MEASURING THE QUANTITY OF INCONDENSABLE GAS REMOVED BY A LIQUID RING VACUUM PUMP

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Abstract

Vessels operating under vacuum, such as vacuum pans and final effect evaporators, require a vacuum pump to remove incondensable gasses that accumulate within the vessel. The vacuum pumps need to be sized to remove incondensable gasses from four possible sources: those that enter with the feed into the vessel, direct leaks from the atmosphere into the vessel, dissolved and entrained gas that is released from injection water used in direct contact condensers, and gas that is contained in steam vented into the vessel body.

It is often necessary to discover why a process vessel is unable to maintain the design vacuum. If the condenser is capable of adequately cooling the gas stream entering the vacuum pump, an inability to maintain the design vacuum within the vessel is an indicator of either insufficient pump capacity or excessive quantity of incondensable gas. To diagnose the cause of the problem, pump performance can be checked by measuring the performance of the vacuum pump, off-line, using a simple test rig, whilst the possibility of excessive quantities of gas are normally investigated only by using a standard vacuum test on an empty vessel to estimate the quantity of the air leakage into the vessel.

As an alternative to off-line pump testing and the conventional vacuum test for leaks, a measurement of the quantity of incondensable gas being discharged from the vacuum pump during normal operation provides a very useful diagnostic test. The test provides a direct measure of the load (of incondensable gas) on the pump whilst also providing a single point on the pump curve and thus can give an instantaneous indication of the reason for the inability to maintain the design vacuum. A simple method of performing this measurement is described.

Keywords: vacuum pump, vessel, leak, air, incondensable gas

Introduction

To concentrate sugar solutions by boiling under a vacuum (as is done in pans and final effect evaporators) conventional practice is to maintain the desired vacuum conditions using a combination of a direct contact, counter-current, condenser and a liquid ring vacuum pump. A previous paper (Love, 2005) has explored how these two pieces of equipment work together under the direction of a control system to maintain a steady pressure within the process vessel as the evaporation takes place. The paper also describes some strange and, in the first instance, counterintuitive behaviour as a result of the interaction between the condenser, the vacuum pump and the control system. (Appendix 1 outlines an error in the calculations described in the paper that affects the quantitative results presented, but not the qualitative behaviour that was the thrust of the paper).

If one focuses on the need to remove incondensable gasses continuously from the boiling vessel, underlying the complexity of the behaviour of the whole system are two important issues:

- The quantity of incondensable gas that needs to be removed.
- The capacity of the vacuum pump to remove the incondensable gas.

Clearly, if the quantity of incondensable gasses is too large, or if the capacity of the vacuum pump is too small, it will not be possible to maintain the desired vacuum (assuming that the condenser is capable of achieving its duty to condense the vapour). Investigating this statement in some detail, exactly what is meant by 'too large' and 'too small' will shed light on how best to design and trouble-shoot the incondensable aspects of a vacuum system.

There is a generally accepted procedure in the sugar industry for estimating the quantity of incondensable gasses to be removed from a vacuum pan, based on its size and expected operating conditions, that is succinctly summarised by Rein (2007). Gomez (1991), when discussing air leakage into vacuum systems in general, terms this approach an *empirical* method. Gomez (1991) contrasts this approach to *operational* methods, where the air leakage is directly measured rather than estimated, and references a Heat Exchange Institute publication (Anon, 1988) on the use of the 'rate of rise' method of measuring air leakage rates. This is an interesting distinction and the rate of rise method is used in the sugar industry both as an *operational* method (see Appendix 2) and, by specifying a maximum rate of rise as an *empirical* method.

A local supplier of vacuum equipment has queried local specifications for the quantity of incondensable gas to be removed from pans and suggested that they are large compared with what would be used in other industries. Prompted by this comment, this paper looks critically at the *empirical* methods used in the sugar industry for estimating quantities of incondensable gas in the light of more general information relating to the chemical process industries.

Estimating the quantity of incondensable gasses to be removed from a pan

To estimate the quantity of incondensable gas that needs to be removed from a pan boiling under vacuum it would be ideal to consider the combination of the following mechanisms:

1. Air that is released from the injection water being used in the condenser to condense the vapour.
2. Plus air that leaks into the vessel from the atmosphere.
3. Plus air that enters the vessel with the feed to the vessel.
4. Plus incondensable gas that may be vented into the body of the vessel from the calandria of the vessel.
5. Less any incondensable gas that is entrained in the tailpipe water leaving the condenser.

Rein (2007) provides details on how to estimate the quantities of gas associated with the first three of these mechanisms and provides a useful simple formula to facilitate the calculation.

