Rate Loss Distribution by Casing

Opening Audit

<table>
<thead>
<tr>
<th>Component</th>
<th>Loss (kJ/kWh)</th>
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<tr>
<td>HP</td>
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<td>LP Gen</td>
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<td>Total</td>
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Closing Audit

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<td>HP</td>
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<td>LP Gov</td>
<td>33.6</td>
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<td>LP Gen</td>
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<tr>
<td>Total</td>
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<td>Recovery</td>
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<td></td>
<td>Description</td>
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<td>-------------------------------------------------</td>
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<tr>
<td>1</td>
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**FIGURE 9.41** Schematic drawing of heat flux $q$ (or heat transfer coefficient $h$) distribution on an uncooled turbine blade (for a free-stream turbulence intensity)
Flow Separation

Let's assume that the static pressure increases from left to right in the above figure.

As the static pressure increases, the velocity profile becomes less full (angle $\alpha$ increase) and the boundary layer becomes thicker, so that where the static pressure is $p_{st\text{\(\circ\)}} > p_{st\text{\(\circ\)}}$ the gradient of the velocity profile is $\alpha_{\text{\(\circ\)}} > \alpha_{\text{\(\circ\)}}$.

In theory, the static pressure can increase up to a point where $\alpha = 90^\circ$.

Any further attempt to increase the pressure will result in flow reversal in the wall region, a phenomenon called “separation”.

\[ \tau_w = \mu \frac{\partial C}{\partial y} \]
\[ \tan \alpha_w = \frac{\partial C}{\partial y} \]
The primary cause of efficiency losses in an axial-flow turbine is the build-up of boundary layer on the blade and end walls.

The profile loss from the boundary layer build-up is due to a loss of stagnation pressure, which in turn is caused by a loss of momentum in the viscous fluid.

The magnitude of the profile losses is mainly governed by the free stream velocity, blade profile, blade roughness, Reynolds number, and surface areas.
The profile loss is the loss due to blade boundary layers, including flow separation, on blade surfaces and due to the wake through viscous and turbulent dissipation.

The energy contained in the working fluid is dissipated into heat within boundary layer and this increases the entropy and results in total pressure loss, even though the stagnation enthalpy is constant for adiabatic flow.

In addition, the non-uniform velocity profiles in both the boundary layer and the wake are smoothed out by viscous and turbulence effects.

Furthermore, the trailing vortex systems in the blade wake and its eventual mixing and dissipation give rise to additional losses.

The profile loss on a typical subsonic profile is mainly governed by the flow behavior on the suction side because of higher velocity and the occurrence of adverse pressure gradients (typically more than 80% of the skin friction loss occurs on the suction side).

The wake is a velocity defect generated by the boundary layers of the blade surfaces. If it is undisturbed by other blades it would move downstream along the direction of outlet-flow angle while decaying slowly over three or four chord lengths.
The flow on the suction side is characterized by the position of laminar-turbulent transition, transition length and diffusion ratio.

To reduce skin friction loss on the profile, the fraction of laminar surface length on the suction side has to be extended, because the friction loss is proportional to \((velocity)^2\) and \((velocity)^3\) for a laminar and turbulent boundary layer, respectively.

The difficulty of predicting transition of the boundary layer remains a major limitation to accurate prediction of the profile loss.

Increase of the laminar length implies a shift in profile loading towards the trailing edge and leads a higher diffusion ratio on the suction side.
New Profiles

- Conventional profile
- Improved profile

Profile loss ratio vs. Theoretical exit flow Mach number, $M_{ath}$

Profile loss ratio vs. Flow angle, $\beta_1$
Shock Loss

- The loss due to viscous dissipation across the shock is called “shock loss”.

- This loss, in principle, could be estimated theoretically, but the estimate of indirect loss associated with boundary-layer-shock interaction has to be based on computation or correlations.

- Sudden jump in static pressure across the shock results in thickening of the boundary layer and flow separation.

- This loss could be substantial portion of total profile losses, depending on Mach number and Reynolds number.

- In general, this loss is normally the smallest loss component.
When the flow on a turbine blade surface is laminar, transitional flows are often encountered.

Laminar flows with laminar separation bubbles are often encountered in the suction side of blade as the flows are accelerated.

The laminar flow may separate because of the local adverse pressure gradient.

Usually, a vortex forms just downstream of this stationary point, with some of the flow reversing direction against the wall.

The backward-turning vortex then forms a “separation bubble”.

Frequently, however, the flow does not reattach but continues as a high-velocity jet flow away from the wall.

This separation bubble will reattach, and this may in turn initiate transition result in turbulent flow downstream.

The presence of these separation bubbles results in increased losses. This is because the new streamlines of the main flow result in a pressure distribution sufficiently modified for the main flow to reattach to the wall.
Profile Loss

Loss with Blade Solidity

1. Larger Profile Loss due to Separation
2. Minimum Profile Loss
3. Larger Profile Loss due to Large Surface Area

Blade-Row Solidity (chord/spacing) vs. Losses in Total Pressure
Effect of Surface Roughness

Losses in stage efficiency as a function of surface roughness

- L.P. Turbine
- H.P. Turbine
- I.P. Turbine

Graph showing the relationship between surface roughness and loss in stage efficiency for different turbine units.

- 500 Mw Unit
- 200 Mw Unit

Parameters:
- Equivalent Sand Grain Size (Mils)
- Emery Grade
- Surface Finish, Micro-_inches C.L.A. (Flow Across Cut)
- Surface Finish, Micro-_inches C.L.A. (Flow With Cut)
Effect of Surface Roughness [2/6]

Fouling

- 목부분 이물질 부착에 의한 목면적 감소 유발
- 대형 고압터빈 노즐목에 0.01in 두께의 이물질 부착 시 최대 2~3% 출력저하 발생

[ Leading Edge of Nozzle ]

[ Sodium compound deposits on a blade row due to condenser in-leaks ]
Effect of Surface Roughness [3/6]

[ Corrosive pitting on blades and wheel ]  [ Pits and raised portions on the rotating blade inlet due to FOD ]
A surface finish is considered hydraulically smooth as long as the roughness does not penetrate the laminar sub-layer.
As the steam passes through the turbine, the pressure and temperature decrease, the steam loses its ability to hold the deposits in solution, and the impurities deposit on the turbine blades.

