STEAM AND VAPOUR DISTRIBUTION

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General

Whole books, chapters of books, papers, correspondence courses etc. etc. *ad infinitum* are available on the subject of steam, its uses, generation, conservation and distribution. It is not the purpose of this paper to serve as a substitute for this literature or even to quote from it more than is necessary. The purpose of this paper is to consider the problems of steam and vapour distribution with regard to the Cane Sugar Industry in South Africa, with some reference to economy.

From this point of view it is considered best to follow the process of design and to consider the problems in detail as they arise. This gives rise to the first problem which is, where should the design start? In some industries, textiles for instance, steam is generated, distributed to whichever machines or processes require it, possibly with some pressure reduction where required, and what condensate can be recovered is returned to the Hot Well for boiler feed. In such cases, all that is needed is a plan of the building, showing the positions of the vessels and boiler, and upon this with little trouble can be marked the runs of the steam distribution. In a Sugar Mill this is far from being the best point at which to start. First of all, where turbine mill drives are used, the steam not only is distributed but must also be re-collected. Secondly, due to multiple effect evaporation, more than one type and "quality" of steam is available. Thus the correct point at which to start the Steam distribution design is that which is used in the Chemical and Oil industries, namely the Flow Sheet.

Flow Sheets

Flow sheets have many advantages for initial design. They are completely diagrammatic and, therefore, a pipe which will wind tortuously through the length of the Mill may be shown as a straight line. Vessels and equipment may be shown in positions which will make the reading and understanding of the flow and process as simple as possible. In the early stages of design, alterations and redirection of flow can easily be made and understood. Pipes may be sized in most cases before the general arrangement drawings showing their position in the Mill are commenced. This avoids the necessity for repositioning piping on the building plan due to it being larger than at first anticipated. And so on and so forth.

It is in the Flow Sheet stage, and at no other stage in the design procedure, that the amount of fuel economy should be decided. Whilst, of course, it is possible to make revisions to design at any time, even after the completion of construction, it is at no time cheaper or more easily accomplished than when the flow sheets are being prepared.

The extent of the Steam and Vapour distribution must now be considered. In this connection, the flow of steam and vapour must be followed through to the bitter end, even in the condensed state in order to give the complete picture. If maximum economy is to be attained, every pipe which discharges to atmosphere must be treated with suspicion and examined to ascertain whether useful heat is being thrown away. In the Cane Sugar Industry where the bagasse obtained from the cane is used as fuel, it is generally considered that fuel economy can be overdone and result in enormous quantities of surplus bagasse for which there is no market or use. This is a dangerous thought. The actual balancing of fuel against demand should be an operational problem and not one of design. Design should be based on maximum steam economy. It is very easy to waste steam even with a high efficiency system, but extremely difficult and often expensive to economise once construction is complete. Further, it should be remembered that markets seldom, if ever, exist for produce which is not available except in theory. Therefore, it is preferable for the flow sheets to be laid out on the basis of maximum efficiency. It must be borne in mind, whilst doing so, however, that less efficiency may be necessary under certain circumstances. Unless there is a shortage of water, reduction in efficiency is quite easily obtained by blowing off steam to atmosphere, etc.

The main points which need serious consideration in a Sugar Mill with regard to economy are:

(*a*) Exhaust steam should only be used for heating where it is not possible to use a lower "grade" heating media. This means in effect that it should only be used for the first effect of the multiple effect evaporator and for juice heaters where the leaving juice temperature is such that the use of other media would give too low a temperature difference.

(*b*) Condensate from many of the vessels is at a pressure and temperature above atmospheric boiling point. In these cases the condensate should be allowed to "flash" as soon as possible after leaving the heating surface and this flash steam collected and usefully employed.

(*c*) As all condensate is not required for boiler feed and much in fact is not suitable for this purpose due to contamination in multiple effect evaporation, the surplus condensate heat content should be used as far as possible. Where, as is the case in some Mills, the condensate is used for Maceration on the Mills, it must be cooled before its re-use. In this case the best method of cooling is by passing it through a heat exchanger for primary juice heating.

(*d*) Providing that the minimum process steam requirement is sufficiently greater than the maximum steam requirement for power generation, the possibility of vapour compression on the first effect of the evaporator station should be examined.

When the flow sheets have been completed, consideration can then be given to the preparation of general arrangement and detail drawings and the physical side of the pipework system. Before considering this latter, however, some comment regarding existing installations is considered necessary.

Where flow sheets exist from the original construction, alterations and additions can be quite easily planned, except for the final survey, in the comfort of an office. Even where no flow sheets exist, it is preferable to prepare at least a partial one of the part of the piping which is affected. Upon this may be tried the various alternatives until a satisfactory solution to the problem, whatever it may be, is obtained and only then is it necessary to embark on site measurements and the preparation of working drawings. The alternative of spending lengthy periods climbing around steam pipes and steelwork either when the Mill is working or when the weather is hot is not one which is to be lightly undertaken. The Engineer of an existing Mill can be sure that flow sheets will, in the long run, prove to be worth a thousand times their weight in perspiration.

We now pass to the design of the physical side of the piping, that is to say, the determination of size, route, wall thickness, location and size of valves etc., the order of procedure is generally as follows:

Pipe Sizing

This is a procedure which is quite often assumed by the uninitiated to be difficult and highly technical. For the Engineer who is faced with the problems of the natural circulation of hot water, the gravity flow of water supplies etc., this can often be the case. Indeed, some steam pipe sizing problems can be tricky, but for the Sugar Mill Engineer, who is concerned with the sizing of additions to existing process mains in the great majority of cases, no great difficulty exists.

