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A Generalized Approach Towards Compressor Performance Analysis

P. N. Pandeya

W. Soedel

A GENERALIZED APPROACH TOWARDS COMPRESSOR PERFORMANCE ANALYSIS

Prakash Pandeya
Graduate Research Assistant

Werner Soedel
Professor of Mechanical Engineering

Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University
West Lafayette, Indiana

INTRODUCTION

The term "performance of a compressor" means different things to different people in the compressor industry in general. For instance, isothermal efficiency, adiabatic efficiency, mechanical efficiency, clearance volumetric efficiency, overall volumetric efficiency are some of the terms most commonly used in the air and gas compressor industry [1]. In the refrigeration and air conditioning industry, some of the most common terms used to indicate the compressor performance are coefficient of performance (COP), energy efficiency ratio (EER), performance factor, and relative efficiency, besides other less common terms such as compression efficiency, volumetric efficiency, and mechanical efficiency, etc. [2,3]. None of these definitions of performance either combined together or separately give the total picture of the performance of the compressor. The multiplicity of these definitions creates a lot of confusion in analyzing the performance of a compressor, especially if one were to compare the performance of two different designs of compressors.

We will attempt to concentrate on the compressor by proposing a new term, which we define as Efficiency of Performance (η_p), that should take into consideration every factor that can affect the performance of a compressor. A methodology will be developed to analyze and obtain η_p. Relationship between η_p, COP, and EER will be established and an experimental set-up to obtain η_p will be described. The analysis will be carried out only for refrigerating compressors, although with slight modification it will be applicable to air and gas compressors as well.

COEFFICIENT OF PERFORMANCE (COP)

COP and EER are the most widely used terms today in the refrigeration and air conditioning industry. The COP of a refrigeration machine has been defined in literature [3,4] as the ratio of the heat removed or cooling produced in Btu/hr to the work required in Btu/hr. This is a dimensionless quantity and for a system operating under ideal conditions between the evaporator temperature T_1 and the condenser temperature T_2 with no subcooling it becomes the same as the COP of the Carnot cycle, operating between the same evaporator and condenser temperatures, and is then given by the ratio of T_1 to $(T_2 - T_1)$ [9]. Another term commonly used in industry is known as EER (energy efficiency ratio) and is obtained when the units of the work required are changed from "Btu/hr" to "watts" in the original definition of COP. Thus, EER is not a dimensionless term. As will be clear from the definition, both COP and EER give an idea of the performance of the whole system which consists of evaporator, condenser, expansion device, and connecting lines besides the compressor. Since the performance of any system depends upon the performance of each component of the system to some extent, it is obvious that the system performance should not be considered the same as the performance of any single component. Unfortunately, the refrigeration and air conditioning industry has generally not found it necessary to differentiate between the system performance and the compressor performance possibly because the compressor is the single most important part of any vapor compression type refrigeration or air conditioning system. The result is that COP and EER are being widely used by the industry to indicate the performance of compressors. While this approach may seem to be convenient for the purposes of rating of the compressors, post-production testing and consumer information, it is not very helpful in the objective evaluation of any particular compressor design either at the design stage or for a comparative study of various designs for the following two reasons: (1) While the experimentally determined values of COP or Performance Factor or EER of a compressor do indicate the capacity of <u>one</u> particular sample of a compressor, they do not give any idea of the various energy and mass flow losses occurring in the compressor that affect these factors ultimately. (2) Secondly, unless an investigation of such losses is carried out in detail, it will not be possible to decide as to whether a particular design concept itself is responsible for the low or high values of these factors or it is a poor design practice that is affecting the performance of the compressor one way or the other. In other words, the concept of optimum design should be introduced for a complete and most generalized analysis and evaluation of compressor performance.

Thus, it becomes necessary to evolve a generalized and comprehensive definition of a term that can be used to relate the performance of a compressor in its every aspect and which would take into account all the factors affecting it. Few attempts have been made in the past by some investigators [5,6,7] in this direction. Reference [5] treats compressors in general and considers the non-dimensionalized brake horsepower (given by the area of the non-dimensional P-V diagram where the axes are V/V_s and P/P_d respectively. V_s and P_d are the swept volume and discharge pressure respectively.) of the compressor as the criterion of its performance. The formula is based on a theoretical P-V diagram and the losses occurring in an actual compressor have not been taken into account. Reference [6] looks at the performance of hermetic refrigerator compressors from the point of view of energy consumption and energy losses only. The effect of mass flow losses was not considered. Reference [7] goes a step further and takes into account the volumetric efficiency also in defining the compressor efficiency. Some of the most common losses and their effects on the compressor efficiency have been pointed out and a method to obtain compressor efficiency from the calorimeter test results has been outlined. However, this approach too falls short of giving a complete explanation of how these results could be arrived at analytically rather than going through an experimental set-up. In the following, we will attempt to devise a scheme which will take care of these questions also.