Therefore, the chemical composition of the deposits may be different in the individual stages.

Some deposits are water soluble and will not accumulate on stages operating in the wet region.

The deposits on the last few stages in the LP turbine of fossil and nuclear units and in the HP turbine of the nuclear unit are not water soluble.

In some fossil power plants, copper, which is not water soluble, will deposit in the HP turbine.

The efficiency of the plants is deteriorated rapidly after an effective water wash.
During a turbine overhaul, the deposits are removed by blasting the steam path with aluminum oxide. Usually 220 grit aluminum oxide is used.

The aluminum oxide blasting does some damage to the polished blade surfaces. Less damage results if 300 grit aluminum oxide is used.

Removing the deposits for the turbine does not solve the problem.

The deposits also are in the boiler, cycle components, and station piping and will again collect in the turbine after a few days of operation.

To solve a serious deposit problem, the first step is to change the boiler water chemistry.

Acid cleaning the boiler and turbine piping will help clean up the cycle.

Copper deposits are the most difficult to eliminate.

Maintaining proper pH is critical to controlling the copper deposit problem.

Grit: 연마재의 하나. 뾰족한 입자로 된 연삭 습돌의 성분
In many instances, a separated category called “trailing edge loss” is included to account for loss due to the finite thickness of the blade trailing edge, which causes flow separation and shock-expansion-wave interactions due to sharp corners. This loss could be appreciable in transonic and supersonic turbines. In subsonic turbines, this loss is used to be classified and taken account inside the profile losses.
The wake is a velocity defect generated by the boundary layers of the blade surfaces, and if undisturbed by other blades it would move downstream along the direction of outlet-flow angle while decaying slowly over three or four chord lengths.
<table>
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<tr>
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<td>Mechanical and Electrical Loss</td>
</tr>
</tbody>
</table>
Formation of Horseshoe Vortex

Horseshoe vortex formed around a square bar

Horseshoe vortex formed around a round bar

Steam Turbine

6. Turbine Losses

HloPE
Formation of Horseshoe Vortex

- Endwall flow produces endwall boundary layer.
- Endwall flow is one of major sources of turbine losses, especially in cascades with short length blades and high flow turning.
- The endwall losses occupy a substantial part of the total aerodynamic losses in a nozzle or bucket row, even as high as 30~50%.
- The boundary layer fluid upstream of the leading edge is decelerated by the adverse pressure gradient and separates at a saddle point $s_1$.
- The boundary layer fluid elements form a reverse recirculating flow just before the leading edge.
- This reverse flow produces another saddle point $s_2$.
- The upstream boundary layer is rolled-up in the recirculating zone and it is divided into two legs at the leading edge saddle point of the blade and forms the so-called horseshoe vortex.
- Then, one leg goes into suction side and the other leg goes into pressure side in axial cascades.
**Secondary Flow**

- *Secondary flow* means various vortices passing through blade-to-blade passage in axial turbines.

- The pressure side leg moves towards the suction side of neighboring blade in the passage due to the tangential pressure gradient and becomes the passage vortex.

- The suction side leg called as the counter vortex rotates in the opposite direction to the larger passage vortex.

- There are two distinct (but arising from the same physical phenomena) vortices are present on the suction side of blades and they may merge, interact or stay separate.

- Counter vortex is also called as “stagnation point vortex”, or “leading edge vortex”, or “horseshoe vortex”.

![Diagram of Secondary Flow](image)
Secondary Flow Loss

Secondary flow in a blade cascade

Secondary vortices in short and long blades
At the trailing edge, the secondary kinetic energy of the suction side leg of the horseshoe vortex can be entirely dissipated as a result of shear interaction with the stronger passage vortex.

The vortices, secondary flows, not only dissipate energy by themselves but can also cause separation of boundary layer and change the outlet flow angle. Thus, it affects the incidence on the following blades.

On the other hand, radial pressure gradients cause the blade surface boundary layer to migrate in the radial direction causing low momentum boundary layer material accumulation that can cause local flow separation specially near the trailing edge region of the blade.

The major factor affecting secondary flow is the aspect ratio.

At small aspect ratios the upper and lower endwall secondary flow vortices can grow together up to filling the blade passage along the blade height, thus significantly increasing secondary flow losses.

The result of this is that, despite the often relatively low stage loading, in the high pressure stages of supercritical turbines the secondary flow affects are predominant, with secondary losses that can represent up to 60% of the total stage losses.
Secondary Flow and Annulus Losses

Source: D.G. Wilson, pp.331.
The location of the horseshoe vortex lift-off lines, especially that of the pressure side leg depends on the load of the front part of the blade.

For the case of front-loaded profiles with high flow turning in the front part of the blade-to-blade passage, the lift-off line of the horseshoe vortex pressure side leg reaches earlier the vicinity of the suction surface than for the case of aft-loaded profiles.

All main secondary flows join at the suction side of the blade.

The rotating direction of passage vortex, which is developed from pressure side leg of horseshoe vortex and stronger than counter vortex, is opposite to counter vortex, which is developed from suction side leg of horseshoe vortex.

Therefore, there is shear interactions between passage vortex and counter vortex and this is the main cause of energy dissipation.
Advanced Vortex Blade [1/8]

Blade hub와 tip (endwall)에서 발생하는 2차유동손실을 줄이기 위하여 advanced vortex blade 설계기법 개발

Free Vortex Blade  Advanced Vortex Blade
Advanced Vortex Blade [2/8]

Concepts

① Free vortex blade
② Leaned blade
③ Compound leaned blade
   (Advanced vortex blade)
In order to reduce the secondary flow losses in turbine stages, radial velocity components are accounted for by using CFD techniques.

Radial flow distribution is biased toward the more efficient mid-section of the bucket by the redistribution of the exit angle of nozzle blade.

Stage efficiency can be improved 0.5 to 1.2% by the employment of advanced vortex blades.