A typical formula for the flow of steam through pipes is:

$$Z_1\text{-}Z_2 = 0.00010123 \ \frac{W^{1.889}}{d^{5.027}} \ L$$

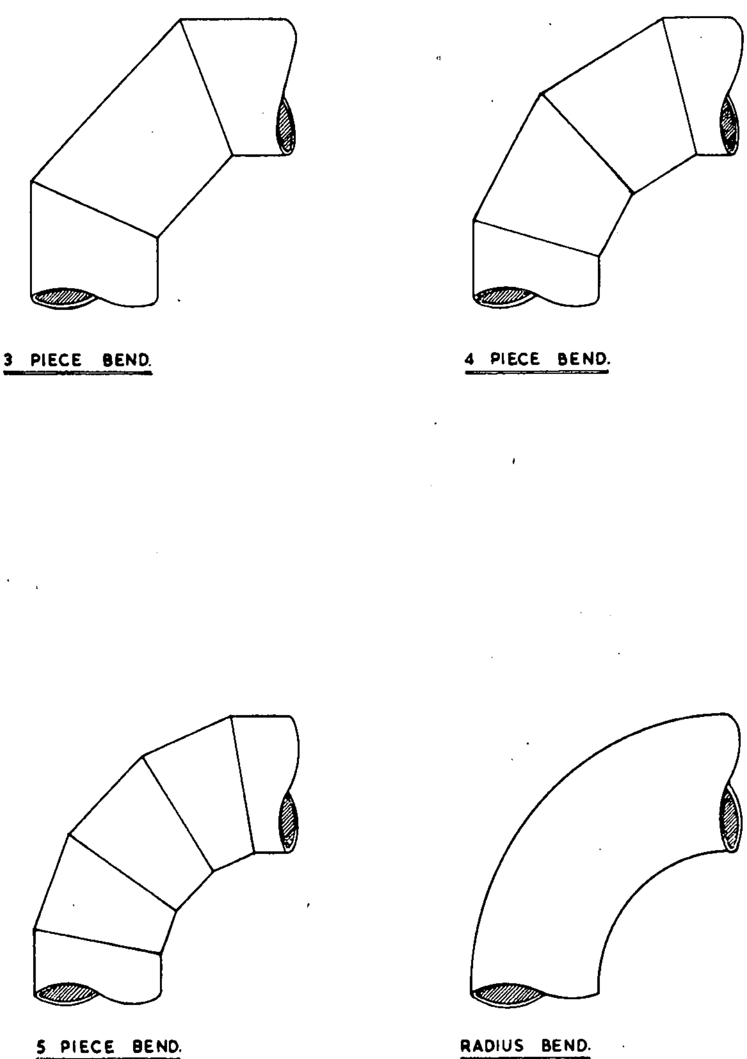

FIGURE 1: Typical Bends $1\frac{1}{2}''$ D.

Where Z_1 = Initial pressure factor, $P_1^{1.929}$
Z_2 = Final pressure factor, $P_2^{1.929}$
P = Absolute pressure lbs/in.^{2}
W = Rate of Steam flow lbs/hr.
d = Internal pipe diameter ins.
L = Length of pipe run ft.

This is, of course, enough to put anyone off for life. Fortunately however, pipe-sizing tables and charts are available in profusion and the solving of such formulae as that given above is very seldom necessary. The available charts, it is necessary to point out, seldom cover piping over 12 in. nominal bore and at first glance this is unsatisfactory for a Sugar Mill where a great proportion of the low pressure process steam piping is much larger than this and may even be as large as 48 in. diameter.

The steam pressures with which one has to deal in the Sugar Industry in general cover a wide range. On the high pressure side (steam for power) pressures may vary from 100 p.s.i.g. to 450 p.s.i.g. depending to a certain extent on the age of the Mill as the more modern trend is towards higher pressures with the trend towards higher efficiencies. On the low pressure side (exhaust steam etc. for process) pressures above 15 p.s.i.g. are not normally encountered and the final vapour from the multiple effect evaporator may be at a pressure of 2 p.s.i.a. (26 in. Hg. Vacuum). In the case of the high pressure side, piping is seldom greater than 12 in. diameter and recourse is best made to tables or charts for this piping. Two things must be checked. Firstly the pressure drop, which should not exceed 1 p.s.i.g. per 100 ft. run at 100 p.s.i.g. At higher pressures, a greater pressure drop may be tolerated, but care should be taken to check that this will not be so great as to effect the performance of the prime movers. Secondly, the velocity should be kept below 100 to 130 ft./sec. for saturated steam and 150 to 200 ft./sec. for superheated steam, regardless of the consequent pressure drop. In other words, if pipe sizes are selected to satisfy both the pressure drop and velocity conditions, the larger size should be selected. There are two reasons for limiting the steam velocity, namely to prevent excessive erosion of bends and to prevent excessive noise due to flow in the piping.

