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Aftercooler Temperature Balance for Air Compressors in Plant Air Service

S. Nowacki

AFTERCOOLER TEMPERATURE BALANCE FOR AIR COMPRESSORS IN PLANT-AIR SERVICE

Stan Nowacki, Project Engineer
Engineering Department, Cleveland District
Republic Steel Corporation, Cleveland, Ohio

INTRODUCTION

Aftercoolers with $15^{o}F$ approach have, for many years, been used in conjunction with air compressors in general plant-air service. The attention given to the performance of this component was varied, ranging from very good in some installations, to fair or poor in the others. The problems related to condensate carryover were many, but were tolerated by compressed air users.

In recent years, conditions have changed radically. Production lines and related equipment are more complicated and refined. Plant-air is used frequently in instruments and control systems. Costs related to forced downtime of production lines have skyrocketed. Not surprisingly, compressed air users are insistant in their demands for dryer and cleaner air. The temperature of aftercooler exit-air has to be controlled more closely. Compressed air dryers are added to air systems for further improvements in the dryness of compressed air. Temperature compatability between aftercoolers and dryers must be assured. Hence, each air system must be thoroughly analyzed, and system design criteria must be established prior to making any decisions concerning equipment acquisition or facility design.

Of all the issues involved in an air system analysis, the least covered in technical literature, and the most unrecognized is that of system temperatures. Therefore, it deserves special attention.

In this presentation, the writer will propose and discuss the techniques for estimating aftercooler exit-air temperatures. The discussion will be subsequently expanded to include the entire air system. The subject of temperature balancing will be given close attention, and the actual methods for reaching the most practical solutions will be indicated.

ESTIMATING THE TEMPERATURES OF AFTERCOOLER EXIT-AIR

Aftercooler performance is the key consideration in the analysis of temperatures of an air system. Methods must be available to estimate the true temperature of aftercooler exit-air in actual operating conditions. To serve a practical purpose, these methods need not be precise, but they must be simple and dependable. In order to satisfy this need, the writer proposes the technique outlined below.

The starting point, as in the past, is the temperature of the cooling water at the aftercooler water inlet. However, instead of adding just the approach temperature of the aftercooler, we take into consideration more factors, each of which has a bearing on the temperature of the aftercooler exit-air. The basic computation will then have this form:

$$T_1 + T_2 + T_3 + T_4 = T_E \qquad \text{Equation (1)}$$

Where: T_1 = The temperature of the cooling water at the inlet to the aftercooler

T_2 = Aftercooler approach temperature

T_3 = The allowance for aftercooler fouling and plugging

T_4 = The allowance for the compressor water-side fouling and plugging

T_E = The temperature of the aftercooler exit-air

The above equation is purely empirical, developed by this writer to serve as a guide for the designers and operators of compressed air systems, and is not intended to substitute for the more advanced basic calculations.

At this point, it might be practical to add a few words concerning the symbols T_2, T_3, and T_4. In a strict sense, these symbols denote temperature rise. But in "shop terms" they are referred to as temperatures, not as temperature differentials. Therefore, it was at the discretion of this writer to decide whether to use the symbols ΔT or T. The writer chose the latter.

In the text which follows, the effect of the four variables, T_1 through T_4, on the temperature of the aftercooler exit-air will be assessed.

(1) Cooling Water Temperature T_1

The temperature of the water which is being used for equipment cooling depends on several factors, the most prominent of which are: the geographical location and the source of water. It can be either river, lake, deep well, city, reclaim water, the water from a high-temperature-gradient cooling tower or from a low-temperature-gradient cooling tower. Accordingly, the summer-high temperature of the water can be as low as 50°F in the far northern states or as high as 100°F or more in the deep south. In the area adjacent to the Lake Erie the typical summer-high temperatures are: city water, 65°F; low-temperature-gradient cooling tower water, 85°F.