The fourth mechanism, not mentioned by Rein (2007), will not apply in those instances where vapour containing incondensable gasses from the calandria is vented to the atmosphere. There are, however, a number of possible reasons for venting the calandria into the vessel:

- Convenience – there is often a mess and a possible safety hazard associated with venting the vapour and gasses to atmosphere, both of which are easily eliminated by venting back into the vessel body.
- Sub-atmospheric conditions in the calandria – in continuous pans the calandria will often operate under sub-atmospheric conditions. In this situation the calandria needs to be vented to a lower vacuum – most conveniently provided by the vessel body.
- Jigger steam – as an energy saving strategy the vented vapour containing incondensable gases from the calandria can be used as jigger steam to improve circulation within the pan (e.g. Rackemann and Broadfoot, 2007).

The fifth mechanism, the quantity of incondensable gas entrained in the tailpipe water leaving the condenser, is normally not taken into account in estimating the quantity of gas to be removed and, in any event, neglecting it will result in a conservative estimate when sizing the vacuum pump. It is difficult to know what the relative magnitude of this fifth mechanism might be. In the extreme case of jet (co-current) condensers, all the incondensable gasses are entrained in the tailpipe water, obviating the need for a specific device (vacuum pump or steam ejector) for removing incondensable gasses. On the other hand, modern counter-current condensers often have tailpipe diameters sized for self-venting flow to minimise entrainment of gas and eliminate the possibility of surging flow of water in the tailpipe. Which leaves the enigma of what the magnitude of the entrained gas might be.

It is instructive to look in detail at the relative magnitude of each of the first four mechanisms influencing the quantity of incondensable gasses to be removed from a vacuum vessel, omitting the fifth mechanism because of the lack of available information. To provide comparative quantitative estimates, the approach used by Rein (2007) of considering an example of a specific batch pan has been followed. The details of the batch pan considered are as follows:

Duty	: A massecuite strike boilings
Massecuite volume at strike	: 85 m^3
Total volume of pan and condenser (massecuite and vapour)	: 170 m^3
Heating surface area	: 510 m^2
Brix of syrup feed to pan	: 66%
Maximum specific evaporation rate	: 55 kg/(h m^2)
Maximum evaporation rate	: 28.05 ton/h
Jigger Steam	: No jigger steam used
(Note that if jigger steam is added, it must be added to the condensing load)	
Operating pressure	:13 kPa abs
Saturated vapour temperature at operating pressure	:51.0ºC
Tailpipe approach temperature of condenser	: 3ºC
Tailpipe water temperature	: 48ºC
Injection water temperature	: 34ºC
Change in water temperature across condenser	: 14 C
Ratio of Injection water to vapour condensed*	: 40.7
Maximum Injection water flow	: 1142 ton/h

*Rein (2007) suggests the following simplified relationship for determining the ratio of injection water used to vapour condensed:

$$\frac{mass\ of\ injection\ water}{mass\ of\ vapour} = \frac{570}{change\ in\ water\ temperature\ across\ condenser}$$

Using this specific example it is possible to estimate the contribution to the total incondensable gas load made by of each of the first four mechanisms.

Mechanism 1 – Air from injection water:
A conservative estimate of quantity of incondensable gas contributed by the injection water is to assume that all of the air estimated to be in the injection water is released from the water under the vacuum conditions within the condenser. The estimate is simply then the flow of injection water multiplied by the concentration of air in the water.

Hugot (1986) quotes the following possible values for the concentration of air in injection water depending on the source of the water:

Calm rivers or ponds	: 20-30 mg/kg
Mountain streams or cascades	: 50-150 mg/kg
Sea water	: 10-20 mg/kg
Cooling pond or cooling tower	: 30-40 mg/kg

Based on these figures, Rein (2007) suggests using a figure of 35mg/kg for water from cooling towers or cooling ponds.

It is possible to investigate the solubility of air in water using Henry's law and published values of the proportionality constant for the air water system (Perry and Green, 1997).

Henry's law proposes that the solubility of gasses in liquids (at low concentrations) is directly proportional to the partial pressure of the gas, i.e:

$$x = \frac{p}{H}$$

where:
x is the mole fraction of the gas in the liquid phase (moles of gas per mole of solution)
p is the partial pressure of the gas
H is the proportionality constant (Henry's law constant).

For the air/water system, the mole fraction x can be converted to a mass fraction w by using the known molecular masses of air (28.964) and water (18.061) as follows:

$$w = \frac{x \cdot 28.964}{(1-x) \cdot 18.016 + x \cdot 28.964}$$

Using tabulated values of H (in units of atmospheres) from Perry and Green (1997) and assuming an atmospheric pressure of one atmosphere, it is possible to calculate the following values for the solubility of air in water.

Temperature	ºC	20	25	30	35
H	Atmosphere	66400	72000	77100	82300
x	mole fraction	1.51E-05	1.39E-05	1.30E-05	1.22E-05
w	mass fraction	2.42E-05	2.23E-05	2.09E-05	1.95E-05
w	mg/kg	24.2	22.3	20.9	19.5

Comparing the concentration of air in water estimated from Henry's law with the values quoted by Hugot (1986), and used as the basis for the recommendation by Rein (2007) it appears that the 'sugar engineering' numbers are making a practical allowance for dispersed air bubbles that are entrained in the water that enters the injection water pumps. Specifically the term 'mountain streams and cascades' implies cool water that has undergone multiple stages of air water contact thereby raising the concentration of the air in the water to a level that cannot be explained by the true solubility (as estimated using Henry's law). It also means that inadvertent aeration (e.g. from a recycle line used for pressure relief) in the sump feeding the injection water pumps might raise the concentration of air in injection water substantially. Although the recommendation of a design value of 35mg/kg (Rein, 2007) seems most appropriate, an extreme range from 20-100 mg/kg appears possible.