PERFORMANCE CRITERION

Before arriving at a logical performance criterion of a machine, it is necessary to outline the objectives in defining the performance itself. In very general terms, the performance of a machine is an evaluation of the ability of the machine to accomplish the task it has been assigned to do. In case of a compressor, its task is to pump the maximum possible quantity of gas from the given suction conditions to the desired discharge conditions with the least amount of energy consumption. Thus, two useful criteria for performance emerge:

(1) The capacity of the compressor, that is, the mass flow rate it can compress and deliver at the given operating conditions. An increase in mass flow rate will improve the performance.

(2) The effective utilization of the energy supplied to the compressor, that is, the energy consumption per unit mass delivered. An increase in energy consumption will reduce the performance.

Mathematically, the two criteria can be written as follows:

(1) Performance $\propto$ mass flow rate

(2) Performance $\propto \dfrac{1}{\text{energy consumed per unit mass delivered}}$

Combining the two criteria into one, we get:

$$\text{Performance} \propto \frac{\text{mass flow rate}}{\text{energy consumed per unit mass delivered}} \quad (1)$$

Equation (1) thus gives a fundamental relation that defines the basic performance criterion of a compressor. To convert it to a more usable form, we remove the proportionality sign and define a new term Performance Ratio (π) thus:

$$\text{Performance Ratio } (\pi) = \frac{\frac{dm}{dt}}{\frac{dE_{in}}{dm}} \quad (2)$$

where E_{in} represents energy input to the compressor. Equation (2) can be rewritten as:

$$\pi = \frac{\frac{dm}{dt}}{\frac{dE_{in}}{dt} \Big/ \frac{dm}{dt}} = \frac{\left(\frac{dm}{dt}\right)^2}{\frac{dE_{in}}{dt}} \quad (3)$$

The Performance Ratio as defined above is a dimensional term and, therefore, it still has a drawback inasmuch as it does not give us a feel for the best possible performance. To alleviate this situation, we define another term, which we call Efficiency of Performance (η_P), and which is obtained by comparing the actual performance ratio of a compressor with the theoretical one that would be obtained if the compressor was running under the ideal running conditions with no losses whatsoever. Thus using "i" and "a" as the subscripts for "ideal" and "actual" conditions and (Δh_i) as the increase in the enthalpy of the gas per unit mass under ideal running conditions, we can develop the following relationships:

$$\pi_a = \frac{\left(\frac{dm}{dt}\right)_a^2}{\left(\frac{dE_{in}}{dt}\right)_a} \quad (4)$$

$$\pi_i = \frac{\left(\frac{dm}{dt}\right)_i}{\Delta h_i} \quad (5)$$

$$\eta_p = \frac{\pi_a}{\pi_i} = \frac{\left(\frac{dm}{dt}\right)_a^2}{\left(\frac{dE_{in}}{dt}\right)_a} \times \frac{\Delta h_i}{\left(\frac{dm}{dt}\right)_i} \quad (6)$$

or
$$\eta_p = \frac{\left(\frac{dm}{dt}\right)_a}{\left(\frac{dm}{dt}\right)_i} \times \frac{\left(\frac{dm}{dt}\right)_a \Delta h_i}{\frac{dE_{in}}{dt}} \quad (7)$$

or
$$\eta_p = \eta_{ma} \times \eta_e \quad (8)$$

where
$$\eta_{ma} = \frac{\left(\frac{dm}{dt}\right)_a}{\left(\frac{dm}{dt}\right)_i} \quad (9)$$

$$\eta_e = \frac{\left(\frac{dm}{dt}\right)_a \Delta h_i}{\frac{dE_{in}}{dt}} \quad (10)$$

As it turns out, we see from Equation (8) that efficiency of performance is the product of two other terms, which we define as mass flow efficiency (η_{ma}) and energy efficiency (η_e), respectively, both of which are dimensionless terms. Philosophically speaking, unity defining the ideal condition, efficiency of performance (η_p) can be defined as the fraction of the ideal performance that can be achieved by a given compressor under actual working conditions. Mass flow efficiency (η_{ma}) can be defined as the fraction of the ideal mass flow rate that can be pumped by the compressor under actual working conditions. Similarly, energy efficiency can be defined as the fraction of the ACTUAL power consumed that would have been consumed had the compressor been running ideally. As it turns out, the efficiency of performance can be broken down further into other well known terms as follows:

$$\eta_p = \frac{\left(\frac{dm}{dt}\right)_a}{\left(\frac{dm}{dt}\right)_i} \times \frac{\Delta h_i}{\Delta h_a} \times \frac{\left(\frac{dm}{dt}\right)_a \Delta h_a}{\frac{dW}{dt}} \times \frac{\frac{dW}{dt}}{\frac{dE_{in}}{dt}} \quad (11)$$

or
$$\eta_p = \frac{\dot{m}_a}{\dot{m}_i} \times \frac{\Delta h_i}{\Delta h_a} \times \frac{\dot{m}_a \Delta h_a}{\dot{W}} \times \frac{\dot{W}}{\dot{E}_{in}} \quad (12)$$

