CO_2 Expander to Improve System Efficiency

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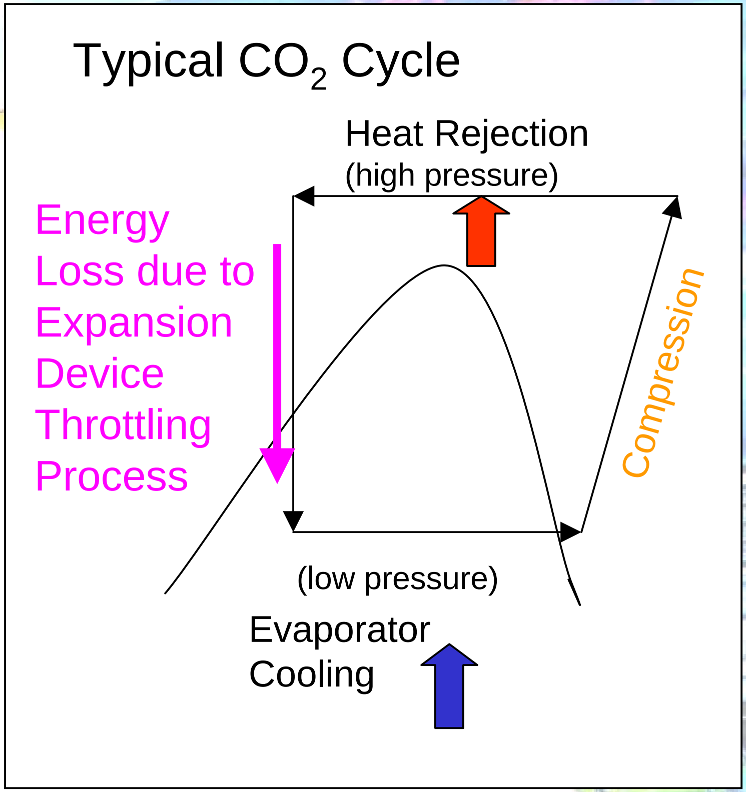
- Introduction and History
- CO_2 Cycle Simulation
- Summary

How can Energy be recovered?



- During the expansion device throttling process, a significant amount of pressure energy is lost
- If all this energy could be recovered for a CO_2 system, then 45% of the compressor work could be recovered (note: for R134a the theoretical recoverable energy is only 21%)
- SANDEN is considering developing a low cost expander/compressor that could recover this “lost” energy that could realistically improve the total system efficiency by 25-30%

History

Modified existing equipment is not a practical way forward, a special design for automotive application is required!

Author	Expander type	Design	Application	Medium
Maurer (FH Gießen)	axial piston and gear expander	modified hydraulic devices	automotive air conditioning	CO_2
Heyl, Quack (TU of Dresden)	free piston expander	own design	heat pump	CO_2
Heidelck, Kruse (University of Hannover)	double sided swash plate compressor/ expander	own design	commercial application	CO_2
Preisner, Huff (University of Maryland)	scroll expander	modified SANDEN scroll compressor	automotive air conditioning	CO_2
Smith (University College London)	screw expander	own design	power recover	water vapour
Carrier	screw expander	own design	refrigeration system	R404a
Pischinger (RWTH Aachen)	screw, roots, scroll devices	AMR380 (Aisin), OA1040 (Opcon), G40 (VW)	fuel cell	air

Definition of Efficiencies

$$\eta_{is} = \frac{\Delta h_{comp,is}}{\Delta h_{comp,real}}$$

Isentropic efficiency

$$\eta_{vol,comp} = \frac{V_{comp,real}}{V_{comp,theor}}$$

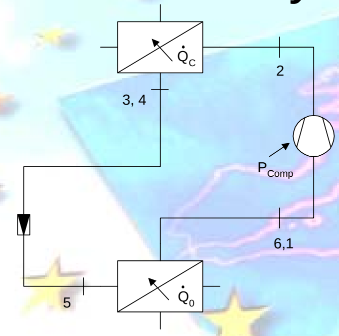
Volumetric efficiency compressor

$$\eta_{vol,exp} = \frac{V_{exp,theor}}{V_{exp,real}}$$

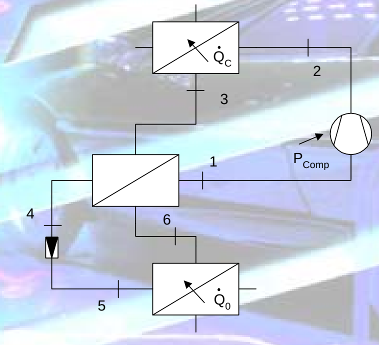
Volumetric efficiency expander

CO_2 Cycle Simulation

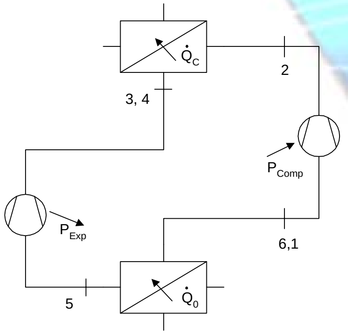
- Baseline Cycle



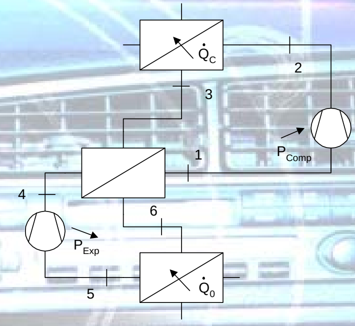
- Baseline Cycle with IHX

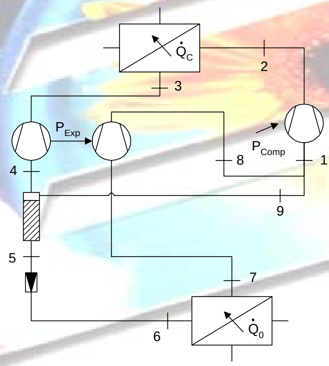


- Expander Cycle



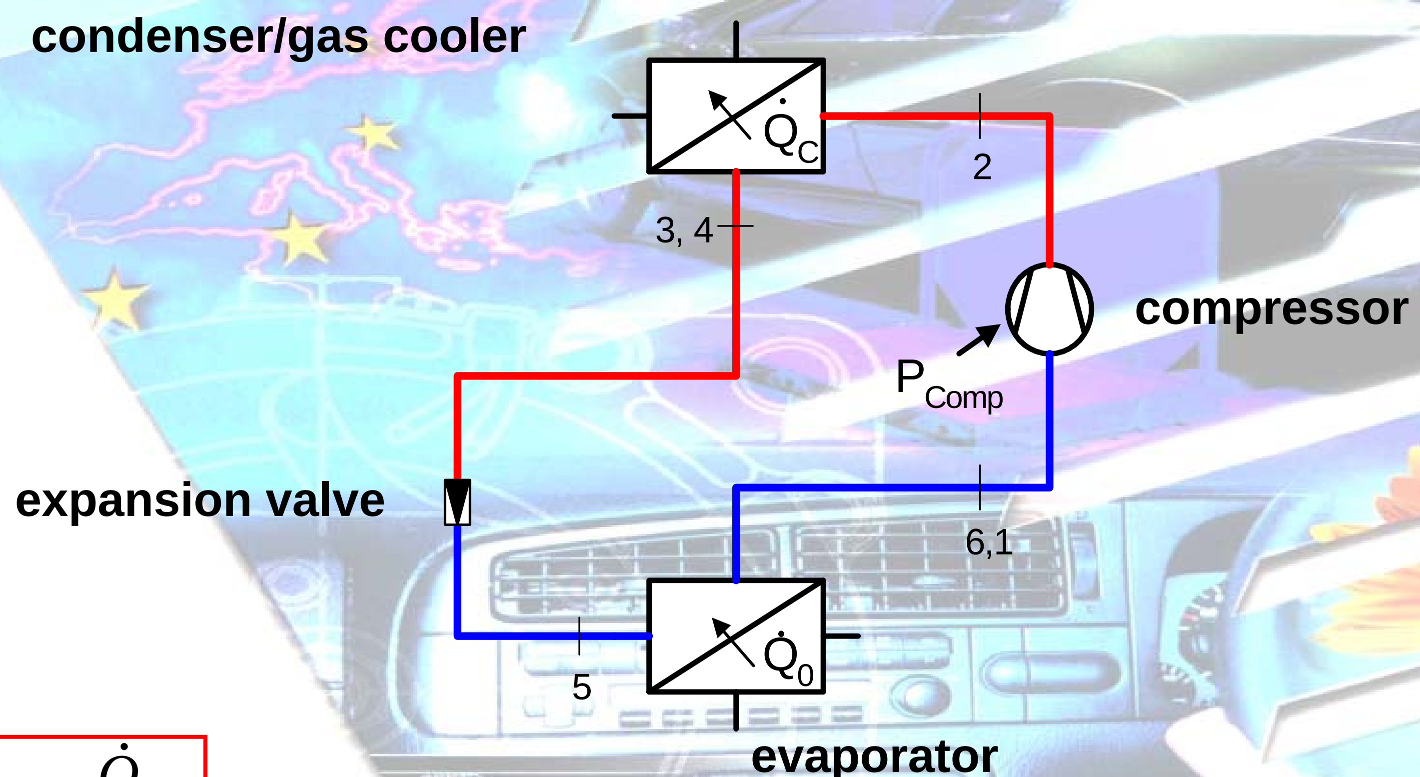
- Expander Cycle with IHX





- Expander **C**ycle with **I**ntermediate **P**ressure (CIP)

Baseline Cycle



$$COP = \frac{\dot{Q}_0}{P_{Comp}}$$

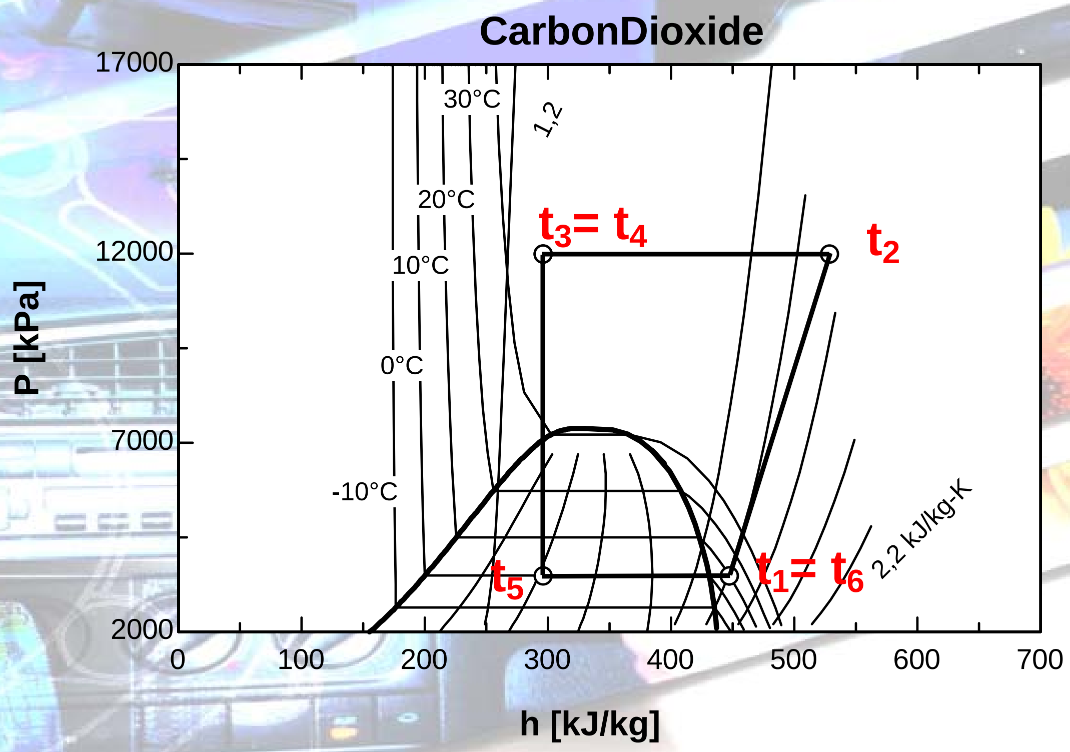
Boundary Conditions for Baseline Cycle

- Fixed system conditions:

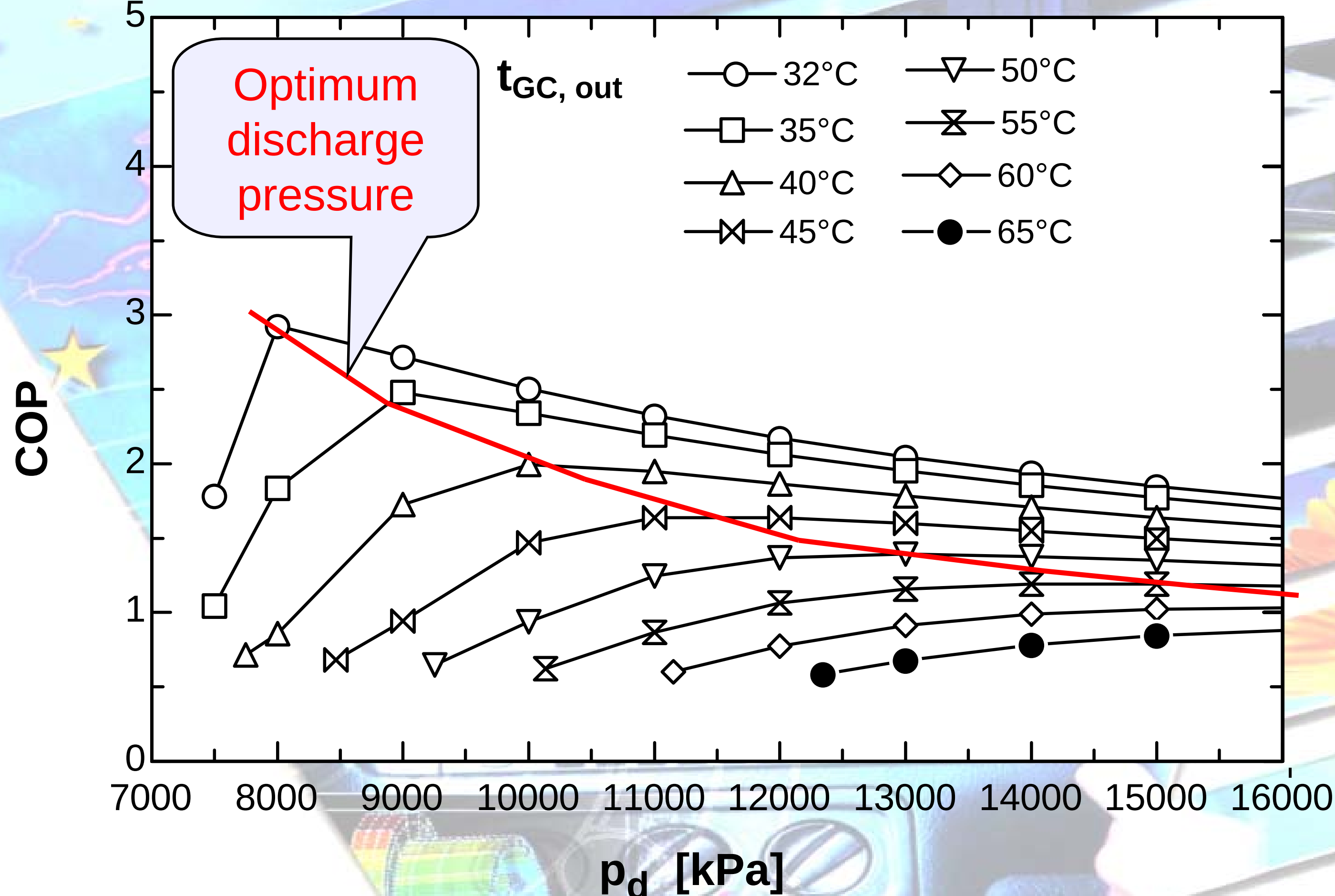
t_5	0°C (3485 kPa)
$t_6 = t_1$	10°C
η_{is}	0,7 (compressor)
η_{vol}	0,9 (compressor)

- Variable system conditions:

temperature gas cooler outlet ($t_{GC,out} = t_3$)
discharge pressure (p_d)

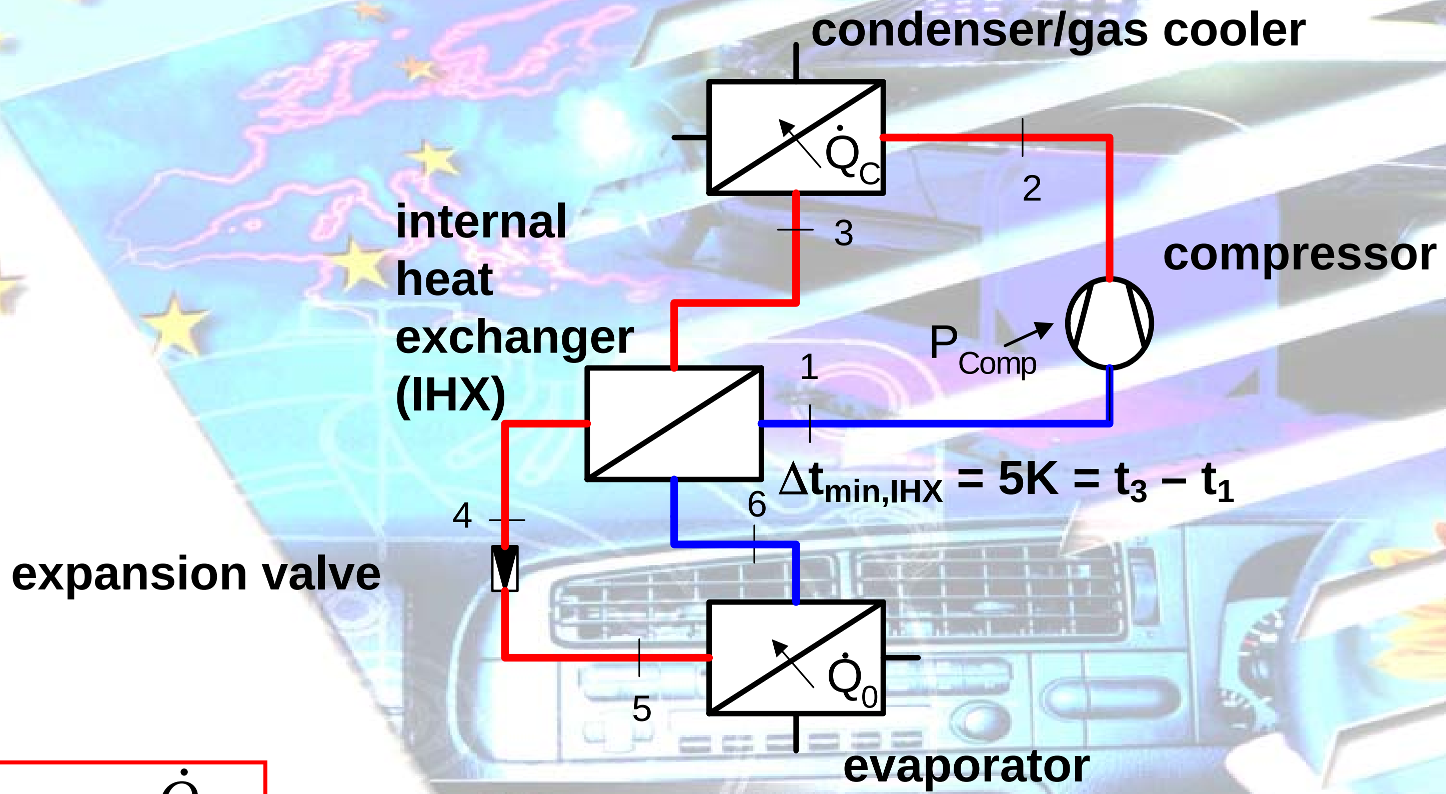


COP Baseline Cycle



p_s = 3485 kPa / $t_6 = t_1$ = 10°C / parameter = $t_{GC, out}$

Baseline Cycle with IHX



$$COP = \frac{\dot{Q}_0}{P_{Comp}}$$

Boundary Conditions for Baseline Cycle IHX

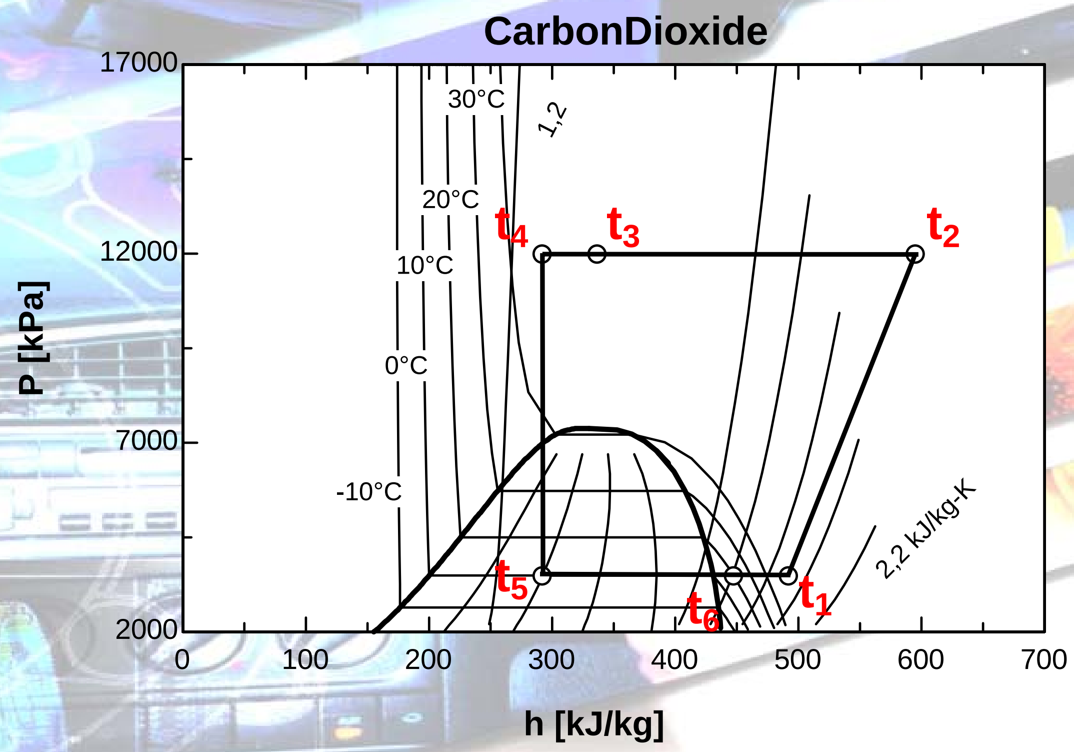
- Fixed system conditions:

t_5	0°C (3485 kPa)
t_6	10°C
$\Delta t_{min,IHX}$	5 K (= $t_3 - t_1$)
η_{is}	0,7 (compressor)
η_{vol}	0,9 (compressor)

