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ENERGY SAVINGS WITH OPEN CYCLE HEAT PUMPS

(MECHANICAL VAPOUR COMPRESSION)

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"Industrial Heat Exchange and

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1. SUMMARY

Mechanical steam recovery, steam compression and steam recompression are viable means of reducing operating costs in paper mills, evaporation processes and cogeneration applications as well as others.
This paper discusses how much cost reduction can be achieved and it also shows how the compressor industry has advanced steam compression over the last years.

Today's compressor technology in conjunction with the materials available makes it possible to compress any kind of steam to almost any pressure. The limits are set by the physical limits of the materials, only.

An availability study performed by Leipowitz U.S.A. for 270 turbo-compressors with 3.2 Mio operating hours, showed a mechanical availability of 99.9%.

Single and multistage centrifugal compressors are capable of achieving an enthalpy difference of 180.6 kJ/kg per stage and handling flows of up to 200.000 m3/hr.

The special steam screw compressor product line achieves pressure ratio capabilities of up to (9) !!! per stage with flow ranges from 500 up to 20,000 m3/hr.

This paper will also discuss the "STATE OF THE ART" of mechanical vapor compression using both centrifugal and screw compressors. Particular reference to compressor design details and operating experience at specific installations will be given.

History has shown that using a mechanical compressor for steam compression does not just reduce the operating costs by a substantial amount of money, but provides a payback as short as a couple of months in certain cases. This is of course the extreme. A normal payback is between two and three years.

II. INTRODUCTION

Mechanical steam or vapor compression or recompression has a long history and it not something just recently discovered.

In 1960 the first steam compressor was built for a Chemical Plant in Germany.

The performance was more than modest and the engineers of that time were faced with problems nobody even talks about today.

But even this compressor performed better than a steam vessel from today. It was able to provide lower operating costs through mechanical steam recompression than would be experienced using water and generating steam.

In the early seventies more and more evaporation equipment manufacturers approached compressor manufacturers to develop and build a compressor for vapor recompression for their evaporation processes.

The idea was simple; use the boil-off vapor and recompress it to a pressure which corresponds to a temperature used in the heat exchangers. This way they could eliminate two to three stages of thermal compression in the process. The reduction in both the investment cost and the operating cost looked very attractive.

When using a compressor, heat balance never comes out equal. The compressor, due to its efficiency, gives superheated steam on the discharge side. Sometimes water has to be injected to get saturated steam. This small problem was solved simply by generating warm water to the equivalent of the excess heat.

These compressors were working with a pressure ratio of 1.6 to 1.7 and were mostly used in dairy evaporation processes, running with a tipspeed of around 380 m/s (1240 fps). From today's point of view they were built on a conservative basis.

Many chemical and process engineers have been waiting for mechanical vapor recompression to prove itself possible and economical. It has done so, with the result that more and more industries have started looking into the economies it offers.

Materials developed, mostly in the aircraft and spaceship industry, made it possible to increase the tipspeed of a centrifugal compressor to 550 m/s (1800 fps). These newer materials also offer excellent corrosion and erosion resitance.

Applications

MVR systems are used in many kind of industries such as:

- Food industry e.g. dairies, sugar, glucose, fruit concentration, breweries etc.
- Chemical industry
- Pulp and Paper
- All other evaporation processes.

Evaporators

Figure 1 shows a basic evaporator, so called single effect. Most of the heat disappears at the top of the vessel, which makes this system very unefficient.
Multi-effect systems are used to make the system more efficient. By doing so the waste heat of the first is fed to the steam chest of the following evaporator.

MVC

With MVC a compressor is brought in the system. The compressor compresses the heat up to a higher pressure and subsequently, to a higher temperature (Fig. 2).
Table 1 shows a cost comparison of a single effect, a multi-effect (3 evaporators) and a single effect with vapor compression.

III. CENTRIFUGAL COMPRESSORS

III. A. GENERAL

Centrifugal compressors are dynamic compressors, i.e. their compression principle is based on taking the gas (steam) at a certain velocity c1 into the compressor wheel. Due to the rotation of the wheel the gas is accelerated and discharged at a a relatively high velocity c2. At the discharge of the wheel, the pressure changes to approx. 55 % of the total pressure ratio. The rest is contained in the gas in the form of kinetic energy and must be converted.