We notice some familiar terms in Equation (12). For example, $(\Delta h_i/\Delta h_a)$ is the adiabatic compressor efficiency, $(\dot{W}/\dot{E}_{in})$ is the motor efficiency, and $(\dot{m}_a \Delta h_a/\dot{W})$ is nothing but the mechanical efficiency. The last one will be more obvious if we realize that $(\dot{m}_a \Delta h_a)$ is the actual increase in the total enthalpy of the gas inside the compressor cylinder per unit time and $\dot{W}$ is the power converted into work by the motor. Thus, we can rewrite (12) as follows:

$$\eta_p = \eta_{ma} \times \eta_c \times \eta_m \times \eta_{motor} \quad (13)$$

where η_{ma} = mass flow efficiency as defined earlier

$= \dfrac{\text{actual mass flow rate}}{\text{ideal mass flow rate}}$

η_c = adiabatic compressor efficiency

$= \dfrac{\text{ideal rise in specific enthalpy during compression process}}{\text{actual rise in specific enthalpy during compression process}}$

η_m = mechanical efficiency

$= \dfrac{\text{actual power delivered to the gas}}{\text{actual shaft work}}$

η_{motor} = motor efficiency

$= \dfrac{\text{actual shaft work}}{\text{electrical energy input}}$

It should be kept in mind that the mass flow efficiency (η_{ma}) does take into account all the efficiencies connected with the mass flow, such as volumetric efficiency (η_v), leakage, suction gas heating, etc. Equation (13) clearly indicates that the efficiency of performance, although obtained from a very basic definition of performance, does in fact encompass every aspect of the compressor performance and is, therefore, its true performance criterion.

The interrelationship of all of these efficiencies is not just for academic exercise. In fact, it gives an insight into how each term affects the total performance of a compressor and it may be a worthwhile exercise to evaluate each term separately to find out which of the terms reduce the efficiency of performance most.

ANALYTICAL DETERMINATION OF η_p

The relationship of η_p as given above still does not tell us much about how it is affected by a particular compressor design and, therefore, needs to be further analyzed to solve this problem. To do this, let us analyze Equation (7) term by term. Thus:

$$\eta_p = \frac{\dot{m}_a}{\dot{m}_i} \times \frac{\dot{m}_a \Delta h_i}{\dot{E}_{in}} \tag{14}$$

The four apparent unknowns in this relationship are the ideal mass flow rate ($\dot{m}_i$), actual mass flow rate ($\dot{m}_a$), the increase in specific enthalpy of gas during compression under ideal running conditions (Δh_i) and actual energy rate or power input to the compressor ($\dot{E}_{in}$). However, $\dot{m}_i$ is in reality a known quantity as it depends only on the compressor geometry and the "known" suction conditions and is, thus, given by:

$$\dot{m}_i = \frac{VN\rho_s}{60} \tag{15}$$

where V, N, and ρ_s are the cylinder displacement, RPM, and the suction gas mass density respectively, all of which are known. Furthermore, Δh_i will also be known once we define the ideal compression process. In hermetic compressors, isentropic compression should be regarded as the best that might be possible and hence it should be treated as the reference or ideal process [4,8]. Referring to Figure 1, since the suction conditions and discharge pressure are known, Δh_i can be easily found from the pressure-enthalpy chart or tables of the gas. Thus:

$$\Delta h_i = h_d - h_s \tag{16}$$

where h_d and h_s are the specific enthalpies of the gas at the discharge point and suction point respectively for the isentropic compression process. $\dot{m}_a$ and $\dot{E}_{in}$ can be expressed in terms of $\dot{m}_i$ and Δh_i as follows:

$$\dot{m}_a = \dot{m}_i - \Sigma\dot{m}_L \tag{17}$$

$$\dot{E}_{in} = (\Delta h_i)\dot{m}_a + \Sigma\dot{E}_L$$

or $$\dot{E}_{in} = (\Delta h_i)(\dot{m}_i - \Sigma\dot{m}_L) + \Sigma\dot{E}_L \tag{18}$$

where $\Sigma\dot{m}_L$ and $\Sigma\dot{E}_L$ are the total mass flow rate loss and the total power input loss respectively. Combining (14), (15), (16), (17), and (18), we finally get:

$$\eta_p = \frac{\left(\frac{VN\rho_s}{60} - \Sigma\dot{m}_L\right)^2 (h_d - h_s)}{\left(\frac{VN\rho_s}{60}\right)\left[(h_d - h_s)\left(\frac{VN\rho_s}{60} - \Sigma\dot{m}_L\right) + \Sigma\dot{E}_L\right]} \tag{19}$$

