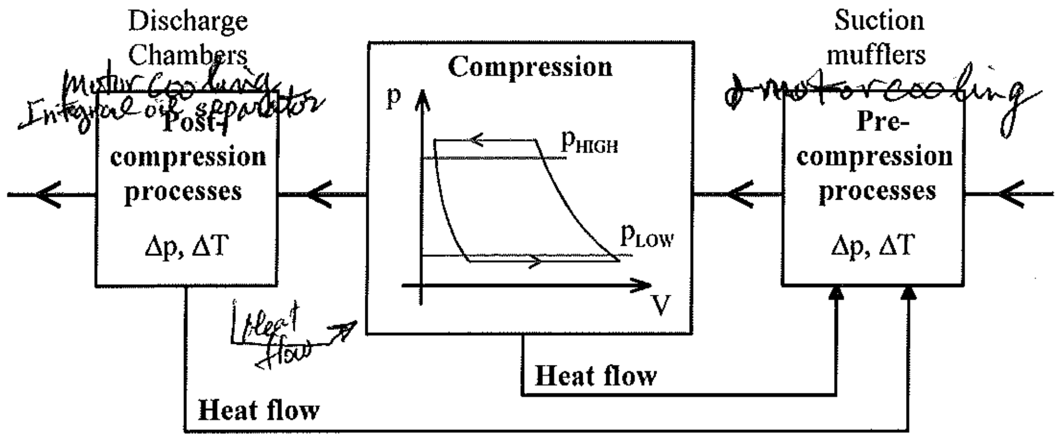
IDEALIZED COMPRESSOR



- Pre-compression processes (internal heating of suction gas and pressure drop in flow passages)
- Compression process (conversion of energy from work to internal energy and heat)
- Post-compression process (cooling of discharge gas and pressure drop in flow passages)

COMPRESSOR PERFORMANCE

- In how many ways can the steady state performance of a refrigeration compressor be described? Mfgr hide η_V & η_{is} typically
  - » Examples.... $\dot{Q}_E, \dot{W}_{in}$ $\dot{Q}_E, COP_C$

    η_{is}, η_V η_p, η_V

    $\dot{m}_R, \dot{W}_{in}, \dot{Q}_{loss}$

- How much information is needed to describe the performance of a refrigeration compressor at steady state operation?
  - » Need data on "capacity"
  - » Need data on "consumption of power"
  - » Need data on "rejection of heat" (T discharge)

Herrick Laboratories PURDUE UNIVERSITY

SYSTEM RELATED PROPERTIES

- In catalogues etc., the performance of a compressor is normally presented using system related properties like
  - » $\dot{Q}_E$: Refrigerating capacity [kW, Btu/h, Tons]
  - » $\dot{W}$: Power consumption [kW]
  - » COP : Coefficient Of Performance [-] $= \dot{Q}_E / \dot{W}$
- These properties are NOT independent of system parameters such as superheat and subcooling and are therefore, not generally suitable for comparing compressors and for modeling.
- Properties like Q_E, W and COP are often stated for a test condition (ASHRAE or ISO) with values for superheat and subcooling that are very different from the values obtainable in the actual system.

Herrick Laboratories PURDUE UNIVERSITY

PROCESS RELATED PROPERTIES

- Efficiencies are used for a more general description of the processes in a compressor.
  - » η_V : Volumetric efficiency [-] (Capacity)
  - » η_S : Isentropic efficiency [-] (Energy)
  - » η_P : Polytropic efficiency [-] (Energy) $f(T_s, P_s, P_D)$
- Efficiencies depend only on the compressor inlet and outlet conditions.
- Therefore, efficiencies are more general and suitable for comparing compressors and for modeling.

VOLUMETRIC EFFICIENCY (η_V)

- Ratio between the actual volume flow rate in the compressor inlet and the displacement rate of the compressor (maximum theoretical flow rate):

refrigerant mass flow rate

specific volume of inlet gas

$$\eta_V = \frac{\dot{V}_1}{\dot{V}_D} = \frac{\dot{m}_r v_1}{\dot{N}_c V_D}$$

(swept volume)

maximum volume of suction process

number intake strokes per unit time (e.g., compressor rotation rate)

- Then, the mass flow rate is determined by

$$\dot{m}_R = \rho_1 \eta_V \dot{N}_c V_D$$

density of inlet gas

Herrick Laboratories PURDUE UNIVERSITY

VOLUMETRIC EFFICIENCY (η_V)