Diffusers are being used to decrease this velocity c2 and convert the kinetic energy into pressure, simply following the Bernoulli equations.

Figure 3. shows a typical centrifugal compressor package.

Such a package typically consists of:
- Inlet guide vanes
- The compressor stage, with wheel, shaft and seal
- The speed increaser with coupling and guard
- The base frame with lube oil system and main driver
- A local control panel including failsafe system and all necessary instruments.

A packaged centrifugal compressor requires very little space. Installation and maintenance is easy and inexpensive.

These are, besides many other benefits, good reasons to choose a packaged compressor.

III. B. SIZING CALCULATIONS

Learning to size a centrifugal compressor is much easier than learning how a centrifugal compressor really works. Also most users are more interested how much power is needed to bring steam or vapor from one pressure level to another. Below is shown how to calculate this needed power.

The user must pay for the power supplied to the main driver's terminals; i.e. all losses have to be paid by the user. This power at the terminals is where the evaluation occurs, not on some fancy efficiency or power determination used by some manufacturers. It is very common for compressor manufacturers to give the power draw at the input shaft of the compressor because up to this point they have control over their equipment and they know what efficiency the compressor has, including the the mechanical losses of the gears. Not many like to guarantee the motor efficiency, because this is something they do not build and is not really their field.

However, the user still has to pay the bill for the power at the motor terminals.

The power at the motor terminals is determined as:

$$P_{term} = \frac{\dot{m}\,[kg/sec] \cdot \Delta h\,[kJ/kg]}{\eta_{mot} \cdot \eta_{compr.}} + P_{gears} \quad kW \;;\; kW$$

Equation 1.

P term = Power at motor terminals
$\dot{m}$ = Massflow of steam through the compressor
Δh = Enthalpy difference inlet flange to discharge flange
η_{mot} = Motor efficiency
$\eta_{compr.}$ = Compressor efficiency determined from inlet to discharge flange
P gears = Mechanical losses not included in the compressor efficiency.

The massflow is in all cases given by the users. They know how much steam or vapor has to be brought to the higher pressure.

The change in the enthalpy is basically given by the change in pressure, or as many users do it by the change in the saturation temperature.

It can be taken out of the steam tables or simply calculated by using:

$$\Delta h = Z1 \cdot \left[k/(k-1)\right] \cdot R \cdot T1 \cdot \left[(p2/p1)^{\frac{k-1}{k}} - 1\right] \text{ ; kJ/kg steam}$$

Equation 2.

Δh = Enthalpy difference
k = Specific heat ratio cp/cv
R = Gas constant for steam in kJ/kg
T1 = Suction temperature in K
p2 = Discharge pressure in any
p1 = Suction pressure in any
Z1 = Compressibility factor

Graph 1. gives the gas constant of steam already multiplied by the compressibility factor versus inlet pressure. Graph 2. gives the k-value versus inlet pressure.

With these two equations the power of a centrifugal compressor can easily be calculated if the efficiencies for the compressor and the main driver are known.

Motor efficiencies can run anywhere from 90 up to 98 %, depending on the make of motor and the power level.

The compression efficiency of today's centrifugal compressors should not be less than 78 to 80 %. With increasing wheel diameter the efficiency can go up to 82 and 84 %.

To find the wheel diameter, one first has to find the optimum speed the particular compressor should run at:

Equation 3.

n = Optimal compressor speed
Δh = Enthalpy difference kJ/kg
V = Actual inlet volume m3/hr
C = Geometry constant, 484 for ATLAS COPCO COMPTEC

With this speed and the required tipspeed, the wheel diameter is determined as expressed in equation 4. as below.

It should be understood that this sizing and power calculation is only an estimate and a detailed design must be done for each particular application.

$$D2 = \frac{u2 \cdot 60000}{n} \; ; \; [mm]$$

Equation 4.

D2 = Wheel diameter
u2 = Required tipspeed in m/sec
n = Optimal speed in RPM

The required tipspeed is calculated out of the enthalpy difference and the so called pressure coefficient.
Basically there are only two different wheel types used for steam compression, the 90 degree or radial wheel and the 70 degree backward leaned wheel.
The pressure coefficient for a radial wheel as an average is 1.32 and for a 70 degree wheel 1.2. The difference between these two wheels is that the radial wheel needs less tipspeed to achieve the required enthalpy difference but has a very flat compressor curve which can even drop down toward the surge

limit. A 70 degree wheel has a steady rise to surge but needs a slightly higher tipspeed.