Energy Losses

The only two unknowns in the Equation (19) are $\Sigma\dot{m}_L$ and $\Sigma\dot{E}_L$. Thus, the problem has now been reduced to obtaining the mass flow and energy losses. This makes it necessary to first look into the whole process qualitatively in order to define the various loss mechanisms. Refer to Figure 2 showing the energy balance of the major components of the compressor through which the power is ultimately transferred to the discharge gas. Other components of the compressor such as shell, suction, and discharge tubes, etc., are not directly involved in the transmission of power although they do take part in heat transmission. Thus, the total power input loss $\Sigma\dot{E}_L$ can be written as follows:

$$\Sigma\dot{E}_L = \dot{E}_{ML} + \dot{E}_{FL} + \dot{E}_{CL} + \dot{E}_{VL} + \dot{E}_{OL} \tag{20}$$

where:
- $\dot{E}_{ML}$ = Motor losses
- $\dot{E}_{FL}$ = Friction losses
- $\dot{E}_{CL}$ = Compression losses
- $\dot{E}_{VL}$ = Valve losses
- $\dot{E}_{OL}$ = Oil pumping loss.

Some explanation of these loss mechanisms is in order. The term $\dot{E}_{ML}$ (motor losses) is the electrical power lost in the motor and will mostly consist of I^2R, eddy current, windage, and hysterisis losses. Generally they are taken as constant although there will be some effect of temperature inside the shell. The term $\dot{E}_{FL}$ (friction losses) consists of all the losses due to friction occurring in the transmission process, such as piston and cylinder in case of reciprocating compressor, rotor and cylinder in case of rotary compressor, and the bearings, etc. Obviously the amount of friction losses will vary greatly from one compressor to the other as they are highly dependent on the compressor geometry. $\dot{E}_{CL}$ (compression loss) is the increased area of p-v diagram due to the fact that the compression process may not follow exactly the ideal process (in this case isentropic) primarily due to the heat transfer from and to the cylinder. To compute $\dot{E}_{CL}$, either a detailed heat transfer model of the compressor will have to be made or the value of polytropic coefficient n can be assumed from experience. $\dot{E}_{VL}$ (valve losses) consists of both suction and discharge valve losses. The valve losses, in general, can be further subdivided into various other losses, such as overcompression loss (due to throat area in a rotary vane type), valve restrictions, gas oscillations, valve inertia, etc., depending upon the design of a particular compressor. $\dot{E}_{OL}$ (oil pumping loss) is the power wasted in pumping the lubricating oil. Generally, this is a very small fraction of the total loss and can be neglected except in cases where a separate oil-pump is used for lubrication. Thus, once all these individual energy losses have been analytically determined, Equation (20) will give the total power loss $\Sigma\dot{E}_L$.

Mass Flow Loss

The second unknown that remains to be determined is the total mass flow rate loss. To understand the mass flow loss, we will have to use a slightly different concept than the one we used in energy losses because in this case none of the losses are actually "wasted" to the "sink". The "lost" gas does remain inside the shell and is not lost to any sink outside the shell as in the case of energy losses. Actually, the mechanism of losses is in the form of capacity reduction. Figure 3, showing the mass flow loss model will be helpful in understanding this concept. Thus, we can write:

$$\dot{m}_i = \dot{m}_a + \Sigma\dot{m}_L \tag{21}$$

and

$$\Sigma\dot{m}_L = \dot{m}_{CVL} + \dot{m}_{LL} + \dot{m}_{BFL} + \dot{m}_{OL} + \dot{m}_{SHL} \tag{22}$$

where: $\Sigma\dot{m}_L$ = total mass flow rate loss

$\dot{m}_{CVL}$ = mass flow rate loss due to clearance volume

$\dot{m}_{LL}$ = mass flow rate loss due to leakage

$\dot{m}_{BFL}$ = mass flow rate loss due to backflow

$\dot{m}_{SHL}$ = mass flow rate loss due to suction gas heating

$\dot{m}_{OL}$ = mass flow rate loss due to lubricating oil flow.