- Ideally, the volumetric efficiency should be equal to one
- However, general phenomena reduce the volumetric efficiency
  - (1) » re-expansion of gas in clearance volume
  - » pressure drops in valves and flow passages directly affects (1) Internal vs external ?/p
  - » leakage from compression chamber
  - » internal superheat
  - » back-flow through valves
- The volumetric efficiency is influenced by factors such as pressure ratio, compressor speed, and heating of suction gas

VOLUMETRIC EFFICIENCY (η_V)

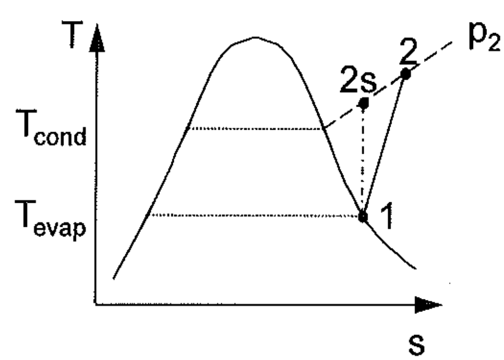
- For reciprocating compressors, the volumetric efficiency would normally be in the following intervals:
  - » AC-conditioning (low pressure ratios) [0.7 - 0.9]
  - » Cooling (medium pressure ratios) [0.6 - 0.8]
  - » Freezing (high pressure ratios) [0.4 - 0.7]
- For scroll and screw compressors, the volumetric efficiency depend less on the pressure ratio than for the reciprocating compressors. Since there are no valves in scroll and screw compressors, higher volumetric efficiencies can be achieved.
- For turbo compressors, the volumetric efficiency is not defined.

ISENTROPIC EFFICIENCY (η_S)

- Ratio of actual compressor power consumption to power consumption needed for an adiabatic and reversible process operating between the actual inlet state (p_1,v_1) and the actual outlet pressure (p_2) and with the actual mass flow rate

isentropic eff

$$\eta_s = \frac{\dot{m}_r \left(h_{2s} - h_1\right)}{\dot{W}_c}$$



- Isentropic efficiency is reduced by:
  - » mechanical friction $+I^2R$
  - » pressure drops
  - » heat transfer to the refrigerant
  - » re-compression due to $\eta_V < 1$

ISENTROPIC EFFICIENCY (η_S)

- The isentropic efficiency is influenced by factors such as pressure ratio, compressor speed and superheat of suction gas.
- For hermetic and semi-hermetic compressors the motor efficiency is included in an "overall isentropic efficiency".
- For reciprocating compressors, the isentropic efficiency would normally be in the following intervals:
  - » Small size hermetic compressors (domestic appl.) [0.4 - 0.6]
  - » Medium size (semi-) hermetic compressors [0.5 - 0.7]
  - » Large size open compressors (at full load) [0.6 - 0.8]

  robust Nick's Mr. Slim

  including motor losses
- For scroll and screw compressors, the built-in compression ratio makes it difficult to provide general intervals for isentropic efficiency. At design conditions, isentropic efficiencies can be higher than for reciprocating compressors

POLYTROPIC EFFICIENCY (η_P)

- Ratio of actual compressor power consumption to power consumption needed for a reversible polytropic process operating with the actual mass flow rate:

  Not much used in HVAC industry

$$\eta_p = \frac{\dot{m}_r w_p}{\dot{W}_c}$$

The polytropic process satisfies

$Pv^n = \text{constant}$

where n = polytropic coefficient

- The reversible, polytropic work is:

$$w_p = \underbrace{\int_{P_1}^{P_2} v\,dp}_{\text{true for a reversible SSSF compression process}} = P_1 v_1 \frac{n}{n-1}\left\{\left[\frac{P_2}{P_1}\right]^{\frac{n}{n-1}} - 1\right\}$$

POLYTROPIC EFFICIENCY (η_P)

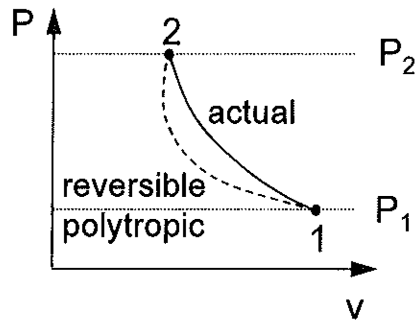
Two different approaches for choosing n

1. Based on actual outlet state (p_2,v_2)

$$n = \ln\left(\frac{p_2}{p_1}\right) \bigg/ \ln\left(\frac{v_1}{v_2}\right)$$

   » **n influenced by the real process (i.e., the effects of friction & heat transfer)**