The tipspeed is:

$$u2 = \sqrt{\frac{2 \cdot \Delta h}{\psi}} \; ; \; [m/sec]$$

Equation 5.

$u2$ = Tipspeed

Δh = Enthalpy difference in kJ/kg

ψ = Pressure coefficient

The last term out of equation 1. to be considered is the mechanical or gear losses.
ATLAS COPCO uses a modified AGMA Q13 speed increasing gear and the losses in the gearbox design point are 3 % of the design power.
Since most gears are used over a certain range of powers, a general number of 3 % cannot be used. As an average for gear losses, 5 to 6 % of the power found in equation 1. is a good number and will represent for most cases the mechanical losses; i.e. bearing losses, gear losses and mechanical driven lube oil pump.

III. C. AERO COMPONENTS

Aero components are considered those parts which directly achieve the enthalpy difference in the compressor. In other words, from the inlet flange to the discharge flange (the inlet nozzle or shroud, the compressor wheel, the parallel diffuser, the diffuser vanes and the scroll or volute casings.)

Inlet guide vanes when required are bolted to the inlet flange and only partially effect compression. Normally they are not counted as aero components.

Figure 4 displays the aero components of an assembled compressor stage.

The compressor inlet nozzle performs the task of accelerating the steam prior to its entering the compressor wheel.

In most cases a smooth Venturi type of nozzle is used to do this. A straight cone nozzle is seldom used because it does not provide a smooth transition between the two diameters, (inlet flange and wheel entrance).
The inlet nozzle is often used for flow measurement because of its Venturi pipe effect. This eliminates an orifice or Anubar flow measurement device.

After the inlet nozzle the compressor wheel is next. It is the most important aero component. It generates approx. 55 % of the necessary enthalpy différence directly and puts the rest of it in the gas in the form of kinetic energy.

The shape and design of the wheel determines the compressor's overall plerformance. It influences the size of the bearings, the running speed of the compressor, the tipspeed pontential with certain materials and many others.

Stainless steel is the most common compressor wheel material in steam ompressors, but certain applications can require Inconel or Titanium wheels. ATLAS COPCO uses precision cast 17-4 PH stainless steel wheels as a standard material. Optional, and used in many machines are materials such as Inconel 600 and Inconel 718, Titanium Grade V, Monel, Hastelloy and AISI 304 and 316.
All these materials require careful review and calculation as some of them cannot be cast but must be milled out of a forging.

Many different types of compressor wheels are used in steam or vapor compressors.

- The 90 degree or radial wheel, or backwards leaning impellers

The wheel anle is determined from the horizontal line where the blade begins to the backward lean of the wheel exit angle.

Higher backward leaned wheels are not useful because they have smaller pressure coefficients and require higher tipspeeds (see equation 5.).

Basically it does not matter whether a 90 or backwards leaning wheels are chosen. The difference in tipspeed is minor and only at the very limit of the wheel's capability can it make a difference.

When changing the wheel material, the possible tipspeed will change, too. As an approximate, equation 6. gives the change in tipspeed when changing the wheel material.

$$u2_{new} = u2_{old} \sqrt{\frac{\dfrac{\sigma 0.2_{old}}{\gamma_{old}}}{\dfrac{\sigma 0.2_{new}}{\gamma_{new}}}}$$

Equation 6.

σ 0.2 = Yield strength

γ = Density of material

This equation requires that one tipspeed for a certain material be known.

For the standard ATLAS COPCO material (17-4 PH S.S.) the maximum tipspeed is 430 m/s.