Nozzle solidity is reduced to allow use of more efficient blade profiles.

Root reaction is moderately increased to increase efficiency, and tip reaction is decreased to reduce bucket tip leakage losses.
Advanced Vortex Blade [4/8]

IP Nozzle Exit Angle and Reaction
- Loss reduction at the endwall can be achieved by employment of a compound lean scheme.

- The rate of entropy generation is reduced by the reduction of velocity at the endwalls (entropy generation $\propto \int W^3 dA$, where $W$ is a relative velocity).
Advanced Vortex Blade [6/8]

Axial Velocity

- Increased inlet angle
- Reduced $c$
- Design $c$
- Increased $c$
- Decreased inlet angle

Inlet
Middle ~ Exit
Tip
Annulus height
Hub
Advanced Vortex Blade [7/8]

Features of Advanced 3D Design

- Reduced velocity at the endwalls resulting from the radial component of blade force.
- Reduced loading at the endwalls (but increased at mid-span) which act to reduce the intensity of the secondary flow.
- The vortex shedding resulting from the spanwise variation of blade load will be in the opposite direction to the secondary vorticity and so may reduce the rate at which the boundary layer is swept off at the end walls.
- More uniform flow conditions for the downstream blade row, resulting from reduced over- and underturning, which also give rise to reduced unsteady interactions.
- Owing to lean effects (i.e. reduced endwall loading and enhanced mild and progressive radial migration) the secondary passage vortex stretches radially and its intensity is thereby reduced. This effect also entails progressive smearing of low energy fluid particles in the mainstream. Note that this argument holds true only for the cases where the baseline features a very limited radial migration.
- Compound leaned blades produce a higher stage reaction both at hub and tip sections. Increased reaction at the tip section increases bucket tip leakage loss and this should be considered in the optimization process. Increased reaction at the hub decreases leakage loss occurred at the nozzle.
Advanced Vortex Blade [8/8]

Efficiency Gain with 3D Blades

Steam Turbine 6. Turbine Losses
Contoured Sidewall Test Data
### Design Process for Steam Turbines [1/3]

**Given**

Overall turbine thermodynamic parameters

**Step 1**

Choice of the overall averaged turbine key design parameters

- Number of stages
- Mean diameter
- Stage loading
- Flow coefficient
- Stage degree of reaction

These parameters are optimized for a single stage

**Step 2**

1D optimization

- Design of the inner and outer contour of the flow path
  - Decision of the best distribution of the turbine key parameters in the axial direction
  - During this step, geometrical or thermodynamic mismatching while moving from one stage to the next should be avoided
Step 3: Quasi-3D optimization

- Determination of the best combination of
  - Appropriate vortex design
  - Blade lean configuration
  - Details of meridional contour (end-wall contouring)

- Achievement of flow uniformity in the radial direction at the exit plane of each stator and rotor, and convenient level of degree of reaction at the hub and tip sections

Typical results of through flow optimization
Step 4  3D optimization

- Evaluation of the secondary flow effects
- Improvement of the profile sections at the end-walls to account for the combined effects of secondary flow and leakage flow interaction

End-wall contouring in multi-stage
Flowchart of Advanced Blade Design

- **Given:** Overall turbine thermodynamic parameters
- **Required:** Efficient & economical turbine stage design
- **Constraints:** Cost, efficiency, structural, etc.

Define global design philosophy

Through flow calc.
- Advanced vortex design

Meridional flow path design

Profile section design
- Advanced vortex design
- 1D loss correlation
- 1d / Q3D optimizer

Profile stacking
- Advanced 3D features
- Endwall corrections
- Leakage flow interaction

Profile section design
- Consistent Q3D
- 2D boundary layer

3D multi-stage CFD (steady / unsteady)
- 3D blade model
- 3D blade + seals

Mechanical integrity
- Manufacturing feasibility

Wind tunnel cascade tests
- Turbine field tests

Laboratory model turbine tests
- Derive 1D design rules for the entire application range
- Establish 1D loss correlations

Order processing

Alstom
1. **Turbine Losses and Performance Degradation**
2. **Steam Path Audit**
3. **Profile Loss**
4. **Secondary Flow Loss**
5. **Leakage Loss**
6. **Supersaturation Loss**
7. **Moisture Loss**
8. **Exhaust Loss**
9. **Heat Transfer Loss**
10. **Mechanical and Electrical Loss**
Turbine Stage Sealing System

Flow Coefficient

- Nozzle Profile: 0.58
- Bucket Profile: 0.46
- Nozzle Secondary: 0.45
- Bucket Secondary: 0.38
- Tip Leakage: 0.30

Breakdown of leakage:
- Tip Leakage (22%)
- Bucket Secondary Leakage (15%)
- Nozzle Secondary Leakage (15%)
- Nozzle Profile Leakage (15%)
- Bucket Profile Leakage (15%)
- Carryover Leakage (4%)
- Rotation Leakage (3%)
- Shaft Packing Leakage (7%)
- Root Leakage (4%)
설치 위치 및 목적

- Gland seal: 로터 양 끝 케이싱 관통 부위
  - HP/IP 터빈: 터빈 내부 증기 외부(대기중) 누설 방지
  - LP 터빈: 외부 공기 LP 터빈 누입 방지
- 버켓 티: 버켓 열 압력차에 의한 증기누설 최소화
- 다이아프램 축 관통 부위: 노즐 열 압력차에 의한 증기 누설 최소화
The installation of steam seals is a cost-effective means for improving steam turbine efficiency and power output.

Steam leakage losses are a major component of controllable losses in large steam turbines.

A demonstrated improvement in turbine section efficiencies of 5% can be achieved for full section seal upgrades.

The leakage loss can be increased up to 1% of available energy of steam turbine.

The amount of steam leakage is strongly related to the operating clearance between the rotating and stationary elements that are separated by the seal.

A second major influence on steam leakage is the pressure difference that separates the inlet and exit of the seal assembly. Seals located in regions of high pressure drop experience greater leakage flow.
Tip Leakage

- Tip leakage flow goes through the tip clearance driven by the pressure difference between the up- and downstream at tip. This flow is a function of clearance area and sealing arrangement.