   » **reversible polytropic process requires heat transfer to achieve the actual outlet state**



added

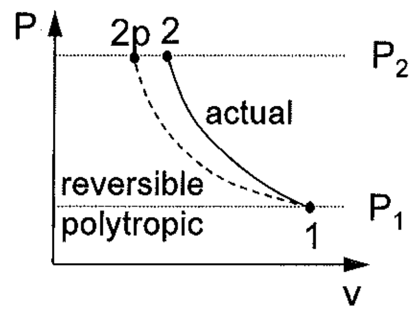
2. Based on specified n and actual outlet pressure (p_2)

   » **specifies an ideal outlet state (p_2,v_{2p}) similar to isentropic process**

   » **n depends on refrigerant and/or the assumed process**

      » n = k for isentropic ideal gas assumption

      » n =1 for isothermal ideal gas assumption



COMPRESSOR HEAT LOSS

- Volumetric and isentropic (or polytropic) efficiencies are not enough to completely describe the compressor performance.
- Information about the heat loss from the compressor is not included.
- Heat loss of a compressor is a consequence of internal losses and the temperature of the discharge gas being higher than the compressor surroundings.
- For compressors operating with medium and high pressure ratios, it can be necessary to cool them with air, water or liquid refrigerant in order to limit the temperature of the discharge gas (and oil).

COMPRESSOR HEAT LOSS

- Energy balance of compressor and compressor outlet state:

$$\dot{W} = \dot{m}(h_2 - h_1) - \dot{Q}_{LOSS} \qquad h_2 = f(T_2, p_2)$$

- Compressor heat loss can be expressed either directly as Q_{LOSS} or indirectly using discharge temperature T_2 or the heat loss ratio f_Q:

$$\dot{Q}_{loss} = UA\left(T_{comp} - T_A\right) \quad or \quad T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \quad or \quad f_Q = \frac{\dot{Q}_{loss}}{\dot{W}_{in}}$$

T_{COMP} = Average compressor surface temperature
T_A = Representative temperature for compressor surroundings
U = Overall heat transfer coefficient
A = Surface area of compressor

COMPRESSOR HEAT LOSS

- The following ranges for f_Q are generally applicable:
  - » Small hermetic compressors : 20 - 75 %
  - » Medium and large (semi-) hermetic compressors : 10 - 20 %
  - » Large open reciprocating compressors : 10 - 15 %
  - » Large screw compressors : (rejected to oil) 15 - 40 %
- The compressor heat loss is influenced by many factors
  - » pressure ratio
  - » temperature of surrounding air
  - » cooling water
  - » etc.
- Thus, large variations in f_Q can occur.

Herrick Laboratories
Purdue University

TYPES OF COMPRESSOR ANALYSIS
From "black-box" to "white-box"

BLACK-BOX

Black Box

not safe for extrapolation

- Statistical correlations for operation at test conditions (ISO and ARI)

$$\dot{Q}_E, \dot{W}, I, \dot{m} = f(T_E, T_C) = c_1 + c_2 T_E + c_3 T_C + c_4 T_E^2 + c_5 T_E T_C + c_6 T_C^2 + c_7 T_E^3 + c_8 T_C T_E^2 + c_9 T_E T_C^2 + c_{10} T_C^3$$

- Statistical correlations comparing performance with reference processes Grey Box

Types: Recip, Rotary, Scroll, Screw

$$\eta_{IS}, \eta_{VOL} = f(T_E, T_C) = c_1 T_E + c_2 T_C + c_3 T_E T_C + c_4 T_E^2 + c_5 T_C^2$$

- Process oriented models Improved Grey Box

$$\eta_{is}, \eta_V = f\left(p_2/p_1 ; V_{CV}/V_D ; \ldots \right)$$

- Phenomena oriented models (more than one control volume)
- Construction oriented models (many control volumes, dynamic models)
- Distributed models (partial differential equations...)