Based on this range of possible values for the concentration of air in water, the quantity of incondensable gas attributable to this mechanism for the example pan will be:

Concentration of air (mg/kg)	Injection water flow (ton/h)	Flow of air (kg/h)
20	1142	22.8
35	1142	40.0
100	1142	114.2

Mechanism 2 - Leaks:
As discussed in the introduction, the sugar industry uses a 'rate of rise' test both to measure air leaks into pans and to characterise acceptable leak rates. The details of the test and the calculations necessary to convert the test results into air flow are given in Appendix 2. Rein (2007) characterises acceptable and maximum leakage rates in terms of the test as follows:

For new pans : less than 10 kPa in 60 minutes
For old pans : less than 10 kPa in 30 minutes
Maximum for a pan : 15 kPa in 30 minutes.

Using the relationships in Appendix 2, these numbers can be converted into incondensable gas flow rates as follows:

New pan : $0.0117 \cdot 170 \cdot 10 / 60 = 0.332$ kg/min = 19.9 kg/h
Old pan : $0.0117 \cdot 170 \cdot 10 / 30 = 0.663$ kg/min = 39.8 kg/h
Maximum : $0.0117 \cdot 170 \cdot 15 / 30 = 0.995$ kg/min = 59.7 kg/h.

It is interesting to compare these leakage rates with estimates that are not specific to the sugar industry.

Gomez (1991) provides equations to replicate the numbers recommended by the Heat Exchange Institute and published by them as graphs (Anon, 1988). These equations estimate the maximum air leakage rate to be tolerated on a vacuum vessel and can be calculated from the volume of the vessel according to the equation:

$$MAL = A \cdot V^B$$

where:
MAL is the maximum air leakage rate in kg/h
V is the volume of the system in m^3
A and *B* are dimensionless parameters.

To provide more stringent tightness requirements for systems operating at higher vacuums, the values of A and B are selected according to the range that the operating pressure of the vessel falls into, as follows:

Pressure range (kPa abs)		A	B
Maximum	Minimum		
101.3	12.0	0.9430	0.6630
11.9	2.8	0.6966	0.6617
2.7	0.4	0.4784	0.6579
0.4	0.1	0.2415	0.6568
0.1	0.0	0.1220	0.6639

Vacuum pans operate close to the boundary between the two zones at the top of the table and thus it is instructive to calculate the maximum air leakage rates for both ranges (termed for convenience 'low vacuum' and 'high vacuum'), as follows:

Low vacuum $MAL = 0.9430 \cdot 170^{0.6630} = 28.4$ kg/h
High vacuum $MAL = 0.6966 \cdot 170^{0.6617} = 20.8$ kg/h.

Ryans and Croll (1981) provide information on detailed calculations for estimating leakage rates based on an allowance for leakage through welds (cracks and porosity) combined with the summation of leaks from all the various fittings, valves and seals on the vessel, based on appropriate standard leakage rates applied to each one of these. This level of detailed calculation is probably inappropriate for the undemanding levels of vacuum in vacuum pans. As a simplification of their detailed calculation procedure, Ryans and Croll (1981) recommend simply multiplying the allowance for leakage through welds by a factor of two to get a preliminary estimate of the total leakage rate.

The equation for the maximum leakage rate allowance for welds given by Ryans and Croll (1981) is:

$$WL = A \cdot P^B \cdot V^C$$

where:
WL is the weld leakage (kg/h)
P is the absolute pressure in the pan (kPa abs)
A, *B* and *C* are dimensionless parameters.

And thus, applying the factor of two to the weld leakage rate:

$$MAL = 2 \cdot WL$$

Even though the pressure is included as a variable in the equation, the parameters *A*, *B* and *C* are dependent on the range that the operating pressure falls into:

Pressure range (kPa abs)		*A*	*B*	*C*
Maximum	Minimum			
101.3	13.3	0.4089	0.0000	0.6000
13.3	1.3	0.2085	0.2600	0.6000
1.3	0.1	0.0234	0.3400	0.6000

(The units used by Ryans and Croll (1981) are a mixture of Imperial and metric and have been converted to SI units for presentation here.)

Again, vacuum pans operate close to the boundary between the two zones at the top of the table and thus it is instructive to calculate the maximum air leakage rates for both ranges (termed for convenience 'low vacuum' and 'high vacuum'), as follows:

Low vacuum $MAL = 2 \cdot 0.4089 \cdot 170^{0.60} = 17.8$ kg/h
High vacuum $MAL = 2 \cdot 0.2085 \cdot 13^{0.260} \cdot 170^{0.60} = 17.7$ kg/h.