- The leakage flow jet mixes out firstly in the clearance space and this mixing process is irreversible because of throttling loss.

- The output of the steam turbine reduces as the leakage flow increases. This is because the amount and velocity of the main flow reduce as the leakage flow increases.

- Front stages, where the blade height is low and pressure difference across the seal is high, are more affected by leakage loss than rear ones.

- In order to decrease tip losses, for any given stage, the radial distribution of the row pressure drop (degree of reaction) has to be optimized. This means that the degree of reaction at the tip should be decreased, but the degree of reaction at the hub should be increased to minimize the leakage losses.

- The modification of the degree of reaction is done by means of twist and lean.

- The seal design is dependent on the pressure drop across the blades. Reaction turbines that high pressure drops across the blades require the labyrinth type seal, whereas impulse turbines with small pressure drops across the blades require only spill strips.
This loss is considered in different ways depending on whether the blade is covered or uncovered.

The major parameters affecting leakage flow are tip clearance, degree of reaction, blade turning, and blade loading (local pressure difference between the pressure and suction side).

The tip flow goes through the tip clearance from the pressure side to the suction side of the blade.

Tip leakage flow rolls up into a vortex and interacts with the secondary flow.

The secondary flow and tip leakage flow interaction produces a distinct interface.
Tip Leakage

Leakage clearance

Tip vortex mixes out downstream

Tip vortex grows starting from midchord

Passage Vortex

A blade

F blade

Pressure ratio $P_{r,m}/P_{r,m, inlet}$

1.06
1.00
0.98
0.84
0.64
0.6
0.5
0.35
0.33
0.31
0.3
0.28
0.26
0.24
0.2
0.18
0.16
0.14
0.12
0.1
0.08
0.06
0.04
0.02
0.0

Steam Turbine

6. Turbine Losses

HIoPE
Tip Leakage

Tip-Clearance Correlation for Unshrouded Blades

- Tip Clearance, fraction of passage height
- Turbine Efficiency, fraction of efficiency without clearance
- Degree of reaction, \( \xi \) (

Graph showing the correlation between tip clearance and turbine efficiency for different degrees of reaction.
Tip Leakage

Steam Whirl

- A slight movement of the rotor in the radial direction reduces tip leakage flow on one side of the rotor, resulting in more flow through the rotor blades.
- This results in a larger force on that side of the rotor.
- On the opposite side the tip leakage flow is increased, resulting in less flow through the rotating blades and this results in a smaller force acting on the rotor.
- The couple produced by these two forces is off center, which causes a vibration pattern that is different than the vibration produced by oil whip or a bowed rotor.
- The vibrating rotor rubs radial spill strips, decreasing their effectiveness and reducing the magnitude of the forces producing the couple.

Oil Whirl or Whip

- Caused by the elliptical motion of the shaft within the bearing and may be as much as one cycle per 2 revolutions of the shaft.
- Possibly caused by rotor imbalance.
- Remedied by increasing the bearing load by reducing bearing length.
- This increases side leakage and reduces oil film flexibility.
1- integrated HP-IP casing and its front (a), central (b), and rear (c) seals, 2-two double flow LP casings and their end seals (d), 3&4-headers of the HP-IP and LP end seals, 5&6-gland governors of steam pressure in these headers, 7-outside source of sealing steam, 8-gland steam ejector, 9-gland steam condenser, 10-to a feedwater heater.
기동 및 저부하운전 시: seal steam을 gland로 공급
정상운전 시: gland로부터 seal steam 배출
**Shaft Packing Leakage**

**LP Turbine Gland**

LP 터빈 끝 단에 형성되는 진공압력으로 인한 외부 공기 유입 방지를 위한 seal steam을 LP 터빈 gland로 공급

Steam이 정체되어 응축수가 생성되는 것을 방지하기 위하여 seal steam과 air를 배출

배출된 seal steam과 air는 gland steam condenser에서 열교환 후 배출됨

정상운전 중 gland seal 고장은 복수기 진공을 파괴하여 터빈 고장정지 유발
1. Turbine Losses and Performance Degradation
2. Steam Path Audit
3. Profile Loss
4. Secondary Flow Loss
5. Leakage Loss
6. Supersaturation Loss
7. Moisture Loss
8. Exhaust Loss
9. Heat Transfer Loss
10. Mechanical and Electrical Loss
The Wilson line is located 60 Btu/lbm below saturated vapor line.

It was more precisely located by Yellot at a moisture of 3.2% compared to the 4% determined by Wilson.
As the steam expands, it first releases its superheat energy until it reaches the saturated condition.

Then, with further expansion, a portion of the latent heat contained in the steam is released. This conversion of latent heat introduces a state where water is formed in the expanding steam.

However, heat transfer from the gas to liquid phase requires a finite period of time, and the expansion of steam in the steam path is extremely rapid.

The elapsed time for steam entering a high pressure section to expand through it, and through the reheat and low pressure sections is about 0.2 seconds, if the time in the crossover pipes and the boiler reheater section is ignored.

Because heat transfer cannot occur instantaneously, the expansion will continue under the saturated vapor line.

At Wilson line, which is located approximately 60 Btu/lb from saturated vapor line, heat transfer will have been completed and approach thermal equilibrium conditions, and moisture will form. This is the point where fog is formed, consisting of particles from about 0.5 to 1.0 microns in diameter.

Steam inside the vapor dome is *supersaturated* above the Wilson line, a term that arises from water existing as vapor at conditions at which condensation should be taking place.
Formation of Wet Steam

- Water Droplet Erosion
- Fog Formation (Condensation Shock)
There is a supersaturation loss with no moisture above the Wilson line.