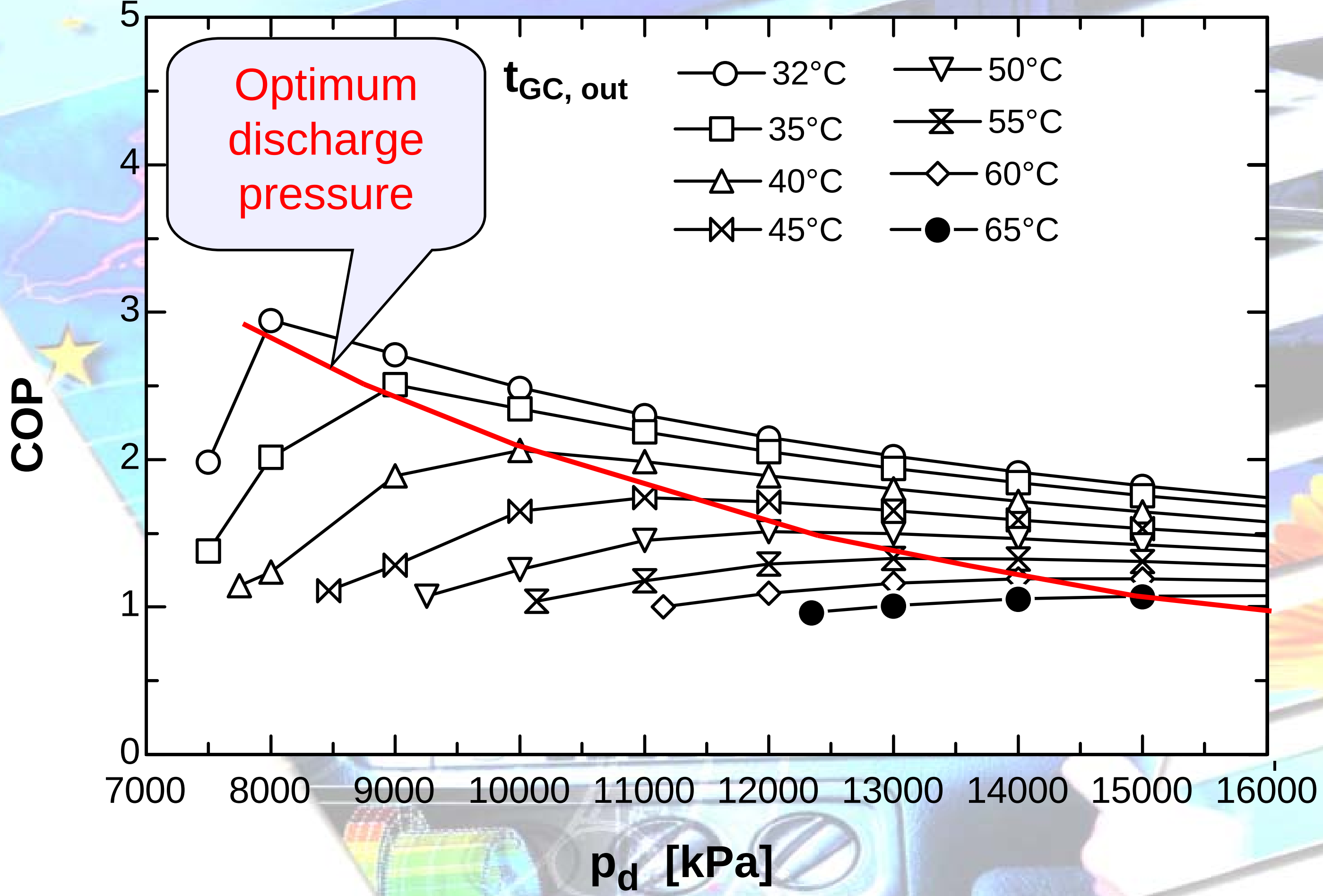
- Variable system conditions:

temperature gas cooler outlet ($t_{GC,out} = t_3$)

discharge pressure (p_d)