In summary, the air leakage rates estimated by these different methods and specifications are as follows:

Basis for estimating leakage	Specification	Estimated air leakage (kg/h)
Rate of rise, Rein (2007)	New Pan	19.9
Rate of rise, Rein (2007)	Old Pan	39.8
Rate of rise, Rein (2007)	For vacuum pump sizing	59.7
Equation of Gomez (1991)	'Low vacuum'	28.4
Equation of Gomez (1991)	'High vacuum'	20.8
Equation of Ryans and Croll (1981)	'Low vacuum'	17.8
Equation of Ryans and Croll (1981)	'High vacuum'	17.7

The air leakage estimates of Ryans and Croll (1981) and the 'high vacuum' estimate of Gomez (1991) are all relatively close to Rein's (2007) specification for a new pan. The figure recommended by Rein (2007) for vacuum pump sizing is, however, three times the standard for a new pan. This indicates that standard sugar industry practice is to allow for a substantial degradation in pan 'tightness' before the vacuum pump becomes overloaded and is unable to maintain the desired vacuum in the pan.

Mechanism 3 – Air from feed:
An approximate estimate of the maximum rate of syrup or molasses fed to a pan is given by calculating the flow rate that provides sufficient water to match the maximum evaporation rate in the pan. For the maximum evaporation rate of 28.05 ton/h and the feed brix of 66%:

Maximum syrup feed rate = 28.05 / (1 – 66/100)
= 82.5 ton/h.

Hugo (1986) gives a figure of 100 mg/kg for the air content of syrup. Given the figures for the solubility of air in water of around 20 mg/kg (and that the syrup is only 34% water) the estimate of air in syrup must be indicative of the presence of a substantial quantity of dispersed air in syrup under practical conditions. Since the contribution of this mechanism to the total quantity of incondensable gas is normally much smaller than the first two mechanisms already described, there has been little incentive to investigate this in much detail.

Using Hugot's (1986) figure of 100 mg/kg and assuming that all of the air is released in the pan, the air flow from this mechanism for the example pan is:

Air released into pan from feed = 82.5 ton/h · 1000 kg/ton · 100 mg/kg
= 8 250 000 mg/h
= 8.25 kg/h.

Mechanism 4 – Air from calandria venting:
The steam feed to the calandria will closely match the evaporation rate in the pan. Any air entering with this steam will ultimately have to be vented and, if it is vented into the pan body it will appear as gas that needs to be removed by the vacuum pump.

Honig (1963) provides the following estimates of the concentration of incondensable gas in multiple effect evaporator vapours (relevant here because vapour bled from the evaporator station is used as heating steam for pans):

Evaporator vessel	Grade of vapour	Concentration of air (mg/kg)
1st effect	Exhaust	5-15
2nd effect	Vapour One	30-40
3rd effect	Vapour Two	30-50
4th effect	Vapour Three	100-200

From this table it might appear at first glance that if a pan is using Vapour One as the heating steam in the calandria the incondensable gas contribution that could be vented back into the pan would only be approximately 35 mg/kg of heating steam used. However, if the pan calandria were throttled to operate under vacuum conditions the concentration of incondensable gas might well rise to the upper level quoted for Vapour Three, namely 200 mg/kg.

Considering these two extremes applied to the evaporation rate of the example pan:

Air from calandria venting (35 mg/kg) = 28.01 tons/h · 1000 kg/ton · 35 mg/kg
= 980 350 mg/h
= 0.98 kg/h
Air from calandria venting (200 mg/kg) = 28.01tons/h · 1000 kg/ton · 200 mg/kg
= 5 602 000 mg/h
= 5.60 kg/h.

Mechanisms 1, 2, 3 and 4 combined

In the absence of any knowledge of the magnitude of the fifth mechanism, the quantity of incondensable gas that needs to be removed from the pan by the vacuum pump is the sum of the contributions of each of the first four mechanisms. To provide a summary of the range of variability of the estimates for each of the mechanisms and their contribution to the total incondensable gas flow, the maximum and minimum estimates have been tabulated for comparison with the recommendations of Rein (2007), termed conventional design.

Mechanism number	Mechanism description	Design maximum (kg/h)	Conventional design (kg/h)	Design minimum (kg/h)
1	Air in injection water	114.2	40	22.8
2	Air leakage	59.7	59.7	17.7
3	Air in feed	8.3	8.3	8.3
4	Calandria venting	5.6	0	1
	Total incondensable gas	187.8	108	49.8

These data are plotted in Figure 1.