Next, right after the wheel, are the parallel diffuser and diffuser vanes. Their function is to decelerate the steam from the wheel exit velocity to an acceptable value before entering the last part, the volute or scroll casing. The width of the

parallel diffuser is determined by the width of the wheel blade. The angle setting of the diffuser vanes is determined by the wheel exit velocity and the acceptable velocity at the volute entrance. The distance between the wheel exit and the diffuser vane entrance is determined by a maximum acceptable Mach Number at the leading edge of the diffuser vanes.
Many compressor manufacturers use a fixed scroll design; i.e. for each wheel, no matter what the actual conditions are, they have only one compressor casing. This eliminates having a large number of casings, with some only used once. With this fixed casing design the length of the parallel diffuser is limited and diffuser vanes are needed to decelerate the steam or gas to the volute entrance velocity. The volute then is only used to achieve a very small amount of pressure rise and so functions as a collection chamber only. This does not influence the compressor performance and provides advantages to both the customer and the compressor manufacturer.

The last part, (the volute with a discharge flange for the customer's pipe connection) is the compressor casing, the visible part of the compressor. Most of the time it must be stainless steel and it has become very popular to fabricate it rather than cast it.
Fabricating the casing and maintaining a fixed design helps provide the customer with a low cost compressor that offers the most flexibility.

III. D. COMPRESSOR ROTOR

The compressor shaft and wheel together form the compressor rotor. Besides the bearings, the rotor is the most sensitive part of the compressor and some special attention has to be paid to this part. The wheel and shaft individually balanced in two planes and then both assembled and balanced in two planes does not provide that the compressor will later run smoothly and properly. Therefore, ATLAS COPCO performs a rotor analysis on each rotor taking into account the wheel's geometry, the wheel's weight, the bearings' geometry and span as well as the shaft geometry including the pinion and gear force. This procedure provides our customers with the highest security possible.

Figure 5 shows such an assembled rotor.

As a standard, tilting pad journal bearings and tapered land thrust bearings support the compressor rotor. These bearings are carefully calculated and equipped with the necessary oil orifices to provide the bearings with only as much oil as need to keep the losses to a minimum.

the low speed shaft has combined journal/thrust sleeve bearings on either side to ensure the input shaft is kept in its proper position.

All bearings are horizontally split for easy disassembly and inspection.

III. E. SEALS

The seals can be considered the part that divides the aero components from the mechanical components. They ensure that no steam enters the gearbox and contaminates the oil. The most common and probably the best seals for steam applications are buffered floating carbon ring and buffered labyrinth seals, as shown in figure 6. and 7.

III. F. GEARBOX

Equation 3. indicates that the compressor wheels require a high speed. However, normal electric motors run with 3.600 or 3.000 RPM or less. A speed increaser or a gearbox is therefore needed to bring this motor speed to wheel speed.
ATLAS COPCO uses a high quality AGMA 421, Q13 gearbox to do this.
Hardened and ground, quality 13 gears provide individual interchangeability of the gears avoiding the cost and problems of changing matching gear sets.
Helical teeth assure low noise operation and counteract the thrust of the wheel.
Integral to the gearbox is the drive mechanism for the main, mechanical driven, lube oil pump, which is flanged to the gearbox.

This pump, as well as the auxiliary oil pump, is a screw type, low noise, positive displacement pump and provides enough oil supply for the compressor to coast down.

What has been described so far are the vital parts of the compressor. These parts should be of the highest quality and the target of careful design and calculation for long and satisfactory performance.

III. G. LUBE OIL SYSTEM

No compressor could run without a lube oil system. Many different opinions exist on how to build the best or right lube oil and failsafe system. In this paper we cannot refer to all of them so we will give some guidance. Basically a lube oil system should consist of the main lube oil pump, driven by the input shaft; an auxiliary oil pump, necessary to lubricate the bearings prior to start up and able to be switched off when the main pump supplies enough oil to the bearings; an oil cooler; a dual oil filter after the oil cooler; check valves before the pumps; an adjustable pressure relief valve; a drain line on the oil reservoir and a drain line in the oil system.
Figure 7. shows the standard lube oil system P & I-Diagram.

Besides the above mentioned parts, the following instruments should be on the lube oil system to protect the compressor from damage when a part fails and to adjust the oil pressure and the oil flow.
A pressure gauge before and after the oil filter, to see whether the active filter is dirty and must be changed; an oil temperature gauge to ensure that the oil has the correct temperature prior to start up; a level gauge to check the oil level in the reservoir; there should be an oil pressure switch with two set points, one for low alarm and auxiliary pump switch on and the second for shutting the compressor off when the oil pressure drops further down; a temperature switch should be on the lube oil system with three set points, one to protect a false start up on a too low temperature, one high alarm when