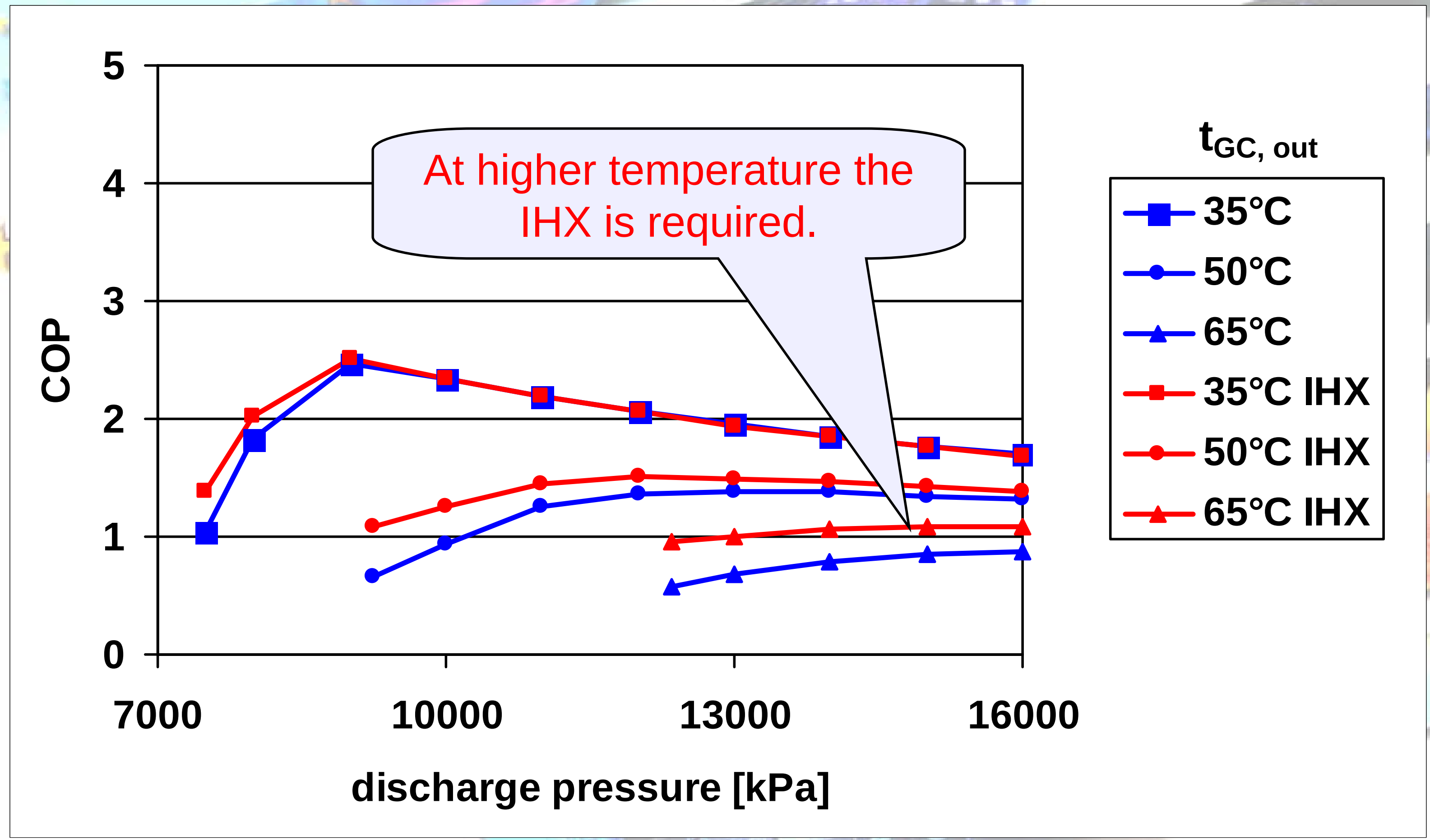


COP Baseline Cycle with IHX

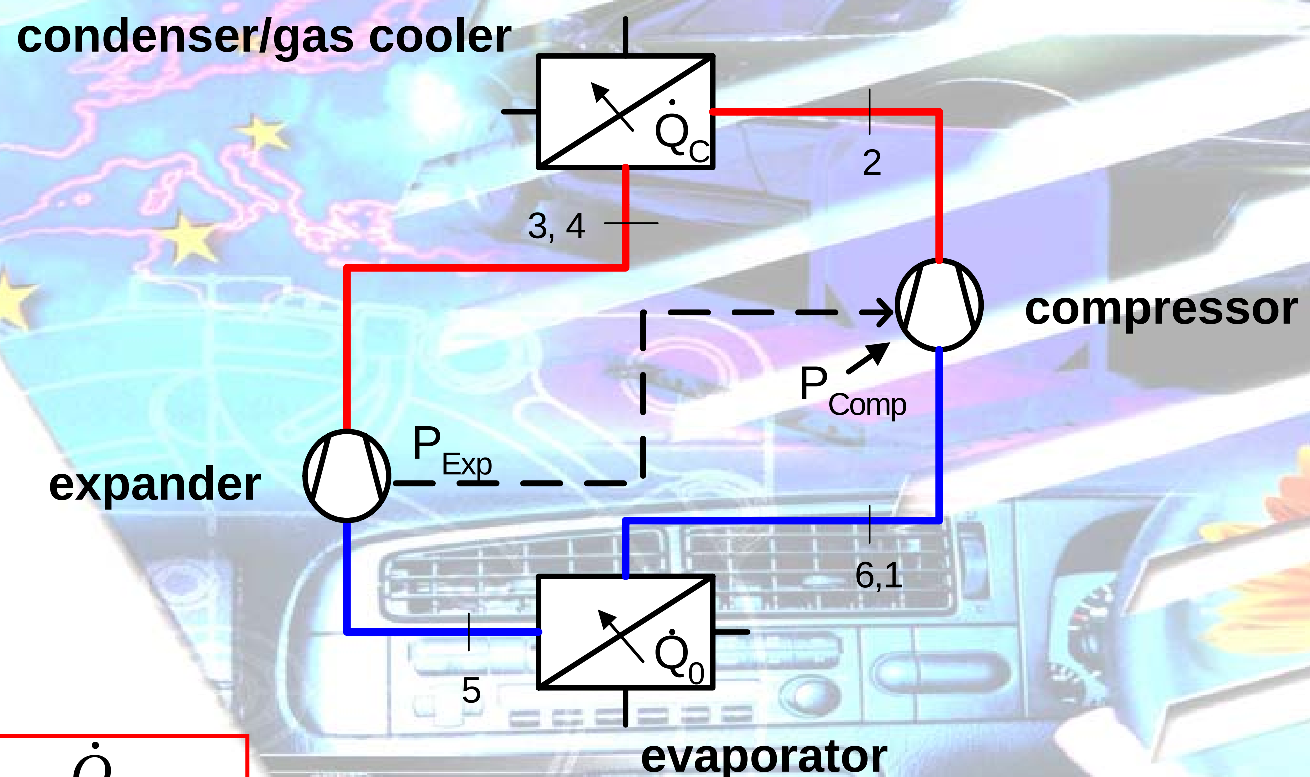


p_s = 3485 kPa / t_6 = 10°C / $\Delta t_{min,IHX}$ = 5 K /parameter = $t_{GC,\ out}$

COP Comparison Baseline Cycle w/o and w IHX



Expander Cycle



$$COP = \frac{\dot{Q}_0}{P_{Comp} - P_{Exp}}$$

Boundary Conditions for Expander Cycle

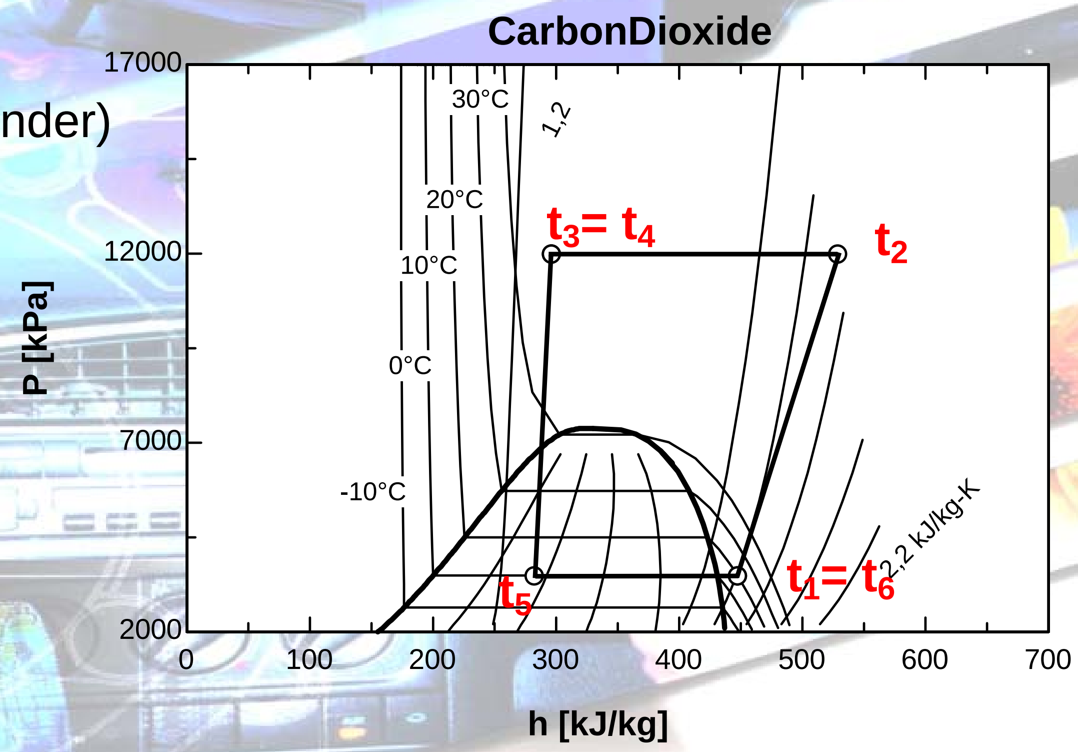
- Fixed system conditions:

t_5	0°C (3485 kPa)
$t_6 = t_1$	10°C
η_{is}	0,7 (compressor)
η_{is}	0,85 (expander)
η_{vol}	0,9 (comp. and expander)

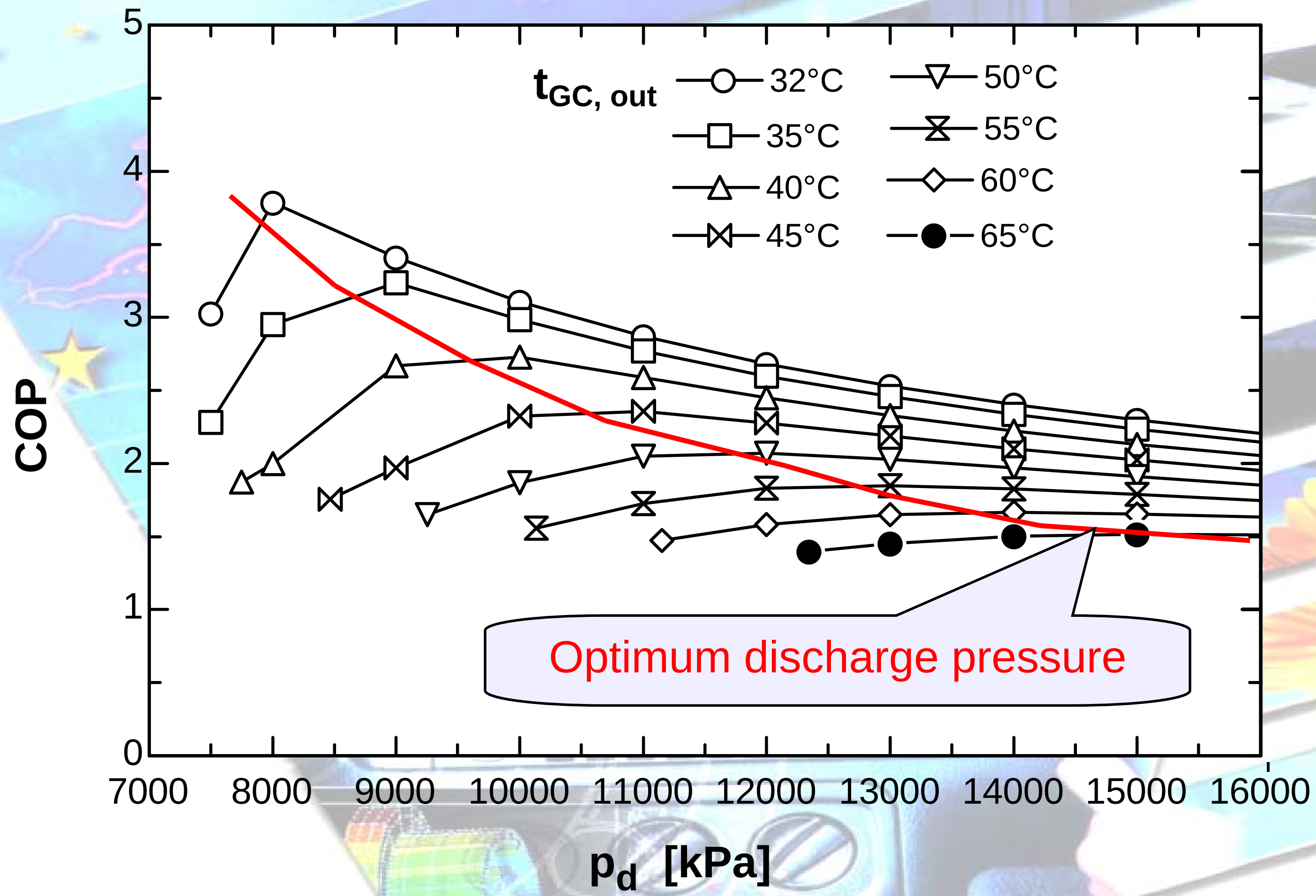
- Variable system conditions:

temperature gas cooler outlet ($t_{GC,out} = t_3$)

discharge pressure (p_d)