Test results from LP turbine facility (GE)

Supersaturation Loss

Average Supersaturation loss 2.0%/%

Average moisture loss 1.0%/%

Moisture loss 0.76%/%

Efficiency loss (%)

Weighted average moisture (%)
Supersaturation Loss

- Supersaturation loss occurs because the expansion in the turbine is so rapid that condensation is delayed.
- Under these conditions, the temperature drops much more rapidly for a given pressure drop.
- For this reason, it is called as “super-cooled”, or supersaturation.
- In the super-cooled condition, the available energy becomes smaller because the work term, $p\nu$, is reduced.
- The weighted average moisture from saturation to Wilson line is the average of the initial and final moisture, 1.6% $(0+3.2)/2$.
- As the steam is expanded from the saturation line to Wilson line, 3.2% moisture, the stages experience a saturation loss of 2% per percent moisture.
증기터빈에서 증기가 너무 빠르게 평창하기 때문에 포화증기선을 가로질러 평창하더라도 증기는 곧바로 응축되지 않고 약간 지연이 발생한 후에 응축(물방울 생성)이 시작됨.

이런 현상은 포화증기선 60 Btu 아래 있는 Wilson line에 도달할 때까지 지속됨.

Wilson line에 도달하면 증기는 잠열의 일부를 방출하여 본격적으로 응축되기 시작하면서 정상적인 열평형상태에 도달하며, 정상적인 증기조건을 따르게 됨. 그리고 포화증기선과 Wilson line 사이에서는 준평형상태(metastable equilibrium)를 유지하면서 비정상적인 증기조건을 따르게 됨.

포화증기선과 Wilson line 사이에 있는 증기는 이론적으로는 습증기지만 증기의 응축이 진행되지 않았기 때문에 이 영역에서 증기는 과열증기(superheated steam)처럼 행동.

Modified Mollier chart에서 상태 'a'에서 상태 'b'로 평창된 증기가 습증기라면 압력은 4.74 psia 이며, 온도는 160°F이겠지만 상태 'b'에서 증기는 과열증기처럼 행동하기 때문에 온도는 105°F임.

따라서 포화증기선과 Wilson line 사이 영역에 있는 증기는 과포화증기 (super-saturated steam), 또는 과냉증기(undercooled steam)라 함.

이렇게 갑자기 온도가 낮아지면 일부 응축된 증기가 기화되면서 엔탈피가 방출되어 압력은 증가하고 비체적과 속도는 감소하여 과포화손실(supersaturation loss) 발생.

이런 현상을 응축충격(condensation shock)이라 하며, 손실을 유발(엔트로피 증가)시키는 비가역과정 임.
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Velocity Triangle of Droplets

$Cs$: Absolute steam velocity

$Ws$: Relative steam velocity

$Cd$: Absolute droplet velocity

$Wd$: Relative droplet velocity

$U$: Peripheral rotation velocity

Diagram:

- Nozzle
- Bucket
- Direction of rotation
- Droplets
Break-up of a water droplet by interaction with steam when Weber number is greater than 14

\[ We = \frac{\rho V_r^2 d}{\sigma} \]

\( \rho \) : density of steam  
\( V_r \) : relative velocity of steam  
\( d \) : droplet diameter  
\( \sigma \) : surface tension of water
Moisture Loss

Break-up of a Water Droplet

(a) Deformation of droplet kernel,
(b) Formation of joined boundary gas + liquid layer,
(c) Surface wave formation due to Kelvin-Gelmgolz instability,
(d) Surface wave formation due to Reley-Taylor instability
 Moisture Loss

- The loss associated with the acceleration of the secondary drops is called \textit{Inter-phase Drag Loss}, and is one of the major sources of mechanical losses.

- For secondary drops entering a rotating blade row, many of them will be impinging on the suction side of leading edge of blade due to their slower velocities than the main flow – exerting a “braking” effect on the rotating blade row. The loss associated with the reduction of the blade work due to braking effect is called \textit{Braking Loss}, which is another major source of mechanical losses.

- The water film from deposition will also be moving radially towards the blade tip under the centrifugal force within the rotating blade row, in addition to being dragged axially toward the blade trailing edge. As a result, some of the work output is wasted to increase the momentum of the water film. This loss is usually called blade \textit{Pumping Loss}, which is the third major mechanical loss.
Wilson line을 지나서 1μm 이하의 작은 물방울 생성

작은 물방울은 하류로 흘러가면서 합쳐져서 큰 물방울로 성장

큰 물방울은 관성을 가지며, 이로 인해 노즐에서 급격한 증기의 유선 변화로 인해 증기유선으로부터 이탈하여 노즐표면에 충돌 부착되어 수막(water film) 형성

수막은 증기유동의 전단력으로 인하여 노즐 뒷전으로 흘러감

노즐 뒷전에서 떨어져 나온 물방울은 직경이 200μm 정도

이 물방울들은 떨어져 나올 때의 속도가 0 이므로 증기속도와의 큰 미끌림속도(slip velocity)로 인하여 평균직경 30μm 정도의 물방울로 잘게 쪼개짐(break-up)

이 과정에서 저속의 물방울들은 증기로부터 운동량을 전달받아 가속되어짐

반면에 증기는 물방울들로 운동량을 전달하여 에너지 수준이 낮아지기 때문에 증기터빈 성능저하 초래
Moisture Loss

Braking Loss

- 2상 영역에서 저속의 물방울은 탈설계각(off-design angle)으로 버켓에 부딪쳐 터빈의 기계적 일을 감소시킴

- 이는 물방울들이 터빈 버켓의 회전 반대방향으로 유입되어 충돌하기 때문임

- 결과적으로 습증기 영역에서 운전되는 터빈 단은 과열증기 영역에서 운전되는 터빈 단에 비하여 효율이 낮음 (1% 습분 = 1% 효율손실 초래)

- 일반적으로 터빈 출력에서의 습분이 12%를 초과하지 않도록 설계
  - 초과하는 경우 water droplet erosion 발생 심화
  - 이 경우 급격한 효율저하 및 부품교체 문제 발생

- 습분이 12%를 초과하는 단의 경우 습분제거장치(buckets with moisture removal grooves, etc.) 사용
Every 1% of wetness gives a 1% loss in isentropic efficiency.

Test results from LP turbine facility (GE)
Moisture Loss

- Typically, every 1% of wetness gives a 1% loss in isentropic efficiency.

- The efficiency of HP turbines in nuclear power plants is lower than that in fossil power plants.