WHITE-BOX

Herrick Laboratories
Purdue University

CHOICE OF ANALYSIS

- Models suitable for system analysis
  - » statistical correlations for operation at test conditions
  - » statistical correlations comparing models with reference processes
  - » process oriented models
- Models suitable for compressor analysis
  - » phenomena oriented models
  - » construction oriented models
- Models suitable for detailed compressor analysis (even, individual parts of the compressor)
  - » construction oriented models
  - » distributed models

MODELS FOR SYSTEM ANALYSIS

- Most simple models are based on compressor maps providing data on cooling capacity, power consumption, COP and mass flow rate for a limited number of operating conditions.
  - » General expressions can be used for data fitting.
- More general models can be developed from the same compressor maps, if the data is used to calculate efficiencies.
  - » General expressions, or more complex expressions, can be used for data fitting.
- Process oriented models can also be developed from compressor maps, but will require more detailed specifications for compressor (clearance volume, internal pressure drops, etc.)

MAP-BASED COMPRESSOR MODEL

- Compressor performance maps are compressor and refrigerant specific
- Input variables:
  - » Evaporation temperature (suction pressure p_1)
  - » Condensation temperature (discharge pressure p_2)
- Ten term ARI 540 correlation:

$$\dot{Q}_E, \dot{W}, \dot{m} = f\left(T_E, T_C\right) = c_1 + c_2 T_E + c_3 T_C + c_4 T_E^2 + c_5 T_E T_C + c_6 T_C^2 + c_7 T_E^3 + c_8 T_C T_E^2 + c_9 T_E T_C^2 + c_{10} T_C^3$$

- Discharge temperature (if data is available) can be fitted as well
- Superheat and subcooling temperature are fixed

MAP-BASED COMPRESSOR MODEL

- Coefficients C_1 - C_{10} are usually available from compressor manufacturers
- Corrections for other superheat temperatures than specified in the compressor map have to be made (Rice - PUREZ):

$$\frac{\dot{m}_{new}}{\dot{m}_{map}} = 1 + F\left(\frac{\rho_{S,new}}{\rho_{S,map}} - 1\right)$$

$$\frac{\dot{W}_{new}}{\dot{W}_{map}} = \frac{\dot{m}_{new}}{\dot{m}_{map}} \frac{\Delta h_{is,new}}{\Delta h_{is,map}}$$

F = Correction factor
($F = 0.75$ is recommended)
$\dot{m}$ = mass flow rate
ρ_S = suction density
$\dot{W}$ = compressor power consumption
Δh_{is} = isentropic enthalpy change
Subscript "new" = at new superheat
Subscript "map" = at map-based superheat

MODELS FOR COMPRESSOR ANALYSIS

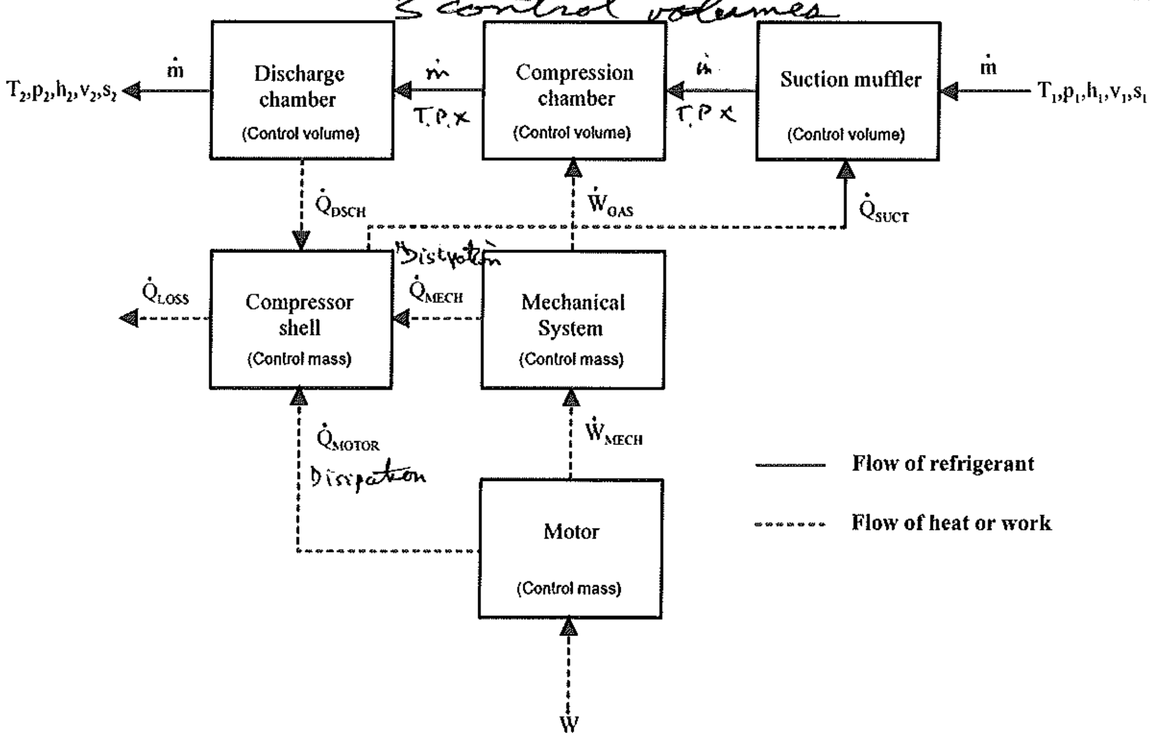
- Phenomena oriented models require knowledge about compressor construction and to some extent also experimental data with measurements of internal temperatures.
- Construction oriented models are dynamic models requiring detailed knowledge about the compressor construction
  - » Validation of construction oriented models requires experimental data with measurements of internal temperatures and pressures (dynamic measurements).
- Distributed dynamic models require very detailed knowledge about the individual parts of the compressor.
  - » Validation requires advanced measurement techniques like laser Doppler anemometry (LDA) and Thermography.