The explanation of these loss mechanisms is as follows. The term $\dot{m}_{CVL}$ is as a result of the clearance volume and is directly related to the well-known clearance volumetric efficiency. In those compressor designs which have no clearance volume, this term will be zero. $\dot{m}_{LL}$ is as a result of internal leakage from the compression chamber to either the shell or the suction chamber or both depending upon the design of the compressor. $\dot{m}_{BFL}$ is the result of (1) the discharge gas flowing back into the suction chamber before the discharge valve has had time to close, and (2) the suction gas flowing back to the suction plenum out of the suction chamber before the suction valve has had time to close. This largely depends upon the valve design and the speed of the compressor. $\dot{m}_{SHL}$ is due to the fact that the suction gas gets heated up between the entrance to the shell and the entrance to the suction chamber, thereby reducing its mass density. Since cylinder displacement is fixed by the geometry of the compressor, the reduction in mass density of the gas results in lower value of incoming mass of the gas. $\dot{m}_{OL}$ shows the effect of lubricating oil on mass flow. This would depend upon several factors such as the amount of oil flowing in and out of the cylinder, concentration of the gas in oil and vice versa, state of the oil (whether vapor form or liquid) inside the cylinder, etc. Looking back at Equation (19) we find that all the terms are now known. Quantities V, N, ρ_s, h_d, and h_s are specified by the operating conditions and the geometry of the compressor. $\Sigma\dot{m}_L$ and $\Sigma\dot{E}_L$ have to be determined from the analytical formulas developed above for the various loss mechanisms. Substitution of these in (19) will yield η_p.

EXPERIMENTAL DETERMINATION OF η_p

Based on Equation (14), a simple experimental set-up can be developed to obtain η_p experimentally. We notice from Equation (14) that the two unknowns are $\dot{m}_a$ (actual mass flow rate) and $\dot{E}_{in}$ (actual power input). A simplified schematic of such a possible experimental set-up is shown in Figure 4. It could either be the usual calorimeter test set-up or a modification of it, except that the experimental data that we are interested in now are average inlet pressure, p_1, average inlet temperature, T_1, average outlet pressure, p_2, average wattmeter reading (watts), and the average actual mass flow rate (kg/sec).

From the inlet (suction) temperature and pressure and the outlet (discharge) pressure readings, and using the pressure-enthalpy chart or tables, we can obtain values of the specific enthalpy (enthalpy per unit mass) at suction and discharge conditions for ideal compression; i.e., compression following the isentropic process, and the density (mass per unit volume) of the refrigerant at suction conditions. Thus, h_s, h_d, and ρ_s are known quantities. The average wattmeter reading will give the actual power consumed ($\dot{E}_{in}$). The actual mass flow rate ($\dot{m}_a$) will be obtained from the flowmeter. Substituting all these quantities in Equation (14) with proper units, we get:

$$\eta_p = \frac{(\dot{m}_a)^2 \ (h_d - h_s)}{\left(\frac{VN\rho_s}{60}\right) \dot{E}_{in}} \tag{23}$$

RELATIONSHIP BETWEEN CONVENTIONAL COP, EER, AND η_p

After having discussed the analytical as well as experimental methods of obtaining the efficiency of performance (η_p), it may be worthwhile also to discuss the relationship between η_p, the conventional coefficient of performance (COP), and EER. Referring to the refrigeration cycle on the p-h diagram (Figure 1) we can see that theoretically speaking, 1-2 is the evaporation process at suction pressure p_s, 2-3 is the actual compression process, 2-3' is the ideal compression process (isentropic), 3-4 is the condensation, and 4-1 is the expansion. Then the conventional COP can be written as:

$$\text{COP} = \frac{\text{Heat removed (Btu/hr)}}{\text{Work required (Btu/hr)}} = \frac{(h_2 - h_1)\dot{m}_a}{\dot{E}_{in}} \tag{24}$$

$$\therefore \ \dot{m}_a = \left(\frac{\dot{E}_{in}}{h_2 - h_1}\right)(\text{COP}) \tag{25}$$

where h_1 is the specific enthalpy of the gas at the inlet to the evaporator and h_2 is the specific enthalpy at the outlet from the evaporator. Substituting (25) in (23) we get:

$$\eta_p = \frac{(\dot{m}_a)^2 \ \Delta h_i}{\left(\frac{VN\rho_s}{60}\right) \dot{E}_{in}}$$

$$= \left[\frac{\dot{E}_{in}}{h_2 - h_1} \times \text{COP}\right]^2 \times \frac{\Delta h_i}{\left(\frac{VN\rho_s}{60}\right) \dot{E}_{in}}$$

$$= \left[\frac{60 \ \dot{E}_{in} \ \Delta h_i}{VN\rho_s (h_2 - h_1)^2}\right] (\text{COP})^2$$

or

$$\eta_p = \left[\frac{60 \ \dot{E}_{in} \ \Delta h_i}{VN\rho_s (\Delta h_{evap})^2}\right] (\text{COP})^2 \tag{26}$$

where $\Delta h_{evap} = h_2 - h_1$ = specific heat of evaporation under actual conditions.