For the low pressure side, a different approach can be adopted. A few examples will show the reason why:

(i) For the same initial pressure and velocity, the pressure drop decreases with increase in pipe size:

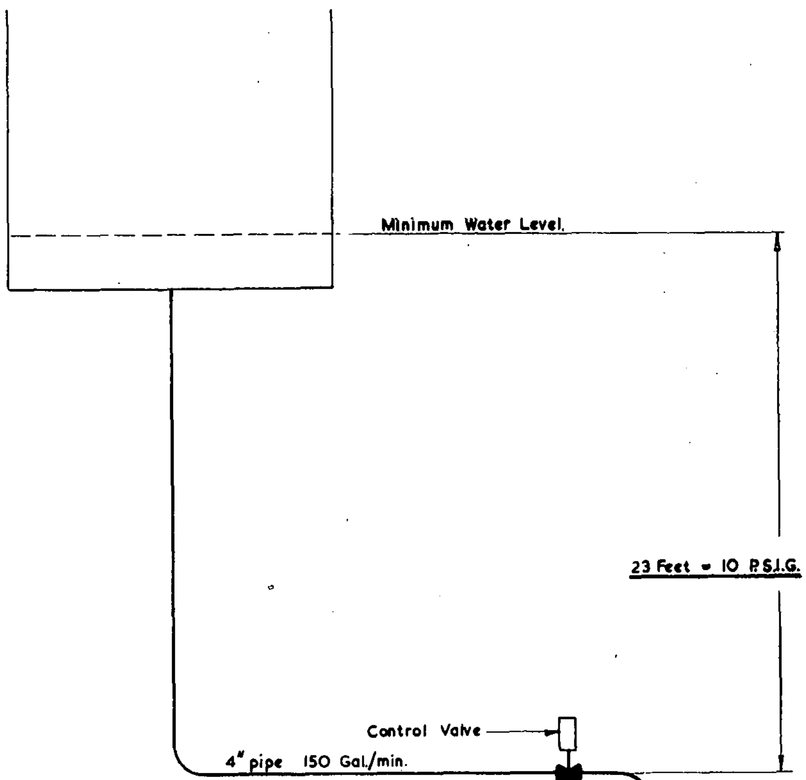

FIGURE 2: Water Control Valve.

Thus: For 15 p.s.i.g. and 100 ft/sec.

2 in. pipe pressure loss = 1.07 p.s.i.g./100 ft.
4 in. pipe pressure loss = 0.48 p.s.i.g./100 ft.
6 in. pipe pressure loss = 0.30 p.s.i.g./100 ft.

(ii) For the same initial pressure and pressure drop, the velocity increases with increase in pipe size:

Thus: For 15 p.s.i.g. and 1 p.s.i.g./100 ft.

2 in. pipe velocity = 94 ft/sec.
4 in. pipe velocity = 147 ft/sec.
6 in. pipe velocity = 194 ft/sec.

As few low pressure pipe runs exceed more than 200 ft. in the average Sugar Mill, and it can be assumed that a total pressure drop of 1 p.s.i.g. can be tolerated on an average, it will be seen from examination of the above figures, that even for a pipe as small as 4 in. diameter, the pressure drop will be acceptable if a velocity of 100 ft/sec. is used for sizing a 15 p.s.i.g. main. One other aspect needs consideration and that is the effect of initial pressure:

For equal velocity and pipe size, pressure drop decreases with increase in initial pressure:

Thus: For 4 in. dia. pipe and 100 ft/sec.

15 p.s.i.g. pressure drop = 0.48 p.s.i.g./100 ft.
10 p.s.i.g. pressure drop = 0.40 p.s.i.g./100 ft.
5 p.s.i.g. pressure drop = 0.35 p.s.i.g./100 ft.

Thus, it will be seen that the tendency towards the velocity being the controlling factor in this class of pipe sizing is accentuated by a lower initial pressure. However, due to the lower density at lower pressure, higher velocities can be tolerated and this tends to offset this last characteristic.

The above examples have been given merely to prove the general trend. Without going into further details, it can be accepted that the low pressure side of the steam piping in a Sugar Mill may be safely sized purely on the basis of velocity for piping over 6 in. diameter. Below this size, a check should be made on the pressure drop. In addition, where exceptionally long runs are encountered, it may be necessary to carry out a check on larger piping possibly even up to 12 in. diameter. This will rarely be justified.

Opinions vary considerably on what velocities should be used as a design basis. Probably the most representative figures are those given by Oliver Lyle in "Efficient Use of Steam". On condition that the exhaust steam pressure is 15 p.s.i.g. and is dry saturated, then the sort of velocities which can be used as a design basis are as follows:

15 p.s.i.g. (exhaust steam)	100 to 130 ft/sec.
5 p.s.i.g. (1st vapour)	120 to 160 ft/sec.
0 p.s.i.g. (2nd vapour)	150 to 180 ft/sec.
12 in. Hg.Vac. (3rd vapour)	160 to 220 ft/sec.
26 in. Hg. Vac. (4th vapour)	220 to 350 ft/sec.

Obviously these figures must be adjusted for particular cases but serve as an indication of normal sizing figures for this sort of application.