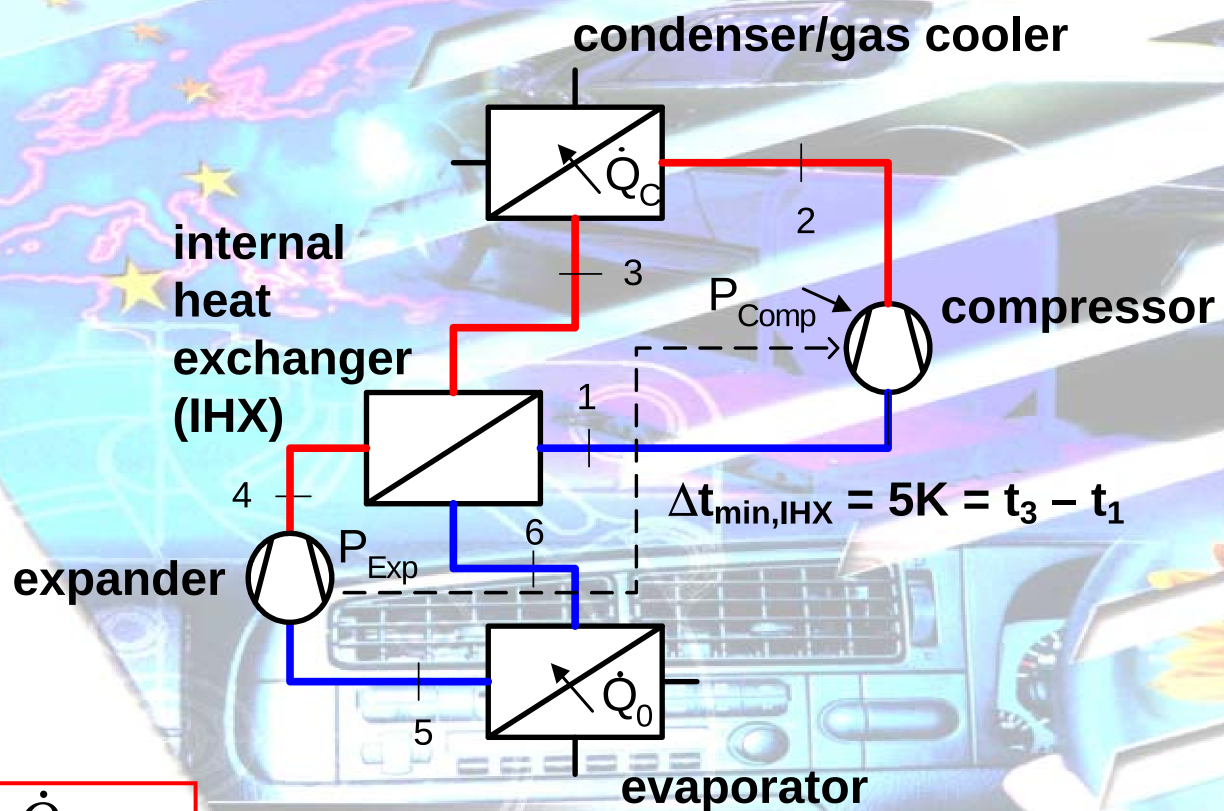


COP Expander Cycle



p_s = 3485 kPa / $t_6 = t_1$ = 10°C / parameter = $t_{GC, out}$

Expander Cycle with IHX



$$COP = \frac{\dot{Q}_0}{P_{Comp} - P_{Exp}}$$

Boundary Conditions for Expander Cycle with IHX

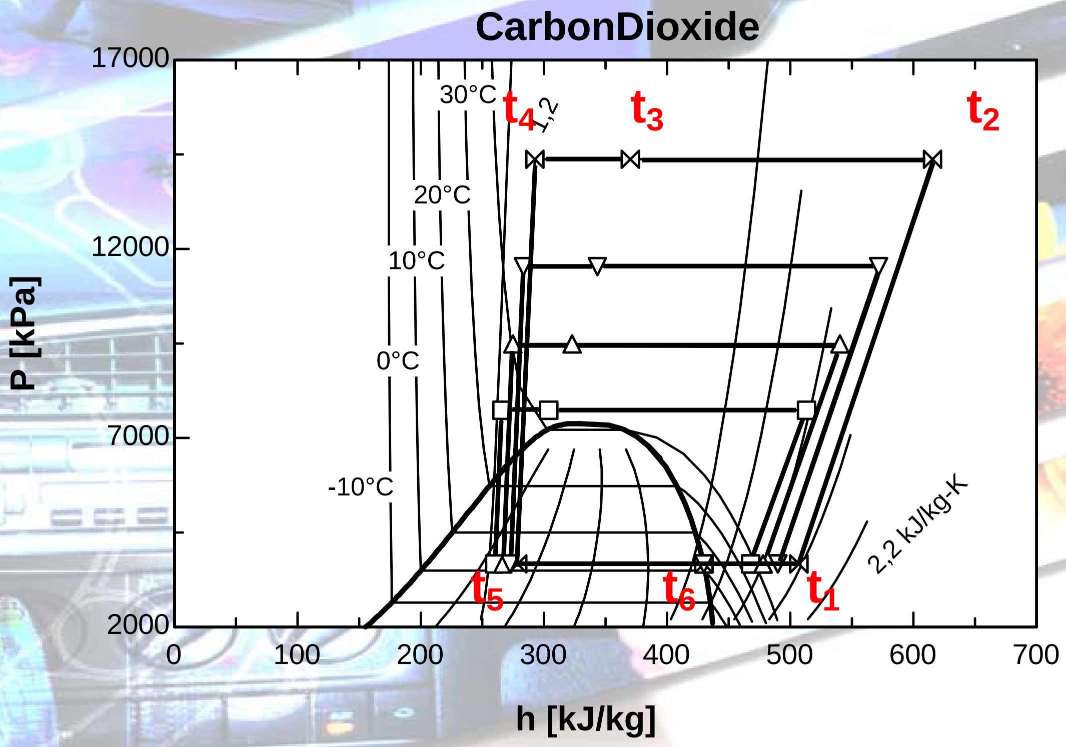
- Fixed system conditions:

t_5	0°C (3485 kPa)
t_6	10°C
$\Delta t_{min,IHX}$	5 K (= $t_3 - t_1$)
η_{is}	0,7 (compressor)
η_{is}	0,85 (expander)
η_{vol}	0,9 (comp. and exp.)

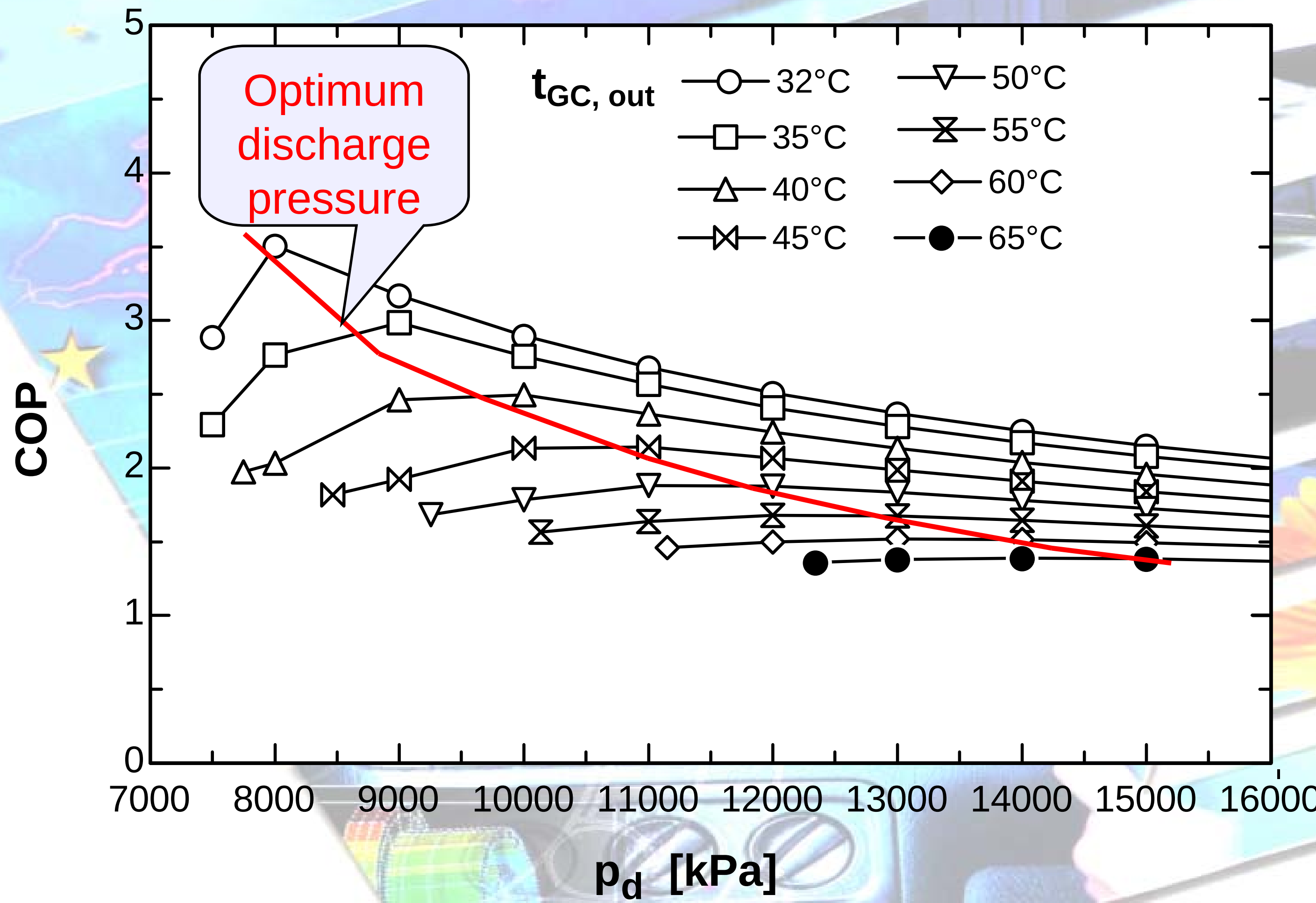
- Variable system conditions:

temperature gas cooler outlet ($t_{GC,out} = t_3$)

discharge pressure (p_d)