It should be noted that since both COP and η_p are dimensionless terms, any system of units can be used for the various quantities in Equation (26) as long as all of them are consistent with each other. However, such is not the case if we use EER (energy efficiency ratio) in place of COP. Relation between EER and η_p can be developed from Equation (26) as follows:

$$\text{EER} = \frac{\text{Heat removed (Btu/hr)}}{\text{Work required (watts)}}$$

$$= \frac{\text{Heat removed (Btu/hr)}}{\text{Work required (Btu/hr)}} \times 3.413 \ \frac{\text{Btu/hr}}{\text{watts}}$$

$$= 3.413 \times \text{COP (Btu/watt hr)}$$

or

$$\text{COP} = 0.293 \times \text{EER (numerically)} \tag{27}$$

Substituting Equation (27) in (26) we get:

$$\eta_p = \left[\frac{5.15 \ \dot{E}_{in} \ \Delta h_i}{VN\rho_s \ (\Delta h_{evap})^2}\right] (\text{EER})^2 \tag{28}$$

It must be kept in mind that (28) is valid only when EER is given in (Btu/watt hr), although all other quantities can be used in any system of units as long as they are consistent with each other.

CLOSURE

A variety of definitions are being used today for the performance of a compressor, none of which give a clear picture of the various losses, occurring in the compressor, that affect its performance. A new term "efficiency of performance (η_p)" was developed which does away with this problem. Various energy and mass flow losses that affect the performance of a compressor were identified and a formula for η_p was developed that is based on these losses. This makes it easier to understand as to how the various design parameters can affect the performance and hence care can be taken at the design stage to improve the performance of the final product. This approach also opens a way for ultimate optimization of a particular compressor design. The criterion for optimization could either be the η_p if overall performance is considered more important, or η_e if only energy consumption is more important, or even η_{ma} if only the mass flow or the capacity is of prime consideration. In any case, the approach suggested in this paper opens a new and different way of looking at the compressor performance and should prove to be helpful in ultimately studying, improving, and optimizing the various design concepts.

NOMENCLATURE

COP	Coefficient of performance (dimensionless)
$\dot{E}_{in}$	Power input (Nm/sec)
$\Sigma\dot{E}_L$	Total power loss (Nm/sec)
$\dot{E}_{ML}$	Motor losses (Nm/sec)
$\dot{E}_{CL}$	Compression loss (Nm/sec)
$\dot{E}_{FL}$	Friction loss (Nm/sec)
$\dot{E}_{VL}$	Valve losses (Nm/sec)
$\dot{E}_{OL}$	Oil pumping loss (Nm/sec)
EER	Energy efficiency ratio (Btu/watt-hr)
h_1	Specific enthalpy at the inlet to evaporator (Nm/kg)
h_2	Specific enthalpy at the outlet of evaporator (Nm/kg)
h_s	Specific enthalpy at suction conditions (Nm/kg)
h_d	Specific enthalpy at discharge point following isentropic compression (Nm/kg)
Δh_i	Ideal specific enthalpy increase during compression process (Nm/kg)
Δh_a	Actual specific enthalpy increase during compression process (Nm/kg)
Δh_{evap}	Specific enthalpy increase during evaporation process (Nm/kg)
m	Mass (kg)
$\dot{m}$	Mass flow rate (kg/sec)
$\dot{m}_i$	Ideal mass flow rate (kg/sec)
$\dot{m}_a$	Actual mass flow rate (kg/sec)
$\Sigma\dot{m}_L$	Total loss in mass flow rate (kg/sec)
$\dot{m}_{CVL}$	Loss in mass flow rate due to clearance volume effect (kg/sec)
$\dot{m}_{LL}$	Loss in mass flow rate due to leakage (kg/sec)
$\dot{m}_{BFL}$	Loss in mass flow rate due to back flow (kg/sec)
$\dot{m}_{OL}$	Loss in mass flow rate due to oil flow (kg/sec)
$\dot{m}_{SHL}$	Loss in mass flow rate due to suction gas heating (kg/sec)
N	RPM of the motor
P_1	Suction pressure (N/m^2)
P_2	Discharge pressure (N/m^2)
T_1	Evaporator temperature (°K)
T_2	Condenser temperature (°K)
V	Cylinder displacement per shaft rotation (m^3)
$\dot{W}$	Shaft work rate (watts)
η_c	Adiabatic compressor efficiency
η_e	Energy efficiency
η_m	Mechanical efficiency
η_{motor}	Efficiency of the electric motor
η_{ma}	Mass flow efficiency
η_p	Efficiency of performance
η_v	Clearance volumetric efficiency
π	Performance ratio $(\frac{kg/sec}{Nm/kg})$
π_a	Actual performance ratio $(\frac{kg/sec}{Nm/kg})$
π_i	Ideal performance ratio $(\frac{kg/sec}{Nm/kg})$
ρ_s	Mass density at suction conditions (kg/m^3)