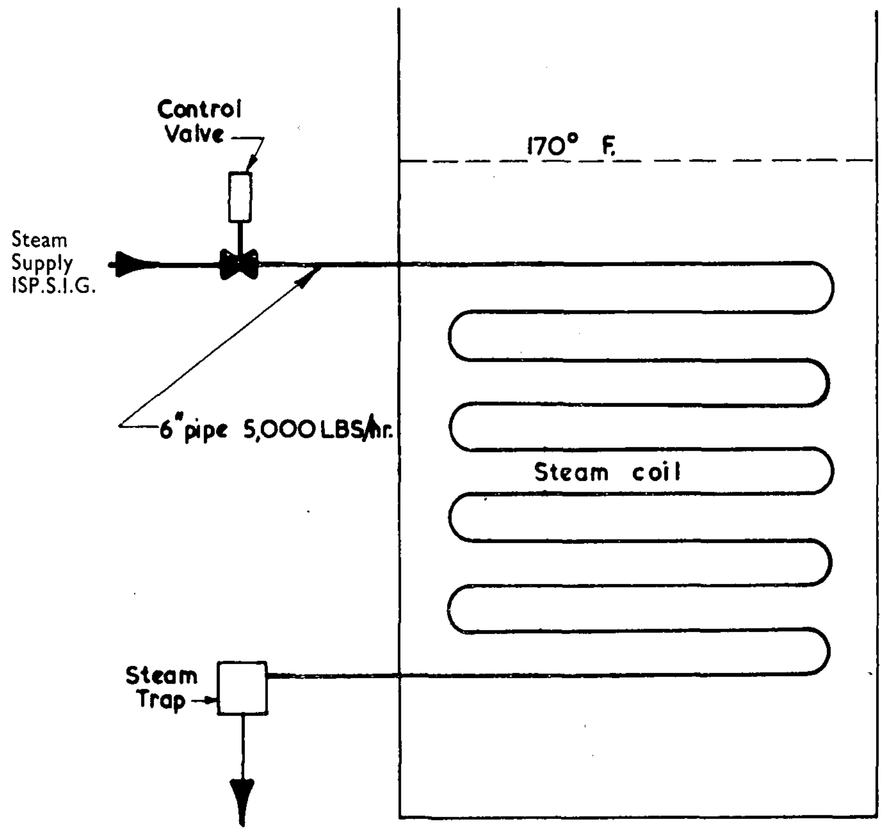

FIGURE 3: Control for Heating Application.

Once the velocities are decided, it is only necessary to apply the formula Q = AV, where Q is the quantity in cusecs, A is the cross-sectional area of the pipe in sq. ft. and V is the velocity in ft/sec. This may be converted to a more convenient form for steam pipe sizing thus:

$$d = \sqrt{\frac{Wv}{19.64V}}$$

where d	= pipe diameter	inches
w	= steam flow	lbs/hr.
v	= steam density	ft³/lb.
V	= steam velocity	ft/sec.

The final selection of pipe sizes depends, of course, on the standard sizes available. It will be noted from references to B.S. 10:1962 which gives details of standard pipe flanges up to 72 in. diameter, that an enormous range of sizes are standard. However, up to 12 in. diameter, 3½ in., 7 in. and 9 in. should be ignored as these are not normally manufactured. The latter two sizes sometimes occur on pumps but as a reducer is invariably fitted directly to the pump, they can be ignored from the point of view of actual piping. Above 12 in. it is normal to use outside diameter sizes, and although a considerable range of these is given, it is as well to standardise somewhat further, and a suggestion in this respect is that 16 in., 20 in., 24 in., 30 in., 36 in., 42 in. and 48 in. should suffice. The use of sizes at closer intervals has really only an academic advantage and only complicates maintenance etc.

Having established the pipe sizes, we must now consider:

Pipe Wall Thickness

This is adequately covered by British Standards 1387 and 806. These standards should be rigidly adhered to as far as all high pressure piping is concerned. On the low pressure piping however, more thought must be given to the matter. Piping over 12 in. diameter must be imported and is often expensive. Trouble may be encountered for piping between 8in. and 12 in. diameter. On low pressure piping, there is no reason why the larger piping should not be fabricated locally, and this course is normally adopted. The only question which may arise is in respect to the wall thickness. What should this be? If the formula given in B.S. 806 is applied to 15 p.s.i.g. working pressure for a 24 in. diameter pipe we arrive at a wall thickness of 0.0146 ins. which is approximately 28 s.w.g. Obviously no one in their right senses would contemplate either manufacturing or installing such a pipe on steam service. Other factors than pressure therefore become critical. Firstly, it is necessary that the piping shall be self-supporting over a reasonable length, secondly it must take stress set up by expansion and thirdly, corrosion must be taken into account. Generally for the low pressures and comparatively moderate temperatures involved, a wall thickness of ¼ inch will be adequate. This will also meet normal corrosion encountered on steam services. Under some circumstances, such as where there is carry over of sugar in vapour on the latter effects of the evaporator station, a greater wall thickness may be considered necessary but this is a matter for individual consideration.

Pipe Bends

On larger size piping, it is usual to employ fabricated "lobster back" bends as illustrated in Figure 1. From the economic point of view, it is obviously preferable to have as few "cuts" or "pieces" as possible. However, most engineers have been thoroughly frightened at an early and impressionable age by the bogey "extra resistance". It has been instilled into them that the biggest crime that they can commit on a piping job is to unnecessarily increase the resistance and pressure drop. On systems where limited gravity flow is available, this may possibly be true in extreme cases. On natural circulation for hot water heating or hot water supply, this can very often be the case, especially as the head available to produce circulation may be as low as 1.0 in. w.g. However, the case of steam distribution is somewhat different. Figures and formulae for the pressure loss in bends are available but not for the various types of bends shown in Figure 1 for steam service. This type of bend is, however, used quite frequently for air ducting (which is one case where resistance can be very important) and figures for this service are available. As the density of air at sea level is approximately 13.5 ft³/lb. and the density of steam at 15 p.s.i.g. is 13.73 ft³/lb. it is reasonable to assume that the figures for air may be applied to steam in order to ascertain how much difference is caused by the number of "pieces" used for a bend. The figures thus obtained may not be 100 per cent accurate but will prove sufficiently so for our purposes. The figures given by the I.H.V.E. "Guide to Current Practice" for the bends illustrated are as follows:

For radius = 1½ times pipe diameter

3 piece bend loss = 0.40 velocity heads

4 piece bend loss = 0.34 velocity heads

5 piece bend loss = 0.30 velocity heads

Radius bend = 0.24 velocity heads

$$\text{One velocity head} = H = \frac{V^2}{2g}$$

where H	= velocity head	ft.
V	= velocity	ft/sec.
g	= 32.2 ft/sec.²	

It must be noted that H is in feet of the flowing medium (air, water, steam, etc.). Thus for air:

$$H = \left(\frac{V}{4008}\right)^2 \text{ ins. w.g. where V is in ft/min.}$$

It can be seen above, that the difference in resistance between a 3-piece bend and a radius bend (6 pieces or more) is 0.16H. For typical pressures, some examples are as follows:

Pressure	*H*	*Velocity*	*Additional loss*
15 p.s.i.g.	$\left(\frac{V}{4080}\right)^2$	120 ft/sec.	0.50 in. w.g.
5 p.s.i.g.	$\left(\frac{V}{4910}\right)^2$	150 ft/sec.	0.54 in. w.g.
0 p.s.i.g.	$\left(\frac{V}{5700}\right)^2$	175 ft/sec.	0.54 in. w.g.
12 in. Hg.Vac.	$\left(\frac{V}{7150}\right)^2$	200 ft/sec.	0.45 in. w.g.
26 in. Hg.Vac.	$\left(\frac{V}{14450}\right)^2$	250 ft/sec.	0.17 in. w.g.

As 1 p.s.i.g. = 27.6 in. w.g. it is obvious that unless there are an enormous number of bends in a run of piping the difference is not worth worrying about and certainly not worth the additional expense involved in fabricating radius (or near to radius) bends. It should also be borne in mind that the insulation of bends is also more expensive the nearer they are to radius type, especially if sheet metal cladding is used.

Pipe Expansion

The subject of pipe stressing is lengthy and involved. A discussion of this subject would very probably double the length of this paper and would not really be of interest to the Sugar Engineer. Suffice to say that the stressing and layout of high pressure piping should be left to the specialist. Low pressure piping is not subject to the large expansion of high pressure piping and here, if normal good practice is followed, difficulties are seldom experienced. The piping should be left as free as possible to move with expansion, with plenty of length and bends on branches connecting to equipment. Anchors should be kept to a minimum and expansion devices fitted wherever it is necessary to confine a straight length of piping.

To be more specific, it is the following points which should be given particular attention:

(i) *Exhaust pipe connections to turbines* must impose as little stress as possible on the turbine. Not only must the expansion of the piping be considered but also the movement of the turbine casing. The use of expansion bellows is indicated in this case. However, it should be noted that the internal pressure in bellows acts on the internal faces of the corrugations and results in a longitudinal force which tends to

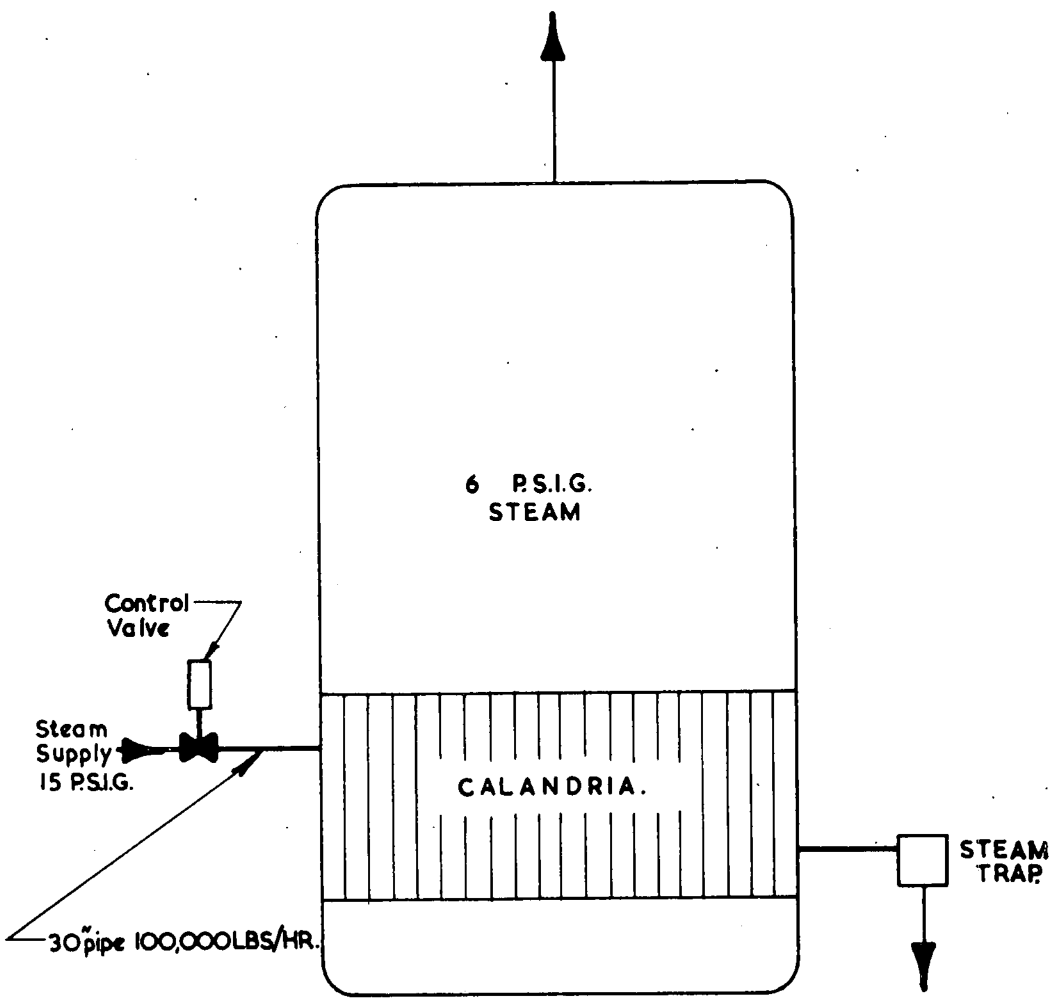

FIGURE 4: Control for Boiling Application.

lengthen them and which must be overcome in order to compress them to absorb expansion. This force may be transmitted onto the turbine unless a very complicated method of containing the expansion is adopted. For this reason it is better to use the lateral (or sideways) movement type of bellows on this application as the forces needed to move the bellows are very much less.