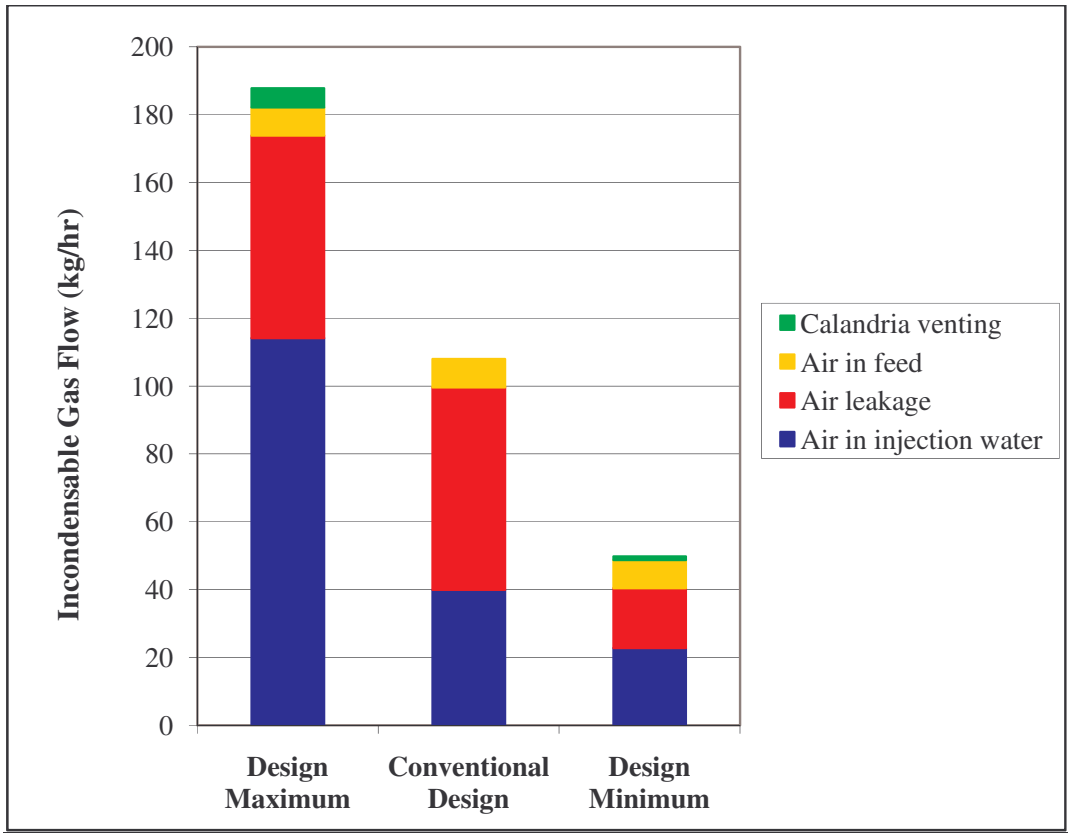

Figure 1. Comparisons of estimates of incondensable gas quantities for example pan.

These figures give a useful indication of the uncertainty in the estimate of the total incondensable gas quantity that will be the basis for sizing a vacuum pump.

Capacity of vacuum pump to remove incondensable gasses

A liquid ring vacuum pump is essentially a volumetric pumping device. The estimated mass flow of incondensable gas needs to be converted into a volumetric flow at the operating conditions of the pan (13 kPa abs for the example pan). The incondensable gas will be accompanied by water vapour since the gas will be fully saturated when it leaves the condenser. The proportion of water vapour will be determined by the temperature of the incondensable gas stream leaving the condenser and can be estimated from fundamental gas laws as described by Love (2005), taking account of the corrections given in Appendix 1 of this paper.

The vacuum pump has to be sized to remove the total volumetric flow of incondensable gas and water vapour. In selecting the pump, the quoted volumetric capacity of the pump (normally quoted for pumping dry air with a specified seal water temperature) needs to be corrected to account for the fact that the actual seal water temperature will normally be different from that of the specification, and to allow for condensation that will normally take place within the pump due to seal water that is colder than the incondensable gas stream.

Pump manufacturers normally supply graphs to determine the necessary correction factors. Rein (2007) provides some examples of typical correction factor graphs.

Conventional testing for load and capacity to remove incondensable gasses

When a pan is incapable of consistently achieving its design operating pressure (13 kPa abs for the example pan), conventional checks to assist in identify the cause of the problem are:

1. Ensure that the condenser is performing satisfactorily – The condenser is condensing as much of the pan vapour as is practically possible if the temperature of the incondensable gas leaving the condenser (and fed to the vacuum pump) is close to the injection water temperature (a difference of less than about 3ºC). The condenser should not need to use an excessive quantity of injection water to achieve this. A practical minimum quantity of water is being used if the tailpipe approach temperature (the difference between the saturated vapour temperature at the pan operating temperature and the tailpipe temperature) is less than 3ºC. Excessive water usage should not however be a major consideration with respect to the maximum incondensable gas contribution from this source, because appropriate sizing of the injection water valve will normally limit the maximum possible injection water flow to an acceptable level.
2. Check that the pan does not have excessive air leaks – When the pan can be removed from service, usually during the weekly stop day, conduct a vacuum test as detailed in Appendix 2. If the pan does not meet the specification for an acceptable leakage rate, find and repair the leaks. It is good practice to do regular vacuum tests on pans as part of a preventative maintenance program – which should detect this type of problem before it affects pan performance.
3. Check that the vacuum pump performance is acceptable – If a standby pump that is known to be performing to specification is available, put this into service in place of the current pump. If this solves the problem it indicates that the current pump is underperforming, providing both a diagnosis and a solution to the problem! For a more direct and quantitative test it is necessary to measure and plot the pump performance curve. The pump testing can be done either by sending the vacuum pump to a specialist repair shop (which may or may not be the local agent for the particular type of pump) or doing the test in the factory. Chilvers and Love (1986) described a simplified procedure for testing and plotting the performance curve of a vacuum and comparing it with the manufacturer's specification curve. This type of performance test could also be undertaken regularly as part of a preventative maintenance program.