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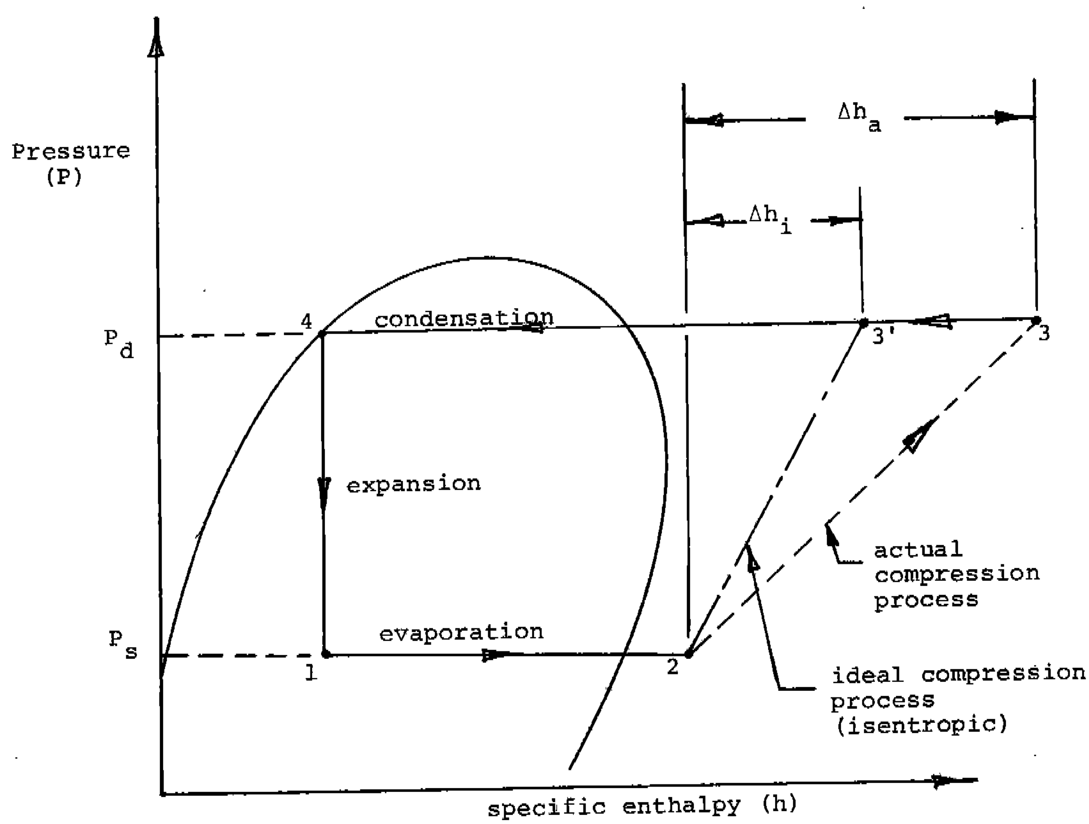

Fig. 1: Pressure-enthalpy diagram of a refrigeration cycle.