(ii) *Connections to vessels and equipment* should be designed so as to keep the stress acting on the vessel body to a minimum. Long branches from the main with one or two bends will usually suffice but each case should be considered separately. On one hand it must be realised that the force exerted by a 24 in. pipe when heating up from 60° F to 250° F is 377 tons, unless it is allowed to expand freely in which case the increase in length will be 1.6 in. per 100 ft. Both these figures should be halved by the use of "cold draw". This means that when the piping is installed, each run should be left with a gap between flanges equal to half of the calculated expansion. The flanges are then drawn together cold and the piping thus stressed in the opposite direction to the expansion movement, so halving the hot stresses. Even under these circumstances, there are some pretty frightening forces still at work and these should be as far as possible allowed to dissipate themselves on the piping. Where this is not possible and considerable forces act on the vessels, consideration should be given to the use of expansion devices to alleviate this. On the other hand, a visit to a few of the Sugar Mills in Natal will show that considerable stresses are exerted on many existing vessels without any detrimental effect. However, it is not advisable to assume that this is not therefore worth worrying about. When in doubt it is as well to check with similar cases in existing installations and if not 100 per cent satisfied, to play safe and fit expansion devices.

Pipe Supports

These should allow the free movement of the piping for expansion except where anchors are particularly required. Detailed design of supports is not a subject for a paper such as this as supports may vary considerably in detail. However, basically they fall into three main types. These are:

(i) *Hanger Supports*

Providing the hanger is of sufficient length, these allow free expansion of the piping for both longitudinal and transverse movement. Where vertical piping occurs the connected and adjacent horizontal piping will also be subject to vertical movement. In this case, spring type hangers should be used in order to accommodate this movement.

(ii) *Rollers*

Horizontal runs may be supported on rollers which will allow free movement of the piping.

(iii) *Sliding supports*

Where these are employed, the pipe should be fitted with a shoe, which should be of sufficient depth to allow the insulation to clear the supporting bracket and should also present as little area of contact to the support as possible.

The pipe supports should be arranged to give a fall on the pipe in direction of flow. Condensate is always present in the bottom of the pipe due to condensation to meet heat losses from the pipe wall. This must be drained. On the smaller piping, failure to provide adequate fall and drainage will result in the build up of condensate which may result in the formation of a "slug" which, driven along by the steam at high velocity, will cause hammer and possibly damage to the piping. On larger piping, this is not so possible and the fall provided may be less. In fact on the largest piping, a level run may be quite satisfactory although condensate left behind when the plant is shut down may give rise to corrosion problems.

Steam Valves

In the past, valves for larger piping have presented difficulties. Valves for 24 in. piping are extremely bulky and weighty. Further, in the past, gate or sluice valves have been the only type available although some large globe or angle valves have been manufactured where tight shut off is essential. Butterfly valves have now become available and solve some of the problems previously encountered. They are much less bulky, much lighter and are more easily operated. Further, for low steam pressures they can be arranged for tight shut off. Butterfly type non-return valves are also available. These enable automatic isolation where required (such as on turbine exhausts) and where automation is required, perform this task more economically. With this wider range of valves available, the Engineer may find easier solutions to many of his previous problems especially for automatic operation.

Whatever type of valve is used, the sizing of these must be carried out entirely separately from the piping. This does not of course apply to valves which are intended purely for isolation or change-over purposes. These are normally of the same size as the piping. Control valves, however, are an entirely different matter. Globe or butterfly type valves should be used for control, as the control characteristics of gate type valves are very poor. There exist a number of pressure reducing valves which have never worked properly. The manufacturers usually are blessed with the blame for this. In the great majority of cases, they do not work due to the fact that they are oversized, being the same size as the piping, or have been incorrectly installed. An oversized control valve will spend most of its life open only a fraction of its full travel and vainly trying to achieve control under these circumstances whilst its seat is destroyed by wiredrawing. A few examples of control problems are given to illustrate why the valve sizing should be considered separately:

Reference is made to Figure 2. This is a water control problem (purely hypothetical). The control valve is assumed to be of the butterfly type, as figures for various degrees of opening are available for this type which enable the full picture to be given. If the resistance of the piping is 8.5 p.s.i.g. at full flow of 150 g.p.m., then the following figures may be tabulated:

Full flow

4 in. pipe loss	8.5 p.s.i.
4 in. valve loss	0.44 p.s.i. at 60° open (full open)
4 in. valve loss	1.50 p.s.i. at 46° open
3 in. valve loss	1.40 p.s.i. at 60° open

From this it can be seen that the 4 in. valves, sized to suit the pipe will be a quarter shut even at full flow. The 3 in. valve will be virtually fully open at full flow.

Half flow

4 in. pipe loss	2.10 p.s.i.g.
4 in. valve loss	7.90 p.s.i.g. at 19° open
3 in. valve loss	7.90 p.s.i.g. at 27° open

As the head available is constant at 10 p.s.i.g. this must be the resistance of the system at all flow conditions, and as the piping resistance drops with reduction of flow, the valve must compensate for this. From the half flow figures it will be seen that the smaller valve achieves this with less movement than the larger, and thus has a greater movement available for the control of very low flows where accuracy is often important.