Some major limitations of this approach are:

1. There are limitations on when a pan vacuum test can be done – as the pan needs to be both empty and taken out of service for approximately one hour. If the loading on the pan floor is light, it may be possible to do the vacuum test on a batch pan between normal pan cycles, otherwise the test would need to wait until the normal weekly stop day. Continuous A-pans are normally only emptied and cleaned on a two-week cycle, whilst continuous B-pans are often not emptied for months at a time. Continuous C-pans are not normally emptied and cleaned for the entire season.
2. The vacuum test only measures one of the five factors that contribute to the quantity of incondensable gas that needs to be removed from the pan.

3. Despite the simplicity of the vacuum pump test proposed by Chilvers and Love (1986) this does not appear to have been taken up as a standard procedure in factories (perhaps because it requires a non-standard piece of test equipment to be manufactured and then kept maintained and available until required – one more administrative headache). Although the test as originally described used simple arithmetic and a graphical procedure to plot the pump performance curve, the calculation procedures could easily be upgraded for the 'PC and spreadsheet age' to allow pump test results to be to be converted into performance curves and have the full test results printed out using a standard spreadsheet program.

Given these limitations, it would be useful to have a simple test that could be conducted on a pan during normal factory operations to help diagnose an incondensable gas problem.

A new test for trouble-shooting incondensable gas removal from pans

Given the limitations of the conventional checks to identify a problem with incondensable gas removal from a pan, a test that measures the quantity of incondensable gas removed from a pan during normal pan operation would be a very useful diagnostic tool. The easiest place to measure the incondensable gas flow is at the outlet of the liquid ring pump. Although the outlet from the pump is a combination of both the incondensable gas and seal water that is continuously flowing through the pump, it is normal (and good practice) to have an air/water separator on the outlet of each pump. The separator will normally have a water seal on the outlet that returns water to the vacuum pump cooling tower and ensures that all the incondensable gas is vented to atmosphere through the vent on the separator.

A very simple device for measuring gas flows from vents to atmosphere (originally proposed by Smith and subsequently recommended by the United Nations Industrial Development Organisation (Anon, 1991)) is shown in Figure 2.

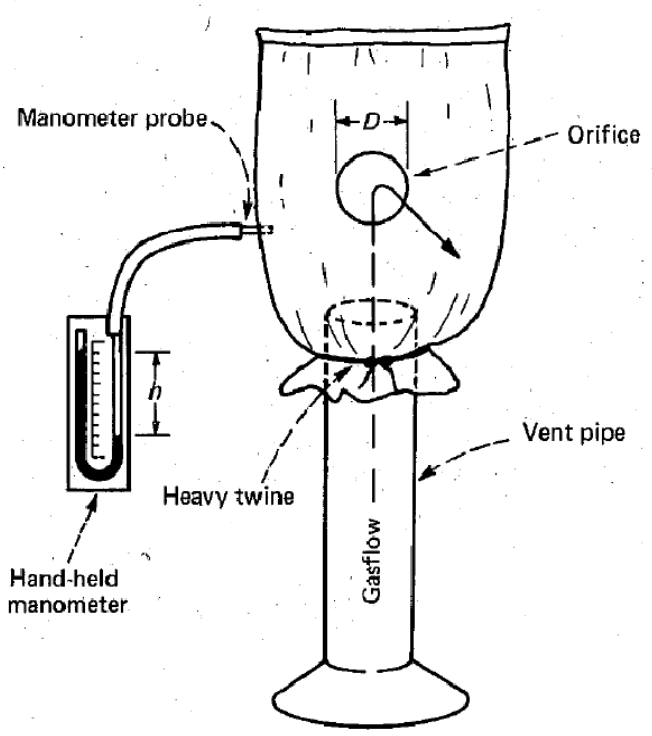

Figure 2. Simple method for measuring air flows from vents (from Smith).

In a more modern context, the 'heavy twine' would no doubt be replaced by a plastic cable-tie. The details of the test method and the associated calculations are given in Appendix 3.

This test procedure has enormous potential for trouble shooting problems associated with incondensable gas removal from pans and evaporators.