- In most nuclear power plants, the wetness of throttle steam is close to zero and the wetness in the HP turbine exhaust is about 13%.

- The average wetness in the nuclear HP turbines is 6.5% and it results in a moisture loss of 6.5%.

- The efficiency of the LP turbines of a non-reheat nuclear power plant is about 5% poorer than a fossil unit. This is because the expansion line end point is 15% wetness in the nuclear unit and 10% wetness in the fossil unit.

- The moisture loss in a reheat nuclear unit is less than in the non-reheat unit.

- In addition, free tip L-1 and L-0 blades results in poorer nuclear LP turbine efficiency.

- Moisture loss tends to decrease as load is decreased. At very low loads the exhaust steam may actually become superheated.
Moisture Loss in Nuclear Power [1/2]

- Expansion in a HP turbine
- Expansion in a LP turbine
- Moisture separator
- Saturation line
- Enthalpy increase by moisture removal
- Expansion line end point
- Cutaway of moisture-removal bucket arrangement and drainage pocket

[ Non-Reheat Nuclear Expansion Line ]
The initial pressure and temperature in nuclear units is at or below the saturation line.

Therefore, the steam passing through turbines has a higher wetness than fossil units.

If steam were allowed to expand from the inlet conditions without moisture removal, the wetness at the exhaust would be higher than 20%.

Complete removal of moisture as soon as it is formed would improve turbine efficiency by 10%.

However, this is impossible because droplets formed during expansion are too small. To remove these droplets, external moisture separators are used.

Therefore, the HP turbine has already had a moisture loss of about 6%.

Part of this 6% loss contains super-saturation loss.
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- LSB와 복수기를 연결
- LSB를 빠져나간 축방향 유동을 반경방향으로 전환
- LSB를 빠져나간 고속의 증기에 포함되어 있는 운동에너지의 압력에너지를 회복 – 디퓨져 역할
Generals and Definition

- The exhaust loss occurs between the last stage of LP turbine and the condenser inlet, and very much depends on the absolute steam velocity.

- The exhaust loss is a kinetic energy loss and the loss associated with the pressure drop occurring in the LP exhaust hood.

- Under full-load condition the exhaust loss is generally around 3% of the available energy of turbine.

- The exhaust loss is a function of the exhaust area and steam velocity because it consists of two components: leaving loss and hood loss.

- The leaving loss is that associated with the kinetic energy of the steam exiting the last stage blades.

- The hood loss is due to pressure drop in the exhaust hood.

- The steam conditions at the condenser inlet are different from those at the exit of LSB because of exhaust loss.

- The steam conditions at the condenser inlet are those at the ELEP(Expansion Line End Point), thus, the exhaust loss is included in the ELEP.

- The steam conditions at the exit of LSB are referred to as those at the UEEP(Used Energy End Point), which is also called as TEP(Turbine End Point).

- By definition the enthalpy difference between two points is approximately equal to the turbine exhaust loss.
### Exhaust Losses

- **Work** ($W$)
- **Hood Loss** ($HL$)
- **Leaving Loss** ($LL$)
- **Exhaust Loss** ($EL$)
- **Effective Exhaust Loss** ($EEL$)
- **Used Energy End Point** ($UEEP$)
- **Expansion Line End Point** ($ELEP$)
- **Static End Point** ($SEP$)
- **Change in Exhaust Loss** ($\Delta EL$)
- **Change in Work** ($\Delta W$)
- **Change in Effective Exhaust Loss** ($\Delta EEL$)
- **Static Pressure at Turbine Exhaust Flange** ($p_c$)
- **Total Pressure at Last Blade Exit** ($p_{TB}$)
- **Static Pressure at Last Blade Exit** ($p_{SB}$)
Read the exhaust loss at the annulus velocity obtained from the following expression:

\[ V_{an} = \frac{m \nu (1 - 0.01Y)}{3600A_{an}} \]

(1)

The enthalpy of steam entering the condenser is the quantity obtained from the following expression:

\[ U_{EEP} = E_{LEP} + \text{(Exhaust loss)}(0.87)(1 - 0.01Y)(1 - 0.0065Y) \]

(2)

This exhaust loss includes the loss in internal efficiency which occurs at light flows as obtained in tests.

(3)
Exhaust Loss Curve

- The exhaust loss curve is unique to a given last stage blade length.
- The exhaust loss curve is used to determine the dry exhaust loss.
- The dry exhaust loss is found from the exhaust loss curve at the exhaust velocity.
- Equations, given on the figure, are used to correct the dry exhaust loss to the actual exhaust loss.
- This corrected exhaust loss is then added to the ELEP corresponding to the exhaust pressure under consideration.
- The exhaust loss plus the ELEP defines the UEEP, that is also called as TEP (Turbine End Point).
- The UEEP is the actual leaving enthalpy of the LP turbine steam and includes losses associated with the hood loss, leaving loss, and restriction loss or turn up loss.
- It is clear that the work produced by the turbine should be determined by performing an energy balance with the use of the UEEP, not the LP turbine ELEP.
The internal efficiency of a steam turbine does not include the loss at the turbine exhaust end. The exhaust loss includes 
(1) actual leaving loss, 
(2) gross hood loss, 
(3) annulus-restriction loss, 
(4) turn-up loss.

\[ \text{Exhaust Loss} = UEEP - ELEP \]
The exhaust area of the steam turbine should be determined by the balance between exhaust loss and capital investment in steam turbine equipment.
Exhaust Loss

1) Actual Leaving Loss [vs. Axial Leaving Loss]

- When the relative velocity leaving the LSB is very low, the absolute velocity leaving the LSB decreases, as the relative velocity leaving the LSB increases.

- As the relative velocity leaving the LSB increases, the angle formed between the relative velocity and the absolute velocity becomes rectangular. In this case, the actual leaving loss becomes minimum.

- As the relative velocity leaving the LSB increases further, the absolute velocity leaving the LSB increases. Thus, the leaving loss increases further.

- One means of reducing the leaving loss is to reduce the absolute steam velocity at the last stage by increasing the LSB length or the number of steam flows.