What the above examples illustrate is the better quality control that can be achieved by the use of the right size of valve which is almost always smaller than the piping. Not only is the smaller valve better, it is also cheaper! This is one of the few examples in this modern age where the better job costs less.

It should be mentioned that were the level of water in the top tank shown in Figure 2 to rise, the difference in control quality given by the different valves would be accentuated. On pumped applications this is particularly important as the pump head tends to increase with reduced flow. For good quality control, the valve resistance should be between 10 and 20 per cent of the total system resistance.

With steam (and gases) a slightly different approach is necessary. Control valves on these services have critical sizes. A particular size of valve has a maximum flow capacity; for a particular inlet pressure, regardless of the pressure differential across the valve, this maximum cannot be exceeded. If a valve is selected as nearly as possible to this critical capacity, control will be of the best quality possible. Figure 3 illustrates an example of this.

The following figures show the action of firstly a valve sized to suit the piping, and secondly, a valve selected for critical flow. Steam temperatures equivalent to the outlet steam pressure are quoted as it is assumed that the heat transfer, and thus the steam flow will be proportional to the temperature difference. The problem shown is one of temperature control in a tank.

Full Flow

6 in. valve 60° open (full)	0.33 p.s.i. drop 250° F
3 in. valve 60° open	7.10 p.s.i. drop 235° F

Half Flow

6 in. valve 25° open	16.00 p.s.i. drop 210° F
3 in valve 43° open	18.00 p.s.i. drop 203° F

Quarter Flow

6 in. valve 17° open	20.5 p.s.i. drop 190° F
3 in. valve 31° open	21.5 p.s.i. drop 186° F

It will be noticed that the differences in valve opening are very pronounced in this case. However, it will also be noticed that even at half flow, the outlet pressure of the control valves must be below atmospheric pressure. Unless a pump is installed on the condensate or sufficient height is available for natural gravity extraction, what will actually happen is that the steam coil will flood, thus reducing the effective heating surface. Another point is that although the 3 in. valve may be cheaper, the pressure loss through this even when full open will necessitate an increase in heating surface which may result in an overall increase in cost of the plant.

On high or medium pressure heating systems, this is not so much the case and the critical flow valve is the one to use. With the low pressure met in Sugar Mill process heating, it is preferable to fit the control valve on the condensate outlet. This gives flooding control, thus varying the effective heating surface to suit requirements. It also ensures that the full steam pressure is always available both for heating and for condensate disposal. As this valve is to handle condensate, with a certain amount of flash steam, it will be considerably smaller and consequently cheaper than a steam control valve on the inlet to the coil.

Figure 4 illustrates the control of the steam supply to an evaporator. Flooding control cannot be used on this application as it has been found in practice to be impractical. Neither can the valve be selected for critical flow. Such a valve for this application would be 14 in. and would have a full open (60°) pressure drop of 7.2 p.s.i. As in this example the total pressure difference available is only 9 p.s.i. this would result in such an increase in calandria heating surface as to be completely uneconomical. However, on such an application, variation in evaporation rate does not cover the full range from maximum to nil. Under operating conditions, a reduction from maximum to half should prove more than sufficient. Therefore, the valve may be selected more from an economic point of view with due regard to the available pressure difference. For instance a 30 in. valve would have a full open pressure drop of 0.19 p.s.i., a 24 in. valve would have a pressure drop of 0.50 p.s.i. The latter should be quite suitable and under some circumstances, an even smaller valve may be practical.

The moral of the foregoing is quite clear. Never, never size a control valve to suit the piping. Consider it's location carefullly. If information is not available to the Engineer to enable him to size the valve himself, then he should give the manufacturers full details of application, steam quantities, allowable pressure drop etc. Never, never consider the pipe size. Reducers are relatively cheap. Oversized valves are expensive, especially if they have to be thrown away and replaced.

Thermal Insulation

Thermal insulation or lagging is required to satisfy three basic conditions:

(i) to keep heat losses to a minimum to prevent wastage of fuel;

(ii) to prevent excessive heat loss from hot surfaces creating uncomfortable temperatures in the building;

(iii) to prevent operating personnel suffering injury from burning by hot surfaces.

Where fuel has to be purchased, (i) is of prime importance and insulation design practice is based on this. Any insulation designed on this basis will automatically take care of (ii) and (iii). Where, in a Sugar Mill, there is a surplus of bagasse, the importance of (i) tends to fall away unless a ready market is available for the surplus. However, in order to satisfy condition (ii) it will be found in practice, that very little reduction in insulation will be possible. Possibly some of the condensate piping may be left uninsulated but only 7 ft. above the floor as otherwise condition (iii) will not be met. Even with an average insulation efficiency of 85 per cent throughout (including vessels) the heat losses from hot surfaces in a 250 ton/hr. Sugar Mill will be equivalent to about 1,500 lbs. steam per hour. This represents a lot of heat. Therefore it is suggested that normal insulation design be used even when there is a bagasse surplus, although in such a case, efficiencies may be lowered and cold face temperatures raised in certain cases. Care should be taken when designing insulation to consider both the efficiency and cold face temperature. Surface coefficients play a large part in heat transfer, and it should be noted that whilst the surface temperature of sheet metal covered insulation is higher than that for a cement finished insulation of the same thickness, the efficiency is higher. This is due to the fact that the proportion of radiated losses and convection losses is different.

The use of sheet metal cladding as insulation on piping is now almost an accepted standard. This finish is not expensive and has the virtue of being far more durable than other finishes.