ADVANCED COMPRESSOR MODELS

- Phenomena oriented compressor models can be expanded into construction oriented compressor models by introducing more control volumes.
- Modeling of processes in each control volume then becomes a matter of describing
  - » Local changes in refrigerant properties caused by heat transfer, pressure changes, mixing with oil, etc.
  - » Friction losses in bearings
  - » Valve movement (reciprocating compressors)
  - » Movement of individual parts in mechanical systems

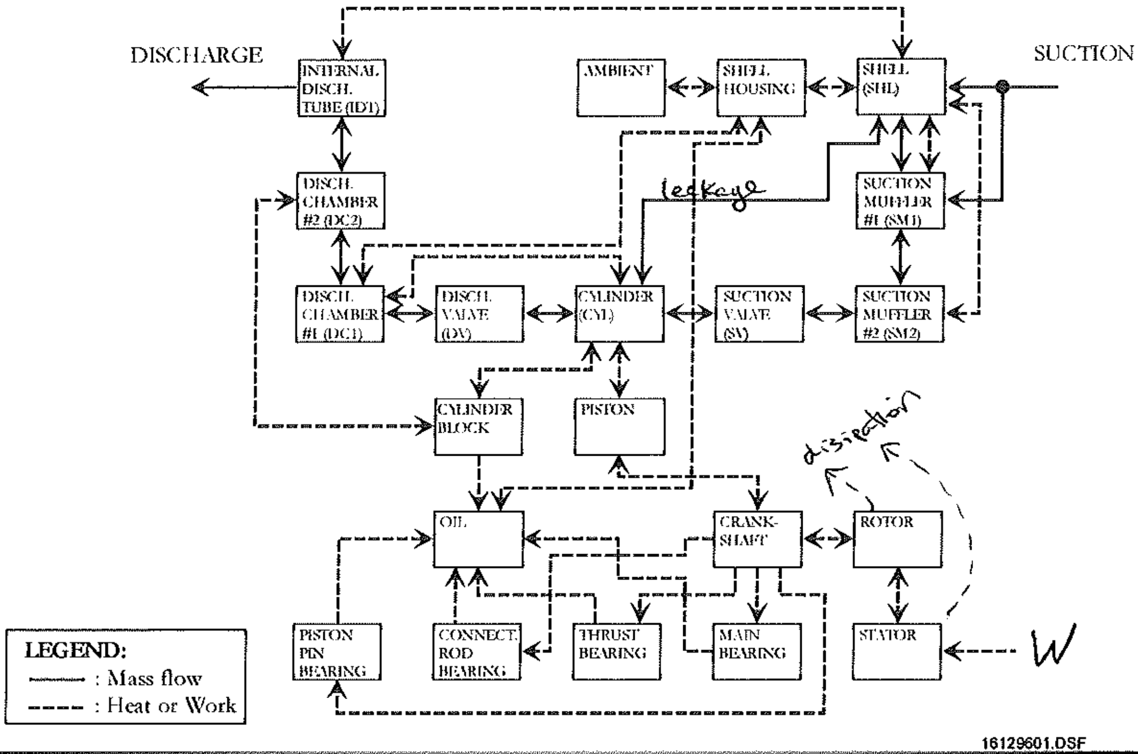
PHENOMENA ORIENTED MODEL

3 control volumes

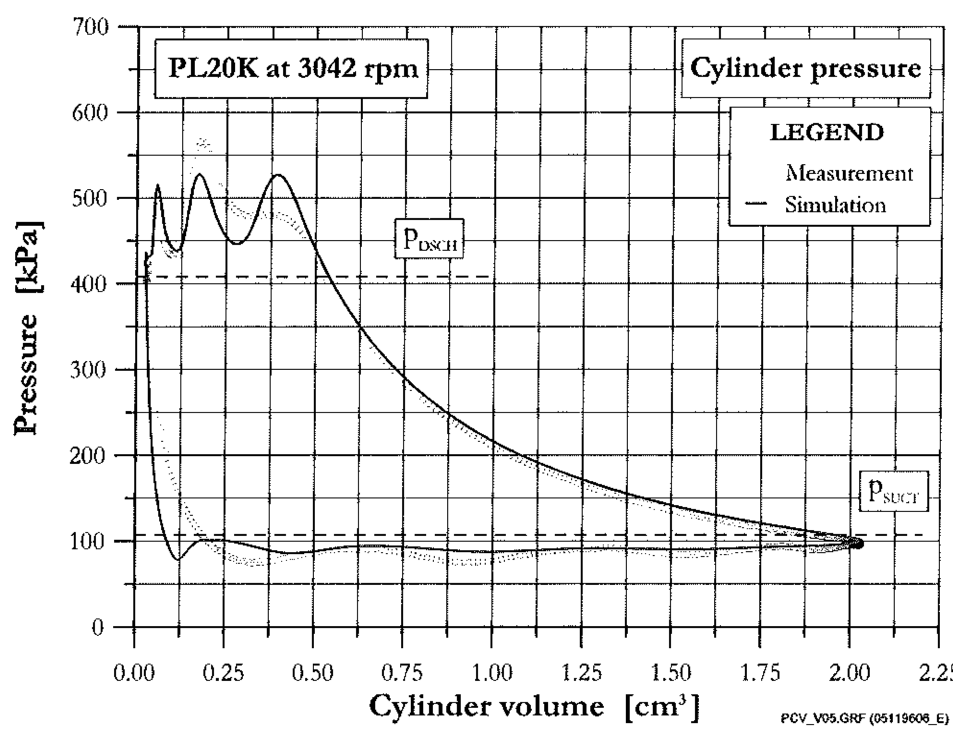


CONSTRUCTION ORIENTED MODEL

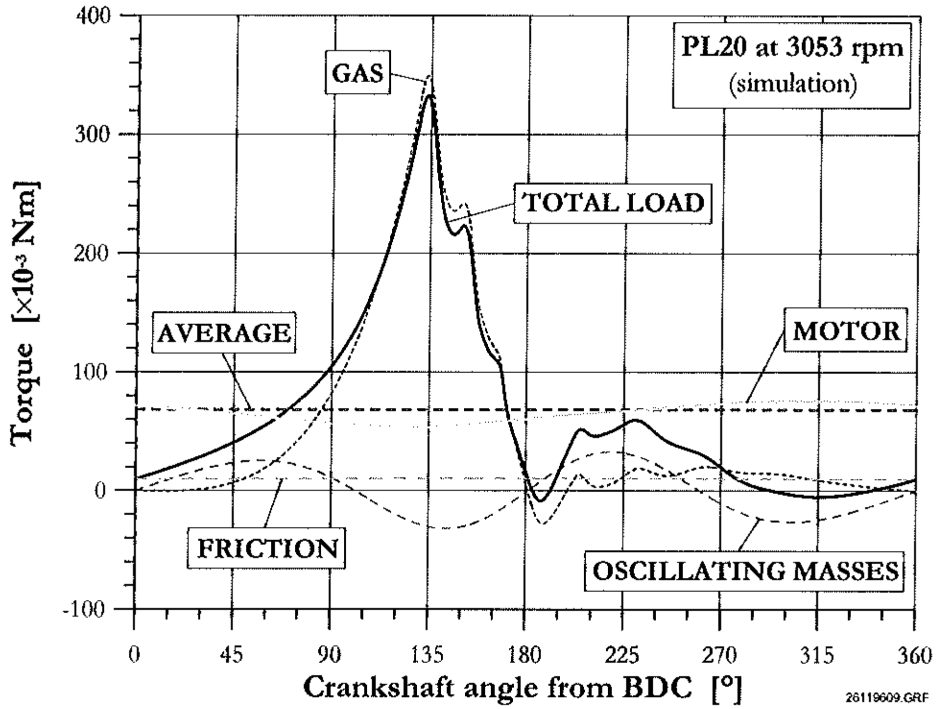
Small scale hermetic reciprocating compressor