- 최종 단 배출속도를 줄이기 위해 longer LSB 사용 ← 경제성 검토 필요
  - 제작비 증가
  - 습분침식 증가로 운영유지비 증가

- 화력발전이 원자력발전에 비해 상대적으로 짧은 LSB 사용으로 인하여 leaving loss 증가.
2) Hood Loss

- The hood loss is due to the pressure drop associated with the LP exhaust hood.
- LP hood에서 발생하는 압력손실은 유체마찰과 와유동에 의하여 발생.
- Hood loss를 줄여주기 위해서는 exhaust hood를 공기역학적으로 설계하여 증기가 exhaust hood를 통과하면서 원활한 확산(diffusion)이 일어나도록 해야 함.
- 이런 이유로 인하여 최근 들어 advanced LP exhaust hood에 대한 설계기술이 경쟁적으로 개발되고 있으며, 이를 통해 증기터빈 출력 및 열효율 향상이 크게 이루어지고 있음.

Path lines start from LSB hub
Path lines start from LSB pitch
Path lines start from LSB tip
Efficiency Improvement in PC-Fired Plant

- **Excess Air**
  - 1.15
  - 1.25

- **Discharge Flue Gas Temperature**
  - 120°C
  - 130°C

- **Main Steam Condition**
  - 300 bar/600°C
  - 250 bar/540°C

- **Reheat**
  - Double reheat
  - Single reheat

- **Back Pressure**
  - 0.88 in.HgA
  - 1.9 in.HgA

**Steam Turbine**
The purpose of diffusers is to convert dynamic pressure at inlet to static pressure at outlet.

Therefore, a useful measure of performance is the ratio of the increase in static pressure to the inlet dynamic pressure: the pressure recovery coefficient, which is also called as pressure rise coefficient

$$C_{pr} = \frac{p_{ex} - p_{in}}{p_{T,in} - p_{in}}$$

Ideally, $C_{pr}$ could reach unity.

In practice, however, a pressure recovery coefficient of over 0.80 can be obtained only in favorable circumstances.
Design Concepts for Advanced Hoods

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Inlet

Exit

Condenser Flange

Condenser

\[ P_{T,AN} \]

\[ P_{AN} \]

\[ P_{T,AN} \]

\[ P_{AN} \]

\[ P_{T,FL} \]

\[ P_{FL} \]

1. \( P_T \) in ordinary LP hoods
2. \( P_S \) in ordinary LP hoods
3. \( P_T \) in advanced LP hoods
4. \( P_S \) in advanced LP hoods
5. Steam velocity
6. Flow area

LP last stage
Exhaust Loss

3) Turn-up Loss

- Normal Rating Operation
- Low Load Operation
When the relative velocity leaving LSB is very low, LSB acts like a compressor, and this makes the exhaust loss is getting higher.

Evidence this pumping action can be detected on turbines with an L-1 extraction. That is, the pressure of extracted steam from L-1 stage is lower than the condenser pressure during part load operations.

The heat produced by the pumping action requires cooling on both LSB and LP exhaust hood.

In order to remove the *windage heat* that is generated by re-circulation occurred in the lower half of last stage blade, water is sprayed into the exhaust hood.

The spray water does an effective work of cooling the LSB and exhaust hood.

The spray water starts at 60°C (140°F) in LSB exit and turbine is tripped at 107°C (225°F) in LSB exit or at 260°C (500°F) in L-1 stage.

Additional evidence can be detected by the slight water droplet erosion occurred near the root on the suction side of trailing edge of LSB.

This water droplet erosion is caused by the suction of the spray water into the trailing edge of LSB because of reverse pressure gradient between L-1 and the last stage.

It had also been found that a large recirculation flow is formed near the root of LSB because of reverse pressure gradient between L-1 and the last stage.

This recirculation flow produces another loss, which is called as “turn-up loss”.
Exhaust Loss

3) Turn-up Loss

[ Eroded Trailing Edge of LSB near the Hub ]

[ Recirculation in the Exhaust Hood ]

Water supply line

Water spray

Water running down casing walls

Recirculating steam
Exhaust Loss

3) Turn-up Loss

![Diagram showing exhaust loss with distance and exit axial velocity parameters.](image-url)
Annulus Restriction Loss

When the pressure ratio across the last stage is high enough the axial steam velocity to become sonic.

Beyond this point any increase in pressure will not produce additional work.

The additional pressure drop beyond the sonic pressure ratio is the pressure drop occurred in the hood, and it is the annulus restriction loss.
[Exercise 6.1] 
아래 그림은 실제 증기터빈에 대한 heat balance 가운데 LP터빈 부위를 나타낸 것이다. 이 그림에 나타나 있는 자료를 참고하여 UEEP를 계산하여 그림에 나타나 있는 값과 비교하시오. 그리고 계산된 UEEP를 이용하여 LP터빈 출력을 계산하시오. 이 증기터빈의 주요사항은 출력 700 MW, 6 flow, 30"LSB이다. 그리고 exhaust loss curve는 GE사 자료를 참고하시오.

Steam Turbine

6. Turbine Losses
[Solution]

배기손실을 계산하기 위하여 \( V_{an} \) (annulus velocity)를 계산한다.

\[
m = \text{total mass flow rate exiting LSB/number of flows} = 2,906,038/6 = 484,340 \text{ lb/hr}
\]

\[
\nu = 274.86 \text{ ft}^3/\text{lb} \text{ (based on the condenser pressure of 2.5 in.Hg)}
\]

Exhaust loss curve에서 30" LSB에 대한 LSB 출구 단면적을 찾는다.

\[
A_{an} = 55.6 \text{ ft}^2 \text{ (from GE data)}
\]

ELEP를 이용하여 습분을 찾는다.

\[
Y = 9.36\% \text{ (from steam table based on } p=1.22833 \text{ psia, } h=1012.23 \text{ Btu/lb)}
\]

\( V_{an} \)은 다음 식을 이용하여 구한다.

\[
V_{an} = \frac{\dot{m} \nu (1-0.01Y)}{3600A_{an}}
\]

\[
= \frac{484,340 \times 274.86 \times (1-0.01 \times 9.36)}{(3600 \times 55.6)} = 602.8 \text{ fps}
\]