Whilst this is outside the scope of this paper, it is felt necessary to mention that condition (iii) should be considered with relation to the clarified juice piping. This has a surface temperature of up to at least 220° F in parts and this temperature is dangerous to personnel. When one considers that all but the most hardened humans cannot tolerate more than a temperature of 110° F when washing their hands, the danger is brought home to one more effectively. Any surface at a temperature greater than 180° F should be at least provided with a token insulation to protect personnel unless it is at least 7 ft. above the floor.

Condensate

As mentioned earlier in relation to flow sheets, condensate is essentially a part of the steam and vapour distribution. Detailed consideration of the condensate system would, however, it is felt, be out of place in a paper directed mainly at the actual steam side of the installation. Only a few points are therefore dealt with here.

Condensate is corrosive. Any idea that the Engineer may have of it being pure distilled water, may be safely dismissed from his mind. Condensate from the latter effects of the evaporator station may be contaminated with sucrose and therefore acidic. Even with a "clean" system, the condensate often picks up carbon dioxide or other gases and forms an acidic solution. In hospitals and industries where the condensate piping is fairly small in size, although extensive in run, it has been found time and time again that copper piping pays in the long run due to lower maintenance and replacement costs. The absolute minimum specification for condensate piping has been found to be galvanised mild steel. With the larger sizes of piping encountered in a Sugar Mill, copper piping is not available. Therefore galvanised mild steel piping should be used. A lighter tube may be used (medium weight) than if black piping were employed and the cost is therefore much the same. The life of the piping will be extended considerably and in the long run financial savings will accrue.

In a Sugar Mill, condensate must, sooner or later reach a stage at which it must be pumped in order to return it to the boiler feed tank or to dispose of the surplus in other ways. It must be noted that the condensate may be near to atmospheric boiling temperature and if the pressure in the pump suction pipe or in the pump casing should drop below the pressure at which the condensate temperature is equal to boiling point, cavitation will occur in the pump. This is not only noisy, it is disastrous for the pump itself and can result in the necessity for the complete replacement of the unit. Wherever condensate, or any hot liquid for that matter, is to be pumped, sufficient positive pressure must be provided at the suction of the pump to prevent cavitation. This will mean generally fitting the pump well below the collection tank from which it is drawing. The pump manufacturers should be consulted on this matter as different pumps have different requirements in this respect.

The bane of the life of an Engineer responsible for maintenance, is the task of maintaining vast numbers of steam traps. Fortunately, from this point of view vessels working under vacuum do not need traps as an atmospheric leg and seal will suffice, unless, of course, the height is not available for this in which case an extraction pump must be fitted. The draining of steam mains is of course one case where nothing can be done to reduce the number of traps. The case of the larger vessels, however, deserves more consideration. It is the practice in some cases to avoid the use of traps altogether by using U-tubes. These devices have it to be said in their favour that they are simple. That they are not the ideal solution is also quite obvious. Apart from their being unsightly, untidy and cumbersome, they will not pass air. This is an extremely important point. It is assumed that in this enlightened day and age, every engineer dealing with steam is aware of the evils of air being present in any steam heating system. Steam traps are, of course, available with thermostatic air releases. Normally, group trapping, that is to say, one trap serving two or more vessels is completely unsatisfactory, due to the fact that any difference in working pressure between the vessels will result in the condensate being unable to flow from that having the lower pressure. There is one case where this does not apply and that is when the steam trap is located at a lower level and the condensate piped separately from each vessel to the trap. The height of the vessel above the trap must be such that any differences in pressure can be balanced by a water column in the respective vertical condensate pipes. If sufficient height is available and this system can be adopted, separate traps need only be provided where it is essential to segregate the various "qualities" of condensate. Thus a trap would be necessary for exhaust steam condensate. A separate trap for first effect vapour condensate would be needed as this may be clean most of the time but is subject to occasional contamination. Also, in some cases, this may be capable of producing flash steam in which case, it must be separated from cooler condensates. Finally, a trap is necessary for the other grades of condensate. Thus with this system, a minimum of three traps are required. In practice it may prove necessary to provide four or five but even this number is reasonable especially if they can be located in a group. This system does away with U-tubes and their disadvantages, and at the same time centralises the main traps for maintenance. Unfortunately, standard steam traps cannot be used as they are not available for the required capacities. However, there is no reason why purpose-made traps cannot be used for these large sizes. The separate condensate pipes should be brought to and connected into a closed tank or cylinder. Each pipe should be fitted with a check valve to prevent back flow into idle vessels. The pipes should be carried down in the tank to below the design water level to ensure a seal. All that is needed to make this into a steam trap is an automatic valve on the outlet controlled by level in the tank so that if the water level falls, the outlet valve shuts. In addition, the trap should be able to pass air. An air vent on the top of the tank takes care of this and to meet all possibilities this should consist of a thermostatic device in series with a float device, thus preventing the egress of both steam and water. It should be noted that these external devices are easily accessible for maintenance unlike the standard steam trap which must be opened up for servicing. Whilst this system may not be the ideal answer to the problem, it is felt that it does tend to alleviate the maintenance problem whilst retaining the advantages of proper steam trapping.

Summary

Whilst, needless to say, piping design should generally follow the procedure followed on all engineering design of:

Preliminary examination
General Arrangement drawings
Detail drawings and
Specification of Materials,

it is suggested that in the case of piping, the general arrangement drawings should be preceded by the preparation of flow sheets. Further, particular attention should be paid to the sizing of low pressure piping (which may need a different approach than that given to high pressure piping), the economics of the bends to be used, the sizes of control valves, expansion, supports and insulation. The condensate system also needs particular attention in respect of pipe material, pumping arrangements and steam trapping.

It is further suggested that, the more attention that is given to the above in the design stage, the less will be the amount of attention required during the construction and the less the likelihood of alterations to the actual installation.

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