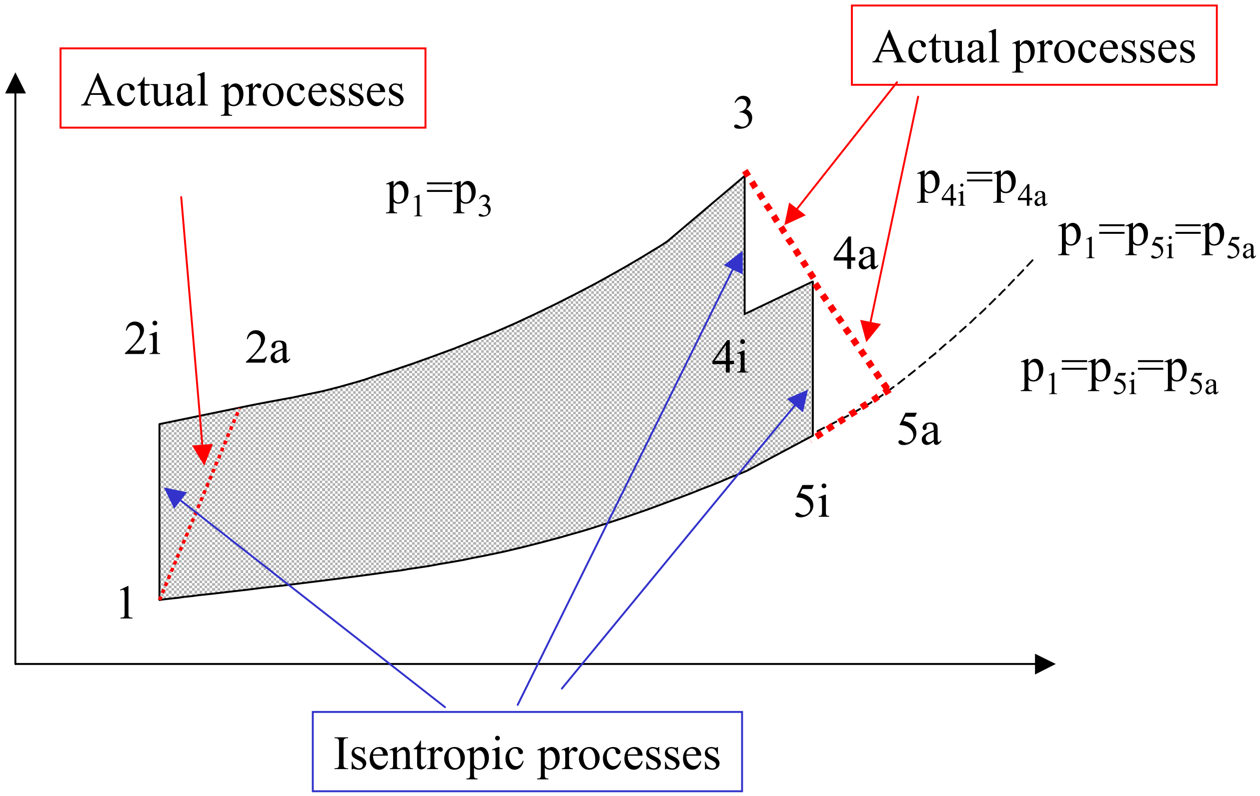
Non-ideal Cycle



From 1 to 2i, isentropic process: $p_2/p_1=Pr_2/Pr_1$, $p_{2i}=p_{2a}$

From 3 to 4i, isentropic process: $p_{4i}/p_3=Pr_{4i}/Pr_3$, $p_{4i}=p_{4a}$

From 4a to 5i, isentropic process: $p_5/p_4=Pr_{5i}/Pr_4$, $p_{5i}=p_{5a}$

Other Effects (Inlet Diffuser)

- Air entering the inlet diffuser section experiences slowdown, can be modeled as an isentropic deceleration to stagnation condition (assume air velocity entering the compressor section is negligible compared to the flight velocity)

From isentropic relation: $\dfrac{T_{o1}}{T_{in}} = 1 + \dfrac{\gamma - 1}{2} M_{in}{}^{2}$,

where M is the flight Mach number. The temperature

increase is due to the conversion of inlet flow kinetic energy into the

internal energy: $h_{in} + \dfrac{V^2}{2} = h_{o1}$. If M_{in}=1, $\dfrac{T_{o1}}{T_{in}} = 1.2$,

there is a significant difference.

However, $T_{o1} \approx T_{in}$ if the velocity (Mach number) is small enough.