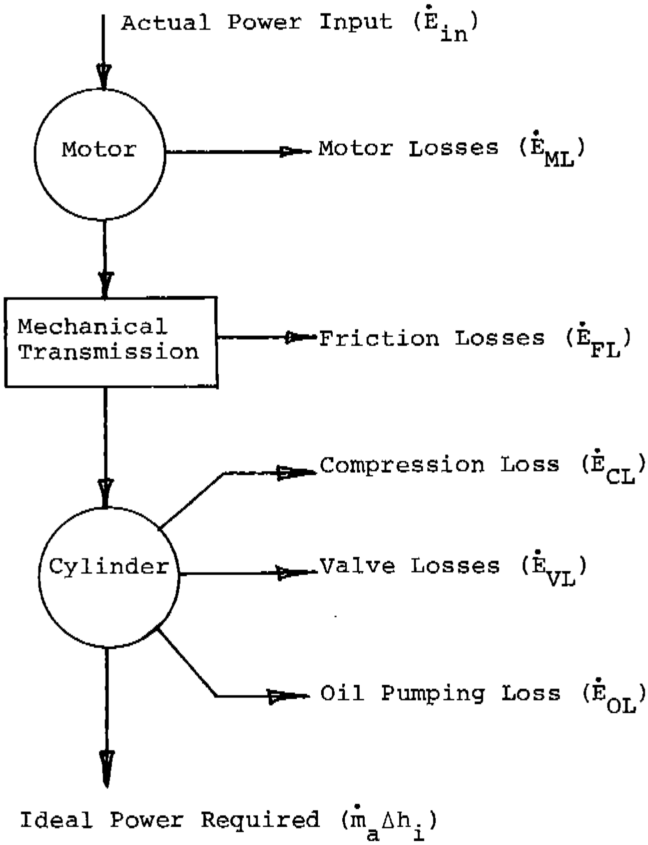

Fig. 2: Energy-balance of a compressor

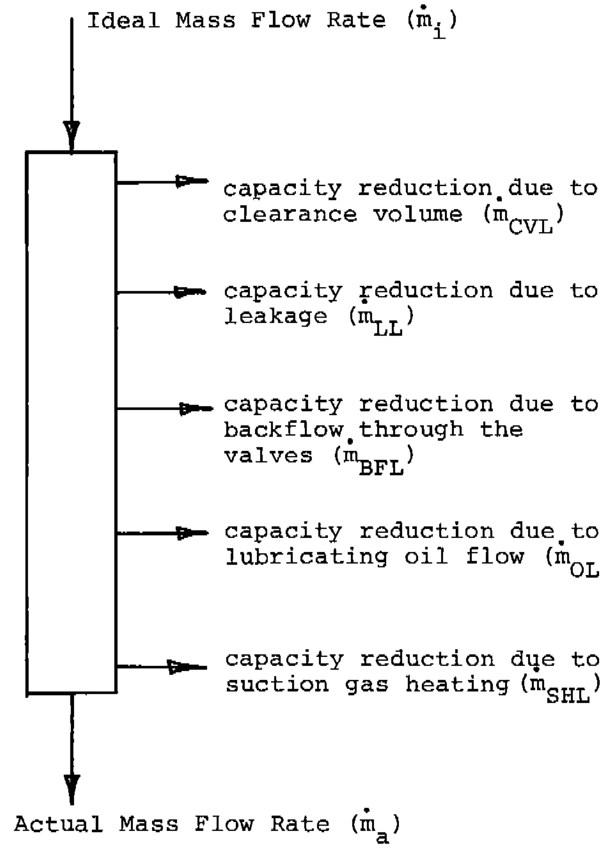

Fig. 3: Mass flow loss model of a compressor

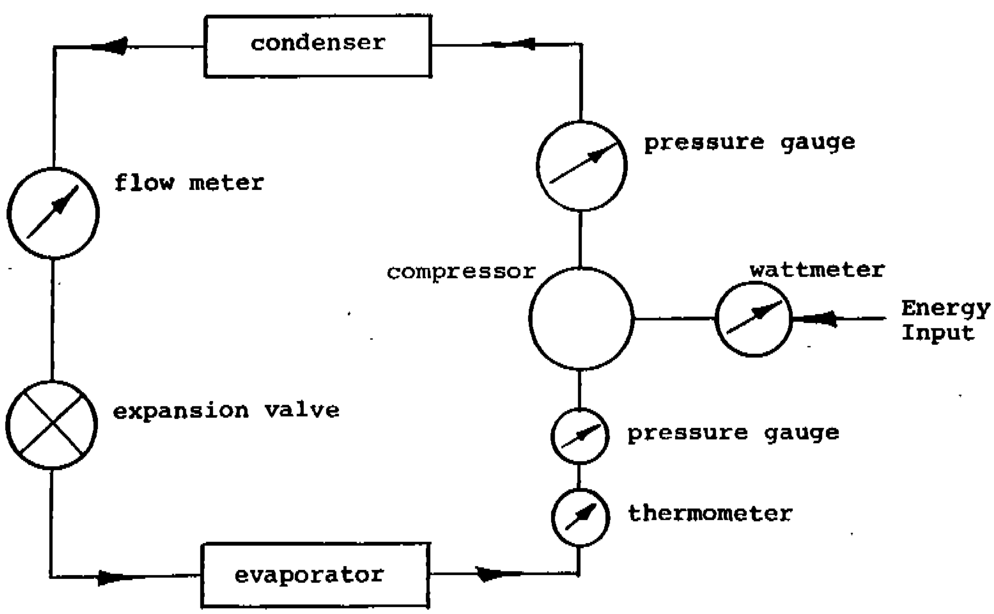

Fig. 4: Experimental Test Set-Up for η_p