In the first instance, a single test on an operating pan will provide a direct measurement of the quantity of incondensable gas being removed from the pan at that time. This can be compared with the quantity that would be expected (using the standard methods for estimating incondensable gas quantities that have been described in this paper ie mechanisms 1 to 4). The measurement will also provide a single point on the pump curve, allowing the current capacity of the vacuum pump to be compared with its design capacity.

The influence of the various mechanisms that contribute to incondensable gas ingress into the pan can be investigated using this test procedure. If the calandria is not operating at a sub-atmospheric pressure, incondensable gas venting from the calandria can be switched between venting to the atmosphere and venting into the pan and seeing how this effects the quantity of gas rejected via the vent.

If the flow from the vent is measured while the steam to the pan is shut off for a short period and the injection water valve is shut, this will give an indication of the contribution of air from injection water to total incondensable gas load.

In the preparation for a conventional vacuum test on a pan, if regular flow measurements are taken whilst the vacuum pump is used to raise vacuum in the pan, the flow measurements can

be combined with the corresponding pressure measurements in the pan to plot a performance curve for the pump.

Conclusions

The standard methods used for estimating the quantities of incondensable gas to be removed from a vacuum pan have provided a successful means for sizing vacuum pumps for new installations. However, a review of the methods used for estimating each of the mechanisms that contribute to the total incondensable gas load has indicated that they are often based on old parameters (e.g. concentrations of incondensable gas in injection water, pan feed and heating steam) that do not appear to have been checked or updated to see that they remain applicable in all local instances. Comparing sugar industry norms for estimating incondensable gas quantities with methods from the broader chemical processing industries has shown some interesting differences.

In the light of the uncertainties in estimating the quantities of incondensable gas to be removed from a pan, a simple method of measuring the actual quantities of incondensable gas being removed from a pan during normal operation is proposed. By varying operating conditions, the test method has the potential for investigating the magnitude of the contributions that different mechanisms make to the total quantity of incondensable gas that needs to be removed.

The test method can be a useful diagnostic tool for problems with incondensable gas removal from a pan or final effect evaporator. A single test during normal operation gives a measurement of the actual quantity of incondensable gas being removed – which when compared with design estimates of the expected incondensable gas flow and the specified capacity of the pump will immediately give an indication of whether the pump is underperforming or it is being overloaded.

The method can also be used for a comprehensive measurement of the performance of a liquid ling vacuum pump, enabling a pump performance curve to be plotted and compared with the manufacturers specification.

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Appendix 1:
Corrections to calculations in paper by Love (2005)

In Appendix 1 of the paper by Love (2005) the mass and mole fractions of a stream, X, comprised of a mixture of air and water are denoted by:

FXw	mass fraction of water
FXa	mass fraction of air
fXw	mole fraction of water
fXa	mole fraction of air.

The equations that are given for calculating mass fractions from mole fractions and vice versa in the paper are incorrect. The correct formulae and the incorrect versions that they replace in Appendix 1 are:

$$FXa = \frac{1}{1+\frac{mw}{ma}\cdot\left(\frac{1}{fXa}-1\right)}$$ replacing: $$FXa = \frac{1}{1+\frac{ma}{mw}\cdot\left(\frac{1}{fXa}-1\right)}$$

$$FXw = \frac{1}{1+\frac{ma}{mw}\cdot\left(\frac{1}{fXw}-1\right)}$$ replacing: $$FXw = \frac{1}{1+\frac{mw}{ma}\cdot\left(\frac{1}{fXw}-1\right)}$$

$$fXa = \frac{\frac{FXa}{ma}}{\frac{(1-FXa)}{mw}+\frac{FXa}{ma}}$$ replacing: $$fXa = \frac{FXa\cdot ma}{mw\cdot(1-FXa)+FXa\cdot ma}$$

$$fXw = \frac{\frac{FXw}{mw}}{\frac{(1-FXw)}{ma}+\frac{FXw}{mw}}$$ replacing: $$fXw = \frac{FXw\cdot mw}{ma\cdot(1-FXw)+FXw\cdot mw}$$

The specific versions of these formulae used in Appendix 2 of the paper by Love (2005) for calculating the mass and mole fractions of water in the incondensable gas stream from the condenser (*FIw* and *fIw* respectively) must thus be modified to match the corrections given above.

Appendix 2:
The rate of pressure rise vacuum test

The procedure for conducting a rate of pressure rise vacuum test on a pan is as follows:

1. Empty the pan.
2. Isolate the pan by closing all valves including the injection water valve, but leaving the vacuum pump connected.
3. Isolate the calandria – including shutting all incondensable gas vents even if they are connected back to the vapour space of the vessel.
4. Draw the maximum vacuum on the vessel using the vacuum pump.
5. Isolate the vessel from the vacuum pump and stop the pump.
6. Time the rise of pressure within vessel – if the pressure rise is relatively fast record the number of minutes to rise 10 kPa – if the pressure rise is relatively slow record the pressure rise that takes place in 60 minutes.