$\dfrac{T_{o1}}{T_{in}} = 1.018$ if M_{in}=0.3. This suggests that the flow kinetic energy inside

the engine can usually be neglected (usually at a low Mach numer).

Other Effects (Actual Medium)

• Previous discussion involves only air as the working medium. However, the actual engine has more than air passing through the combustion chamber. The exhausting gases might include unburned fuel, burned products, and excess air. For simplicity, we are going to discuss only the special case where combustion has completed and excess air is the only unknown quantity needed to be calculated. The calculation of excess air for a steady flow combustion has been discussed in chapter 4.8.

• Once the composition of the product gas is determined, all thermodynamic properties can then be calculated according to their respective percentage distribution in the gas (see example 5.4 in FGT and adiabatic flame temperature example).

Other Effects (Air Equivalent)

• The previous consideration requires very time-consuming trial-and-errors calculations due to the fact that all gases involved have to be accounted for.

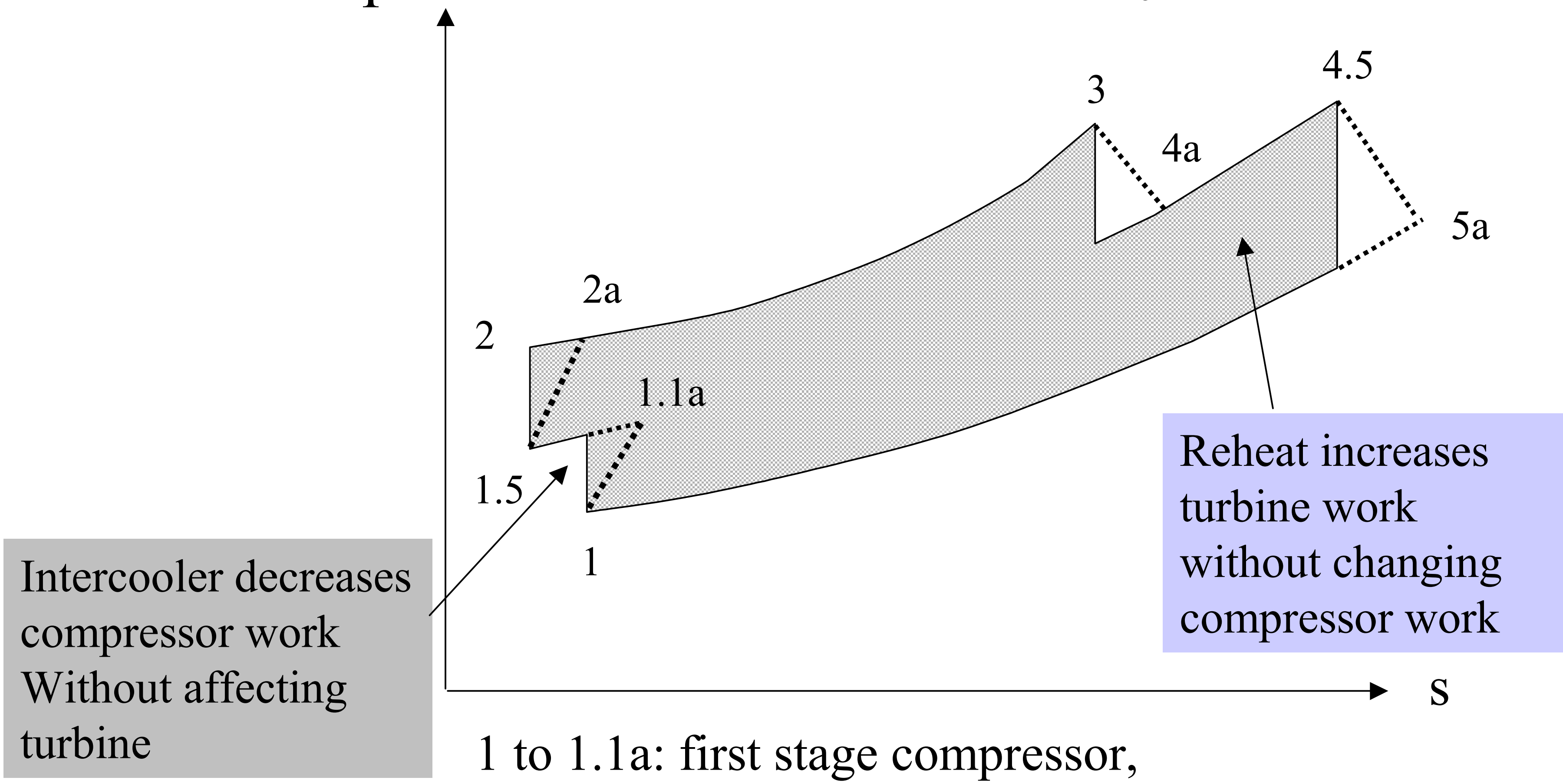
• Simplify the problem by approximating the actual medium using the air equivalent. In other words, we do not consider the difference in thermodynamic properties for different gas composition by replacing the contribution from the fuel and its products by equivalent amount of air. In other words, we assume that the amount of air expanding through the turbine is $m_{in}(1+f')$, where f' is the fuel-air ratio ($f' = m_{fuel}/m_{air}$, example 5.5).