배기손실 곡선으로부터 \( V_{an} \)에 해당하는 값을 찾으면 건배기손실은 약 11 Btu/lb이다. 따라서 증기의 단위 질량유량당 실질적인 배기손실은 다음과 같이 계산한다.

\[
\text{Actual exhaust loss} = (\text{exhaust loss})(0.87)(1-.01Y)(1-.0065Y)
\]

\[
= 11 \times 0.87 \times (1-.01 \times 9.36)(1-.0065 \times 9.36) = 8.15 \text{ Btu/lb}
\]

UEEP는 다음과 같이 계산한다.

\[
\text{UEEP} = \text{ELEP} + \text{Actual exhaust loss} = 1012.23 + 8.15 = 1020.38 \text{ Btu/lb}
\]
[Solution]

열역학 제1법칙으로부터 증기터빈 출력에 관한 식을 유도할 수 있다.

\[ q_{12} = h_2 - h_1 + \frac{1}{2}(V_2^2 - V_1^2) + g(z_2 - z_1) + w_{12} \]

증기터빈에서 팽창은 단열과정이며, 위치에너지 변화는 무시할 수 있으므로 위 식은

\[ w_{12} = h_{o,1} - h_{o,2} \quad W_{12} = \dot{m}(h_{o,1} - h_{o,2}) \]

d다수의 입구와 출구를 가지는 경우 터빈 출력: \( W_T = \sum \dot{m}h_{o,in} - \sum \dot{m}h_{o,ex} \)

따라서 LP터빈 출력은 다음과 같다.

\[ W_{LP} = 3,566,547 \times 1337.8 - 131,561 \times 1251.01 - 154,340 \times 1206.71 \]
\[ - 174,907 \times 1164.73 - 199,699 \times 1102.91 - 2,906,038 \times 1020.38 \]
\[ = 1,031.27 \times 10^6 \text{ Btu/hr, or } 302,218 \text{ kW} \ (1 \text{ Btu} = 0.252 \text{ kcal} = 1.055 \text{ kJ}) \]

여기서 LP터빈 출력을 계산하기 위하여 ELEP가 아니라 UEEP를 이용하였다는 것을 주의해야 한다.
Exhaust loss curve에 나타나 있는 배기손실은 습증기에 포함되어 있는 습분 효과가 배제되어 있는 건배기손실 (dry exhaust loss)이다.

따라서 실질적인 배기손실을 계산하기 위해서는 습분을 계산하여 습분 효과를 반영시켜 주어야 한다.

Exhaust loss curve는 LSB 길이에 따라서 달라지며, LSB 길이가 동일하더라도 LP exhaust hood 형상이 달라지면 다른 값을 가진다.

따라서 새로운 LP 터빈을 설계하는 경우에 동일한 LSB를 사용하더라도 LP exhaust hood 형상이 달라지면 새로운 하나의 exhaust loss curve가 만들어진다.

LP 터빈의 실질적인 출력은 LP 터빈 입구와 LSB 출구 사이의 엔탈피 낙차에 의해서 결정된다.

따라서 LP 터빈의 실질적인 출력을 계산하기 위해서는 ELEP가 아니라 UEEP를 사용해야 한다.

여기서 계산된 UEEP는 heat balance에 주어진 값과 다른 값을 가지는데, 이는 GE사 자료를 사용하였기 때문에 것으로 판단된다. 참고로, 문제에서 주어진 heat balance는 GE사 자료가 아니다.
9. Heat Transfer Loss

- 열전달 손실 - 터빈내부는 고온의 증기가 흐르며, 외부는 차가운 공기로 채워져 있어 내부의 증기가 가지고 있는 열량 일부가 열전달을 통해서 외부로 빠져나간다. 이를 열전달손실이라 함

- 터빈 케이싱과 foundation 사이의 전도에 의한 열손실

- 터빈 케이싱과 터빈빌딩 사이의 대류에 의한 열손실
  - 증기온도가 높은 고압부에서 활발하게 일어나지만 이 부분은 직경이 가장 작고 단열이 잘 되어 있음
  - 저압 부위는 터빈빌딩 내부온도에 비해 그리 높지 않기 때문에 단열을 고려하지 않음

- 전체적인 열전달 손실은 매우 작아서 일반적으로 무시됨

- 그러나 기계 구동용과 같은 소형터빈의 경우 열전달에 의한 손실은 대개 터빈 에너지의 몇 % 정도에 해당됨
10. Mechanical and Electrical Loss

- Mechanical losses are very small and generally less than 1% of turbine energy.

- Bearing and gear losses arise from oil friction in the bearings (Bearing losses arise due to oil friction in the bearings).

- Auxiliaries such as the main oil pump are usually directly driven from the turbine shaft and add to turbine losses.

- Mechanical losses are practically constant and do not depend on load variation.

- The mechanical loss ratio decreases as the turbine size increases.

- Recently, efficiency has improved to 99~99.5%.

- Recent large-scale liquid-cooled generators have efficiencies of 98~99%.

- Operating at 3,000 or 3,600rpm is generally more efficient than 1,500 or 1,800rpm.
11. Aerodynamic Losses

- Laminar flow around a good shape.
- Turbulent flow around a bad shape. Drag is proportional to the size of the wake.
- Too steep an angle at the rear causes separation and increased drag. The rounded front moves the separation point farther back, compared to the flat fronted case.
- Shallow angle with sharp cut-off (Kamm tail) leaves a smaller wake and less drag.

증기통로에 있는 각종 struts도 공기역학적 특성이 좋은 형상으로 개선
12. Condenser Pressure Rise

- If the exhaust pressure is raised too high, that is, in excess of 6.0 in.Hga, there is a possibility of flutter in last stage buckets.

- If the exhaust pressure is raised too high, there is also a possibility of performance loss caused by recirculation of the discharging steam, which could occur because of the reduction in volumetric flow.

![Graph showing the relationship between condenser pressure and power gain, with source: EPRI Report No. GS-7003.](image-url)
질의 및 응답