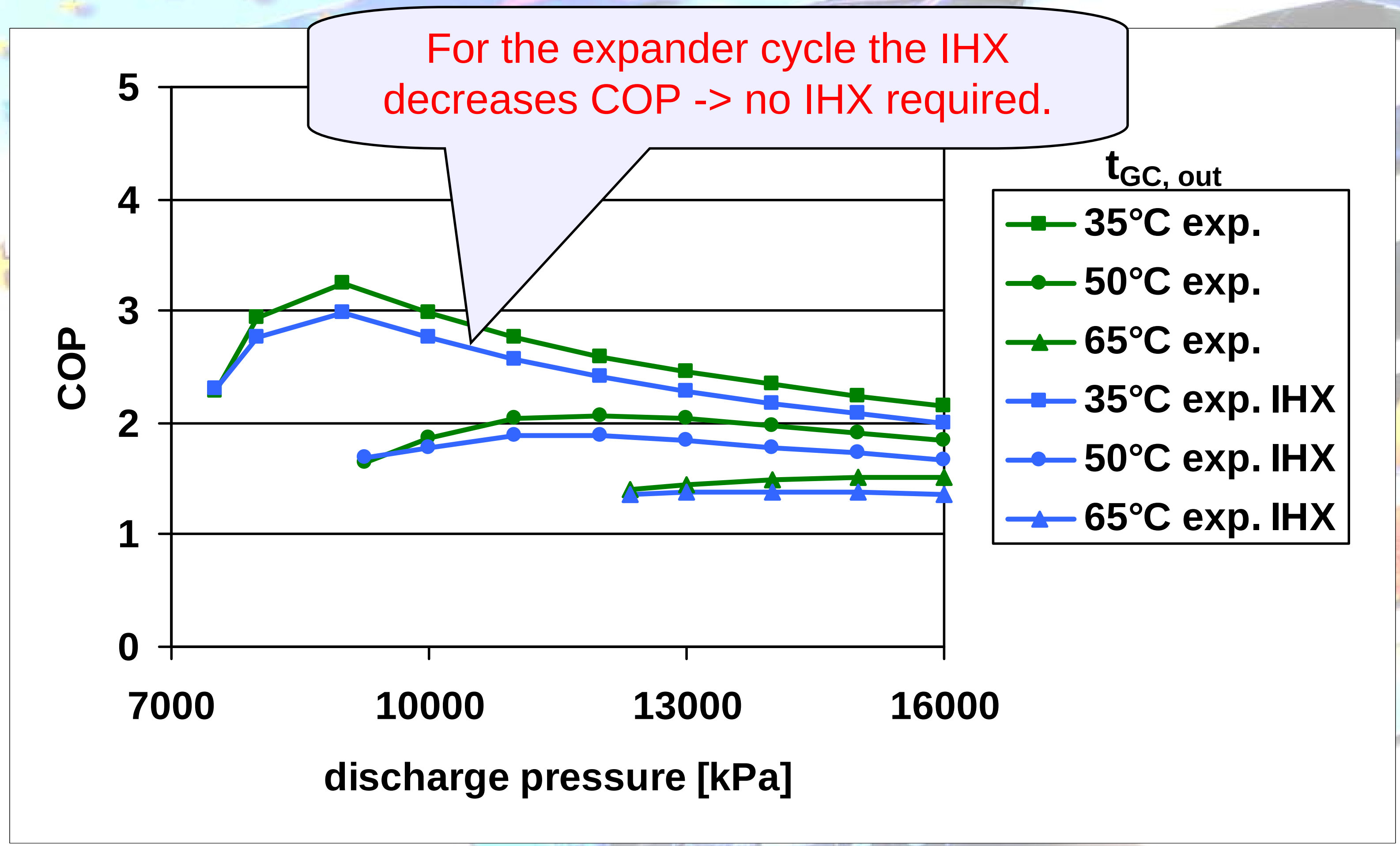


COP Expander Cycle with IHX

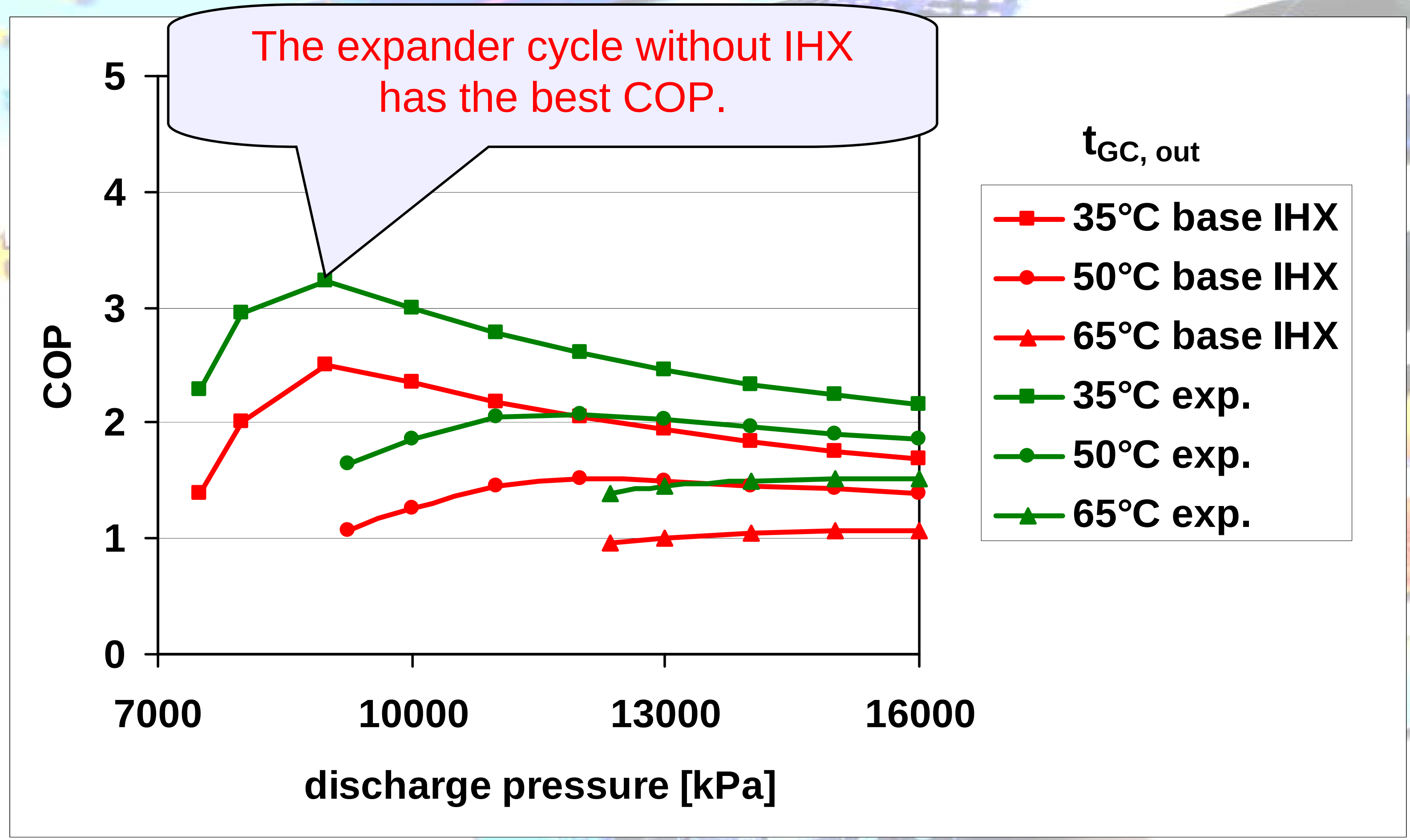


p_s = 3485 kPa / t_6 = 10°C / $\Delta t_{min,IHX}$ = 5 K / parameter = $t_{GC,\ out}$

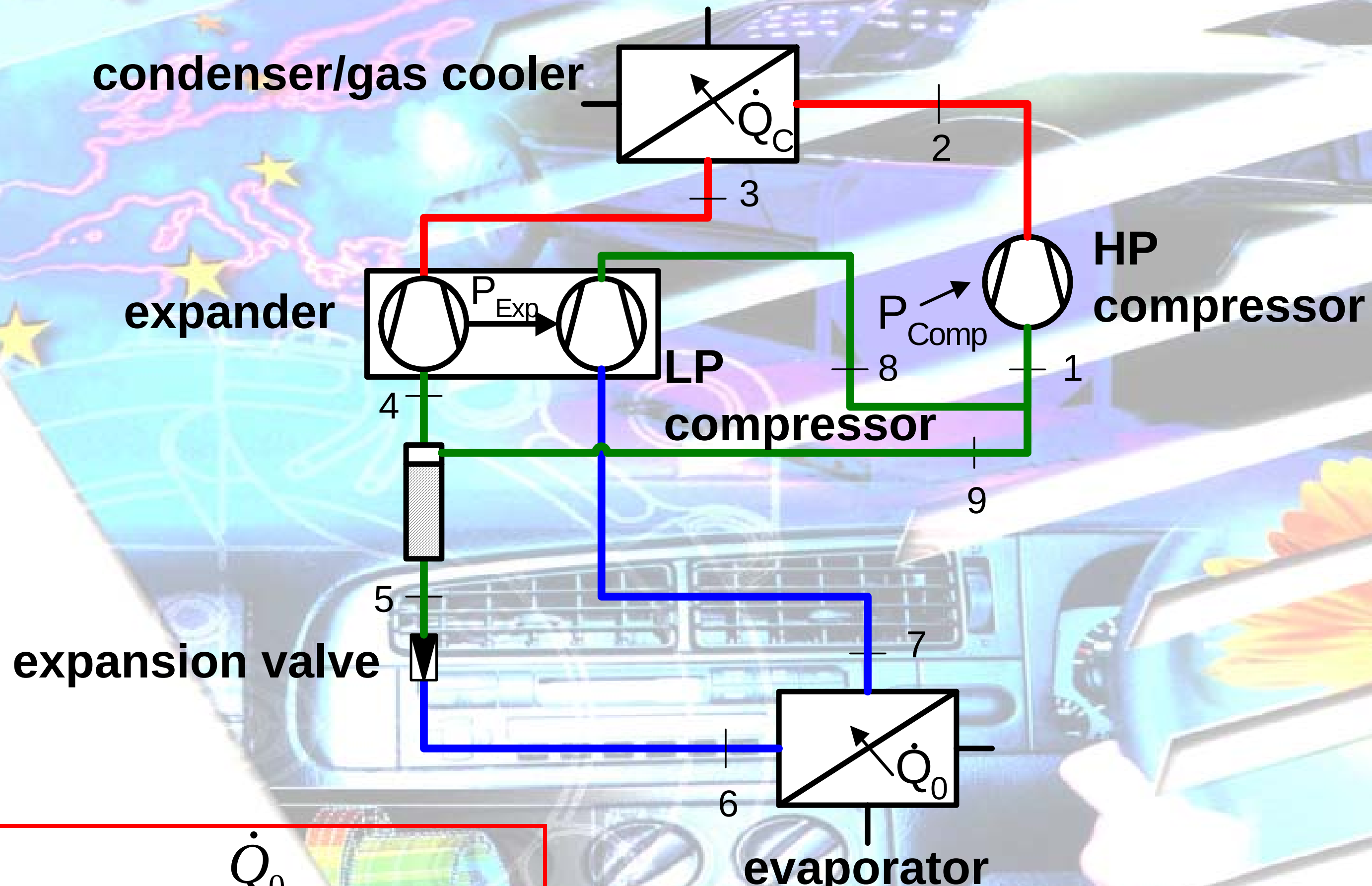
COP Comparison Expander Cycle w/o and w IHX



COP Baseline w / Expander Cycle w/o IHX



Expander Cycle with Intermediate Pressure



$$COP = \frac{\dot{Q}_0}{P_{Comp,HP} + P_{Comp,LP} - P_{Exp}}$$

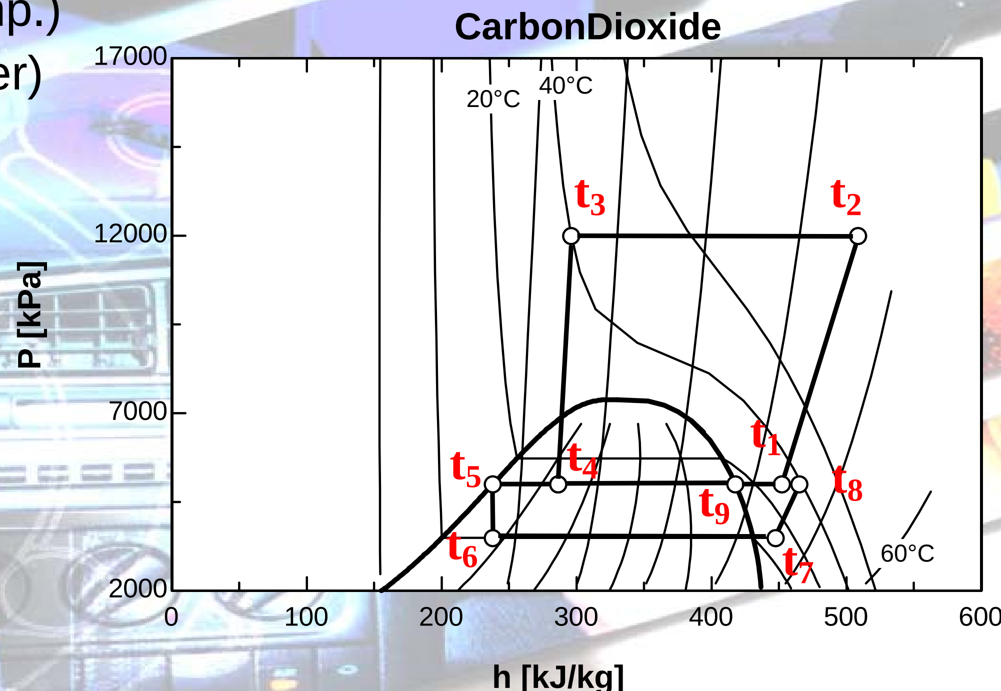
Boundary Conditions for CIP

- Fixed system conditions:

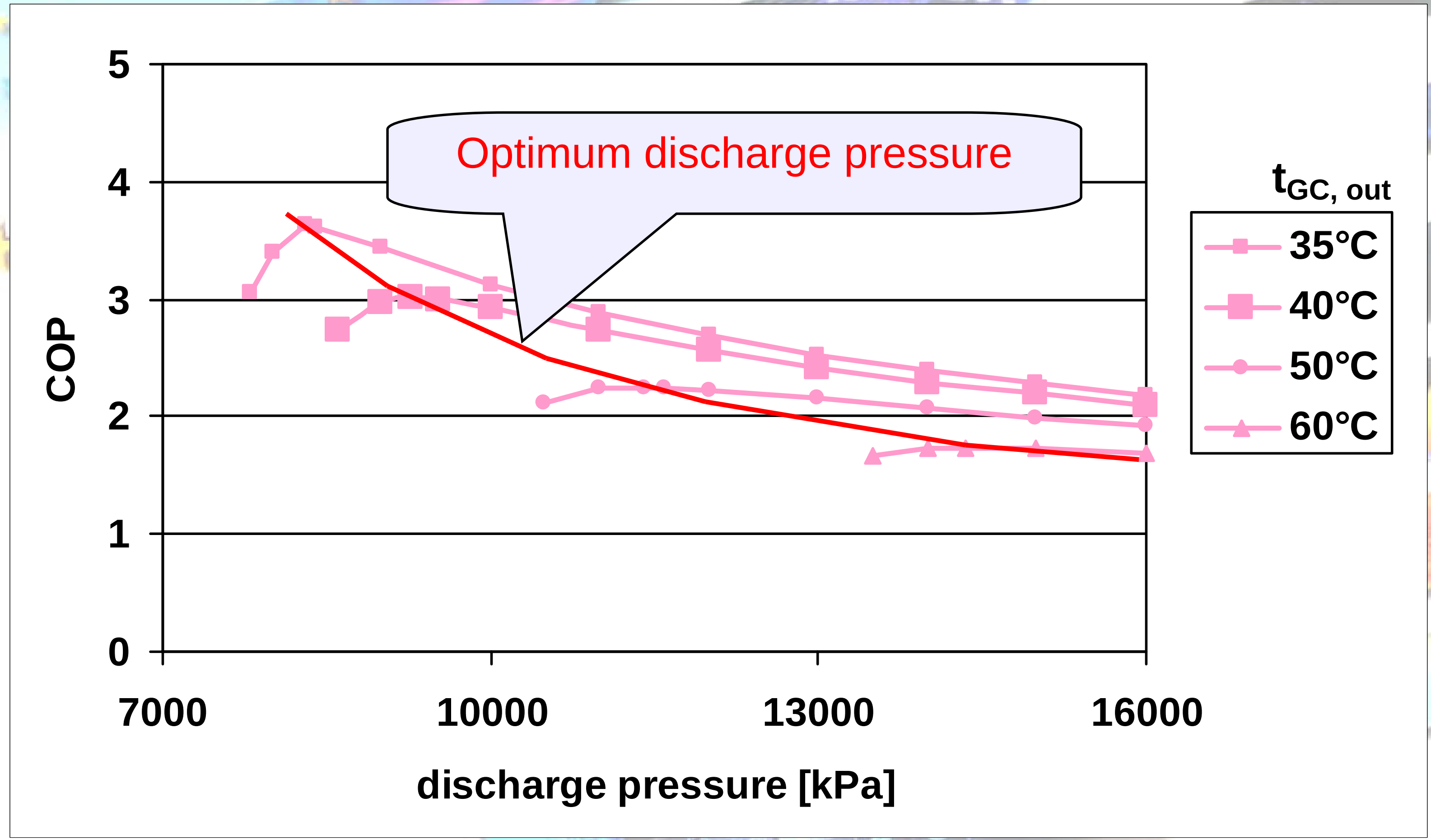
t_6	0°C (3485 kPa)
t_7	10°C
η_{is}	0,7 (HP compressor)
η_{is}	0,85 (expander, LP comp.)
η_{vol}	0,9 (comp. and expander)

- Variable system conditions:

temperature gas
cooler outlet ($t_{GC,out} = t_3$)
discharge pressure (p_d)
intermediate pressure (p_i)
$t_5=t_9=f(p_i)$