Correction: $W_{compressor,a} = (1+f')W_{GT,a} \neq W_{GT,a}$

$W_{net} = (1+f')W_{PT,a}$,

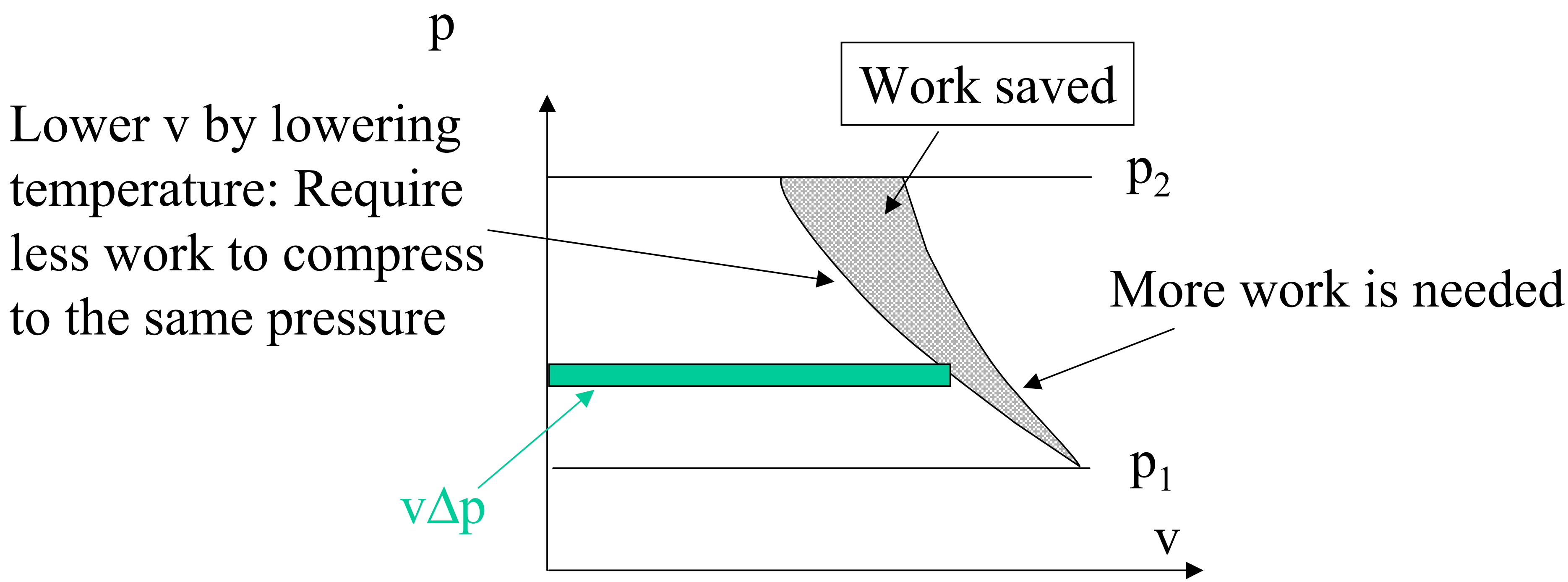
$q = \dot{m}_a (f')(LHV)$, LHV : Lower Heating Value for fuel

Intercooler & Reheat Cycles



1 to 1.1a: first stage compressor,
1.1a to 1.5: **intercooler**
1.5 to 2a: second stage compressor,
3 to 4a: GT turbine
4a to 4.5 **reheat**
4.5 to 5a PT turbine

Intercooler



For a compressor, the isentropic work is:

$$Tds = dh - vdp, dh = vdp$$

work required: $-{}_1W_2 = \int_1^2 vdp$