CONSTRUCTION ORIENTED MODEL



CONSTRUCTION ORIENTED MODEL



SUMMARY

- Review of compressor types
- Introduction to processes inside compressors
- Different ways of expressing compressor performance
- Categorization of compressor models
- Choice of compressor model for system modeling and for compressor modeling
- Examples of advanced models (phenomena and construction oriented models)

BOOK REFERENCES

- General book on refrigeration with excellent chapters on compressors
  - » "Industrial Refrigeration Handbook", Wilbert F. Stoecker, McGraw-Hill, 1998 (www.books.mcgraw-hill.com)
- General compressor book
  - » "Industrial Compressors", Peter O'Neill, Butterworth-Heinemann, 1997 (www.bh.com)
- Specialized compressor books
  - » Proceedings of International Compressor Engineering Conferences at Purdue University, 1972 - 2012
  - » "Strömung und Ventilplattenbewegung in Kolbenverdichtern", Leopold Böswirth, Published by writer, 1998 (L. Böswirth, Argentinierstrasse 28/7, A-1040 Vienna, Austria)

LIST OF NOMENCLATURE

$c_{1..10}$	Constants	[various]
COP	Coefficient Of Performance	[-]
f_Q	Compressor heat loss ratio	[-]
$f_{?,CAP}$	Correction factor for cylinder unloading	[-]
$f_{?,REV}$	Correction factor for compressor speed	[-]
h_1	Enthalpy of refrigerant in compressor inlet	[kJ/kg] or [Btu/lb]
h_2	Enthalpy of refrigerant in compressor outlet	[kJ/kg] or [Btu/lb]
$h_{2,S}$	Enthalpy of refrigerant in compressor outlet assuming reversible and adiabatic compression	[kJ/kg] or [Btu/lb]
k_{CAP}	Relative stroke volume capacity	[-]
k_{REV}	Relative compressor speed	[-]
I_{MOTOR}	Motor current	[A]
$\dot{m}$	Mass flow of refrigerant	[kg/s] or [lb/s]
N	Compressor speed	[s^{-1}]
n	Polytropic exponent	[-]
p_1	Pressure in compressor inlet	[kPa] or [psi]
p_2	Pressure in compressor outlet	[kPa] or [psi]

LIST OF NOMENCLATURE

$\dot{Q}_E$	Cooling capacity	[kW] or [Btu/h]
$\dot{Q}_{LOSS}$	Compressor heat loss	[kW] or [Btu/h]
T_1	Suction gas temperature	[°C] or [°F]
T_2	Discharge gas temperature	[°C] or [°F]
T_A	Temperature of compressor surroundings	[°C] or [°F]
T_{COMP}	Temperature of compressor surface	[°C] or [°F]
T_C	Condensing temperature	[°C] or [°F]
T_E	Evaporation temperature	[°C] or [°F]
V_{CV}	Compressor clearance volume	[m^3] or [ft^3]
V_D	Compressor displacement	[m^3] or [ft^3]
$\dot{V}_D$	Compressor displacement rate	[m^3/s] or [ft^3/s]
$\dot{V}_s$	Volume flow in compressor inlet	[m^3/s]
v_1	Specific volume of refrigerant in compressor inlet	[m^3/kg] or [ft^3/lb]
$\dot{W}$	Compressor power consumption	[kW]
η_{IS}	Isentropic efficiency	[-]
η_P	Polytropic efficiency	[-]
η_{VOL}	Volumetric efficiency	[-]