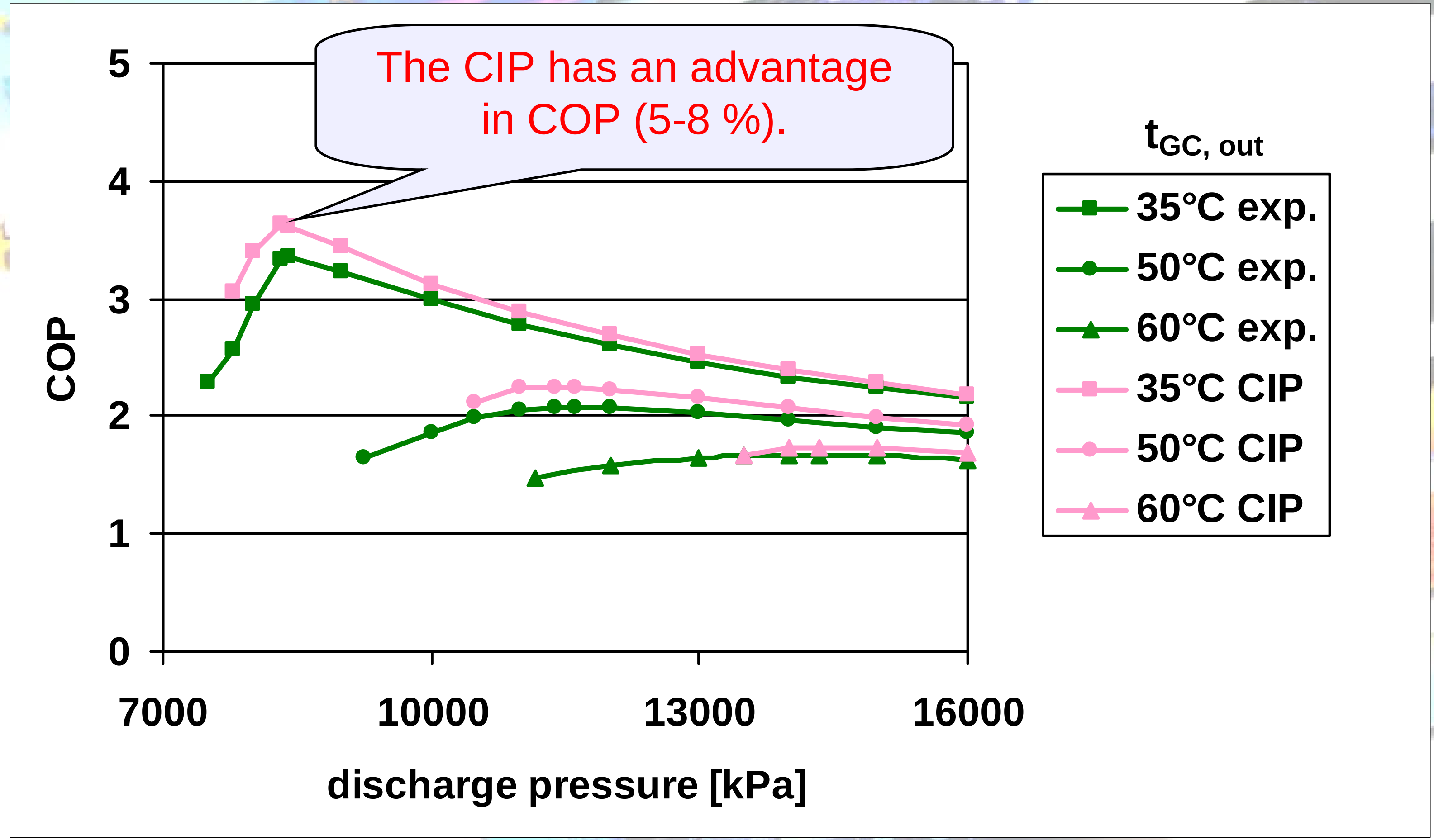


COP Cycle with Intermediate Pressure

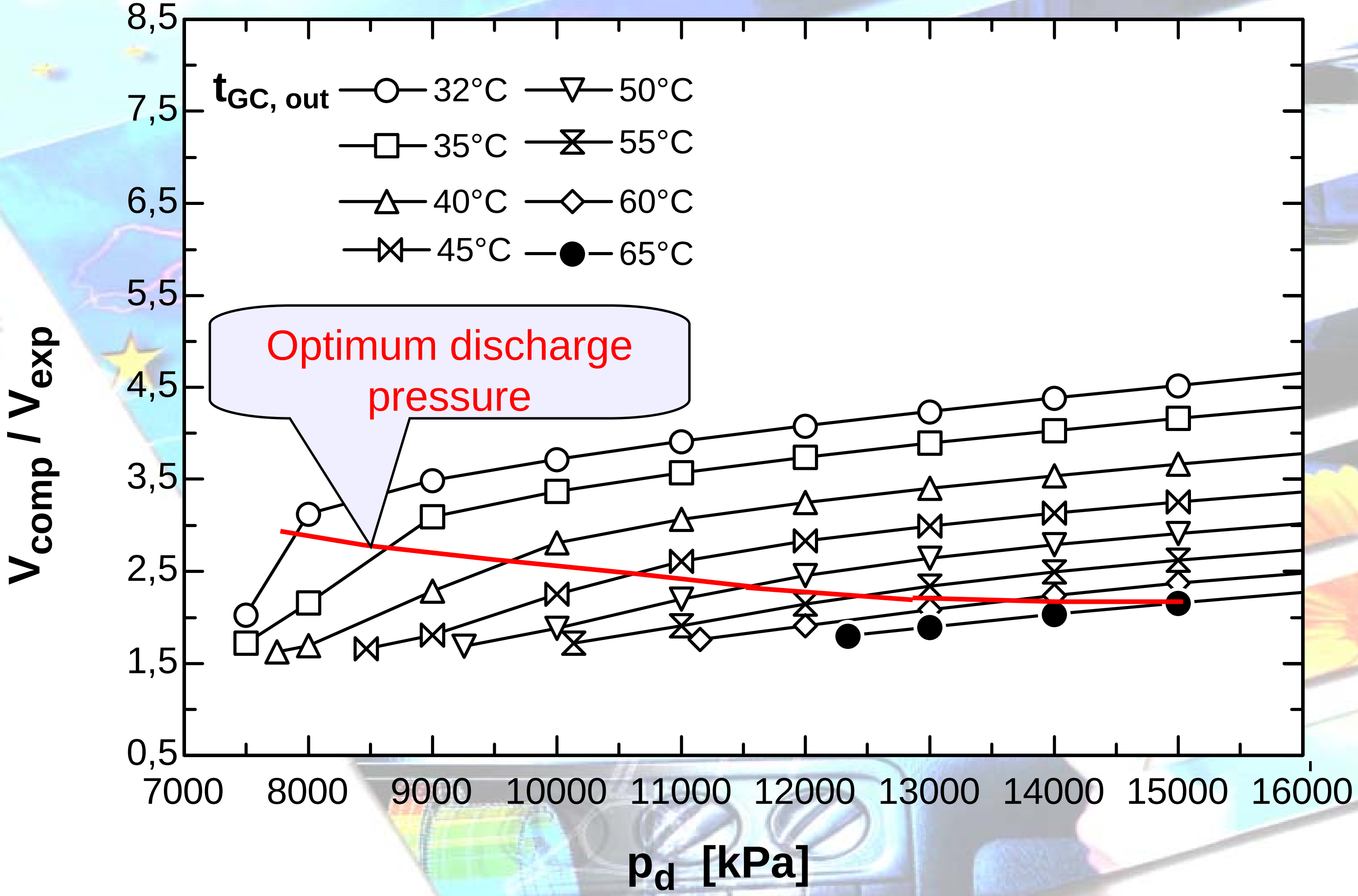


p_s = 3485 kPa / t_7 = 10°C / parameter = $t_{GC, out}$

COP Comparison Expander Cycle / CIP



Swept Volume Ratio - Expander Cycle

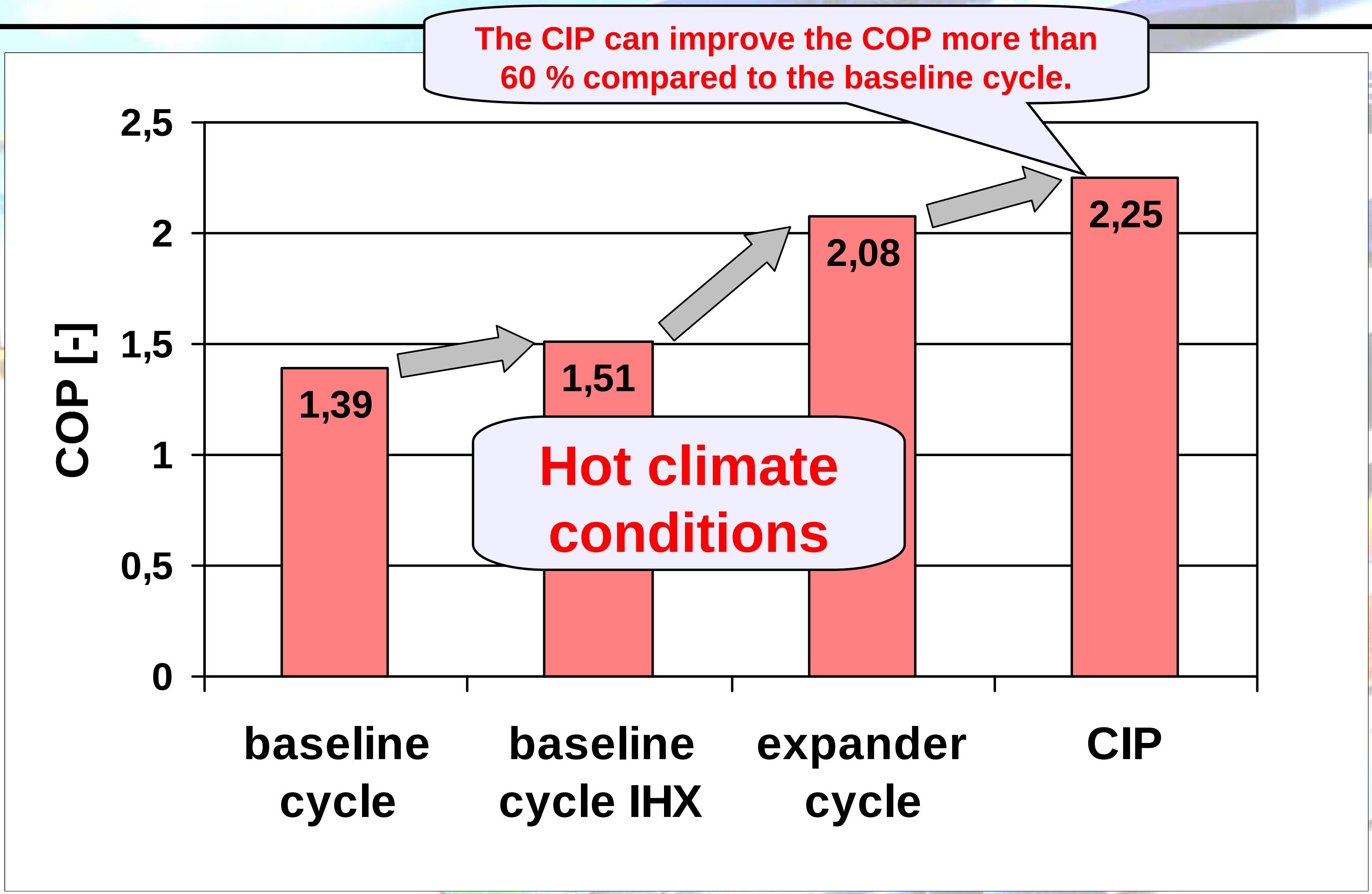


p_s = 3485 kPa / $t_6 = t_1$ = 10°C / parameter = $t_{GC,\ out}$

Conclusion - Volume Ratio

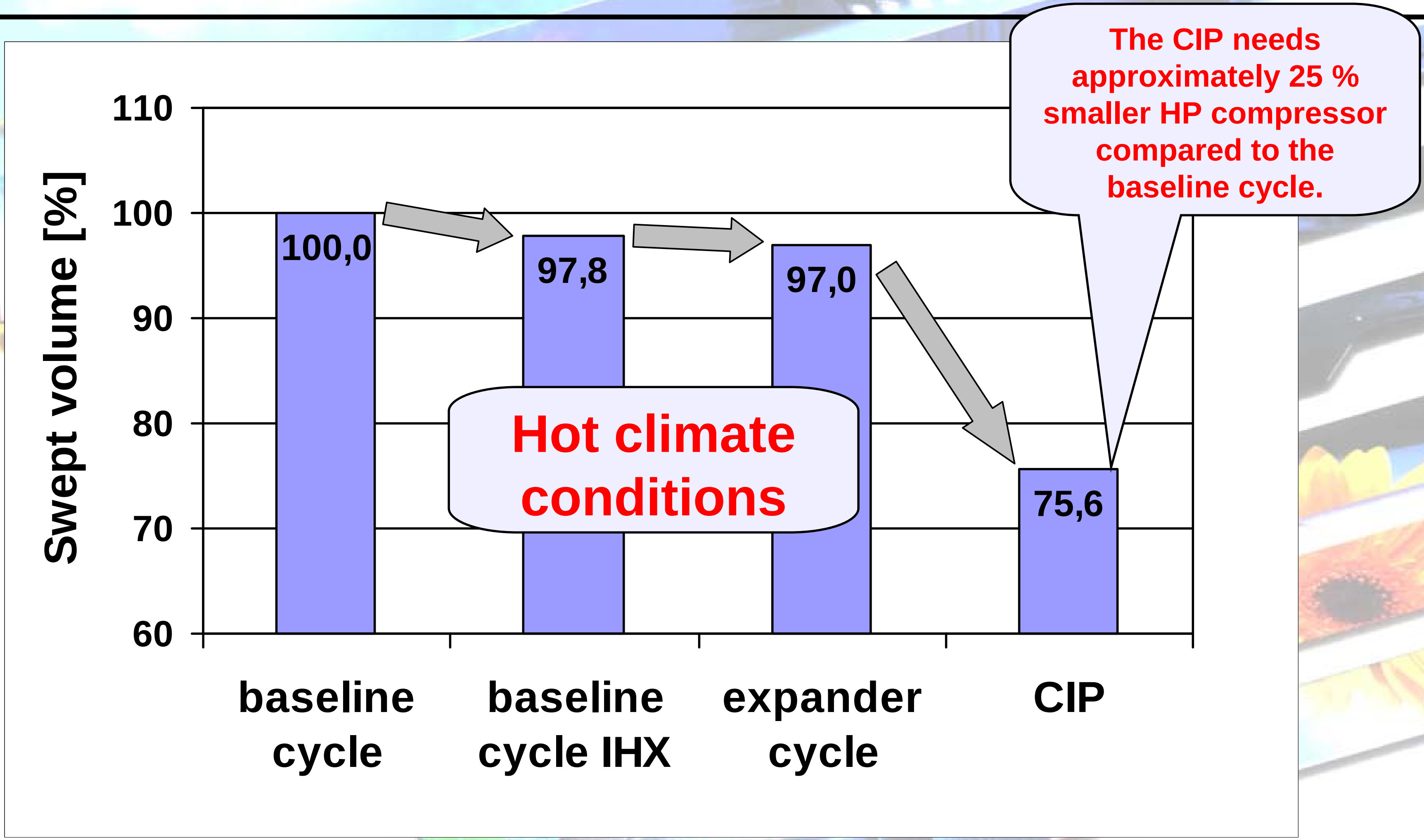
- Advantage:
  - To reach the optimum discharge pressure only small changes in swept volume ratio are required.
- Challenge:
  - Identify possible expander/compressor mechanisms
  - Cost of system components to find an optimum between COP and investment

Comparison COP



t_{ev}	0°C (3485 kPa)
$t_{ev, out}$	10°C
$t_{GC, out}$	50°C

Swept Volume of the Main Compressor



t_{ev}	0°C (3485 kPa)
$t_{ev, out}$	10°C
$t_{GC, out}$	50°C

Summary

	Baseline Cycle	Baseline Cycle with IHX	Expander Cycle	CIP
Advantages:	good at lower temperature	better COP at higher temperature than baseline cycle	can eliminate the IHX	hermetically design for LP compressor and expander → no shaft seal, no leakage
	simplest cycle		similar COP like CIP	lower pressure ratio → better compressor performance
			probably lower investment costs than CIP	smaller compressor swept volume for the high pressure side
				compact design possible → it can be mounted close to the evaporator → shorter pipes → smaller losses
Disadvantages:	compressor needs a bigger swept volume than the CIP to reach the same cooling capacity	compressor needs a bigger swept volume than the CIP to reach the same cooling capacity	compressor needs a bigger swept volume than the CIP to reach the same cooling capacity	one compressor in addition is required → additional costs
	high pressure ratio	high pressure ratio	high pressure ratio	
	worse COP at higher temperature	worse COP at lower temperature	longer pipes to the evaporator → losses	

Outlook

SANDEN is working on expander systems to fulfil the specific requirements for automotive applications.

⇒ update 2004