The test and calculation are based on the assumption of choked flow through the small leaks. Choked flow will prevail as long as the pressure in the pan does not rise to greater than 53% of atmospheric pressure during the test (Perry and Green, 1997). With choked flow, the flow rate is independent of the downstream pressure and is thus constant over the period of the test. Knowing the volume of the pan, it is possible to calculate leakage rates from this test.

The calculation can be derived simply from the ideal gas equation:

$$P \cdot V = n \cdot R \cdot T$$

where:
P – is the pressure (Pa)
V – is the volume (m^3)
n – is the number of moles of gas
R – is the universal gas constant (8.3145 J / (ºK mol)
T – is the temperature of the gas (ºK).

Setting up two versions of the ideal gas equation to represent the conditions at the beginning and end of the test respectively (using the subscripts 1 and 2 to refer to the two conditions) and assuming the volume of the vessel and the temperature of the gas to remain constant, it is possible to subtract the one equation from the other to yield:

$$n_2 - n_1 = \frac{V}{R \cdot T}\left(P_2 - P_1\right)$$

To convert from the number of moles of air to the mass of air it is necessary to multiply by the molecular mass of air (28.964) and then, since the conversions from grams to kilograms and from pascals to kilopascals cancel each other out:

$$\Delta M = 28.964 \cdot \frac{V}{R \cdot T} \Delta P$$

Or, for an air temperature of 25ºC:

$$\Delta M = \frac{28.964}{8.3145 \cdot (273.1 + 25)} \cdot V \cdot \Delta P$$

$$= 0.0117 \cdot V \cdot \Delta P$$

where:
ΔM is the increase in mass in the vessel (kg)
ΔP is the pressure rise (kPa)

Dividing ΔM by the time taken for the pressure rise will give the air leakage rate.

Appendix 3:
Use of the bag orifice

Based on the original recommendations of Smith for using this 'bag orifice' means of measuring vent flows it is possible to provide the following details of the measurement technique and the associated calculations (with units converted from the original Imperial units into metric units and 'rounded' appropriately):

The estimation of the air flow rate out of a vent using the bag orifice technique is based on the standard equation for the flow of gas through an orifice, but ignoring the compressibility factor because of the low pressure drop used in the measurement:

$$Q = K \cdot A \cdot \sqrt{2 \cdot g \cdot h}$$

where, in consistent units:
Q is the volumetric gas flow
g is the acceleration due to gravity
A is the orifice area
h is the head loss across the orifice (measured as head of flowing fluid)
K the discharge coefficient of the orifice including the velocity-of-approach factor

Since the bag size is selected to be large relative to the orifice diameter, the velocity-of-approach factor can be taken as 1.0 and thus the value of K is simply the conventional orifice coefficient C.

Smith states that experiments with different bag thicknesses, flow rates, air densities and orifice sizes enabled the orifice coefficient to be estimated as:

$$K = C = 0.7089 \pm 0.0290$$

independent of bag thickness for the three different bag thicknesses tested.

(Note that the value of 0.7089 is misprinted as 0.07089 in the article in the Calculation and Shortcut Deskbook.)

When the head loss is measured as head of water, h_w, rather than head of the flowing air, the relationship between these two measurements of head is:

$$h = \frac{\rho_w}{\rho_g} \cdot h_w$$

where, in consistent units:
h_w is the head loss across the orifice measured as head of water
ρ_g is the density of the flowing gas
ρ_w is the density of water.

Assuming the water in the manometer is at 25ºC (and thus has a density of 997 kg/m^3) and accommodating a circular orifice, it is possible to write the orifice equation in metric units as:

$$Q = 2.462 \cdot 10^{-6} \cdot d^2 \cdot \sqrt{\frac{h_w}{\rho_g}}$$

where:
Q is the volumetric gas flow in m^3/s
d is the orifice diameter in (mm)
h_w is the head loss across the orifice measured as head of water (mm)
ρ_g is the density of the flowing gas (kg/m^3).

In selecting a suitable orifice size, a pressure drop of 25-100 mm water gauge should be sought. Less than 25 mm is difficult to measure, and greater than 100 mm may make the bag slip off the vent pipe.

If a rough estimate of the gas flow is known, the orifice diameter (mm), necessary to produce a pressure drop of 63 mm, is approximately:

$$d = 7.65 \cdot \sqrt{q}$$

where:
d is the orifice diameter in mm
q is the gas flow in litres/sec.

Several features of the design can minimise error. These are as follows.

1. The position of the manometer probe should project slightly through the bag wall, so that the axes of the vent pipe, the bag orifice and the probe end are all perpendicular (see Figure 2), so that a true indication of static pressure can be obtained.
2. The bag should be large enough to minimise the effects of approach velocity and to prevent flapping or tearing.
3. The orifice diameter should be measured during operation, so as to obtain true operating dimensions; if stretching causes an elliptical orifice, the area should be based on the product of the major and minor axes.
4. Thin-walled bags, high temperatures and high velocities should be avoided since fluting outward of the orifice edges will tend to occur; when pronounced, the effect would be to increase the discharge coefficient as the shape of the orifice approaches that of a nozzle.