When suitable cooling water is not readily available or the cost is excessively high, air-cooled aftercoolers can be deployed. In these cases, the symbol T_1 will denote the temperature of the cooling air; the dry-bulb temperature for dry heat exchangers; and the wet-bulb temperature for the evaporative ones.

Finally, it should be pointed out that some industrial establishments have firmly established policies concerning the source of water to be used for compressor cooling. In other establishments, options might be open concerning the selection of the water source.

(2) Aftercooler Approach Temperature T_2

Within the plant-air operating practice (100 psig, 7.03 kg/cm^2, 6.89 bar), the cooling of the air to within 15°F (8°C) of the incoming water temperature has become a virtual standard, references (1, 2, 3, 4, and 5).

This temperature differential between the air leaving an aftercooler and the water coming in, is usually referred to as the "approach temperature" of the aftercooler. Frequently, just the term "approach" is used. For example, in correspondence or in specifications concerning an aftercooler, one just writes "approach 15°F".

An aftercooler is sized by its manufacturer on the basis of compressor discharge air temperature as specified by compressor manufacturer. Then, the approach of the aftercooler is assessed and quoted at the cooling water rate which is specified by the manufacturer (vendor) of the equipment in his sale proposal and in his equipment specifications. An aftercooler, when new and clean, will have a significantly narrower approach than shown in the specifications. For example, a 15F approach aftercooler, new and clean, can give actually a 10F approach. Increased water flow will result in a much narrower approach of a clean aftercooler. However, as the fouling of the aftercooler progresses, these effects diminish substantially.

Aftercoolers of 15F approach are very practical, well known, and are well liked by the operators of compressors. However, in the view of new developments outlined in the introduction to this paper, conditions may arise under which a narrower approach is needed. On occasions, the aftercoolers with as low approach as 2°F (1°C) are being used (see the section of this paper dedicated to the air dryers).

Compared with the 15F approach aftercoolers, the low approach aftercoolers are much larger (longer), require more space, cost more initially (6), and call for a higher repair and maintenance expense. More clearance is needed for tube bundle removal and replacement. The handling of the tube bundle is more difficult. Finally, cooling water demand is higher. In spite of those drawbacks, low-approach aftercoolers have been recognized as an essential asset in controlling compressed air temperatures. Therefore, the acceptance of these aftercoolers is on the increase.

When an adequate supply of water is available but the quality is poor, some compressor operators favor the use of aftercoolers of the water-in-tube type in order to reduce the effect of fouling. Usually, the specifications call for a 15-degree approach. When the temperature of water is high or the quality is bad, the air cooled induced-draft-type aftercoolers can be considered. These units usually have forced-air circulation induced by an electrically driven fan. They are generally specified with a 20-degree approach, the reference in this case being the dry-bulb temperature of the cooling air.

(3) Allowance for Aftercooler Fouling and Plugging T_3

The allowance for the fouling of the water side of an aftercooler will depend primarily on two factors:

a. The quality of the cooling water.
b. The frequency of aftercooler cleaning.

Whether the system is once-through or recirculated, the main threats are: silt and mud deposition, biological growth, corrosion and iron oxide depositions, crystalline deposits of calcium carbonate and calcium sulfate, and perhaps dissolved fines and dust in the case of cooling tower water.

Depending on the combined effect of these threats, an aftercooler can be fouled and/or plugged in a few months of operation or might perform with a full efficiency without cleanups for as long as two or three years (7). Most air compressor operators favor scheduled cleanups of compressors and aftercoolers once a year, usually planned for late spring or early summer. In situations where there is no standby compressor capacity, more frequent cleanups would delay production and are difficult to arrange.

It is difficult to provide typical figures concerning the allowance for aftercooler fouling; occasionally the fouling can be very severe. It is not unusual that an aftercooler is operated at 20 or 30 degrees above the design temperature. Nevertheless, it is the opinion of this writer that with a concerted effort of the designers and operators of the facility the actual fouling allowance can be kept in a range of 10 degrees. When water of high quality is available this allowance can be narrowed down to a range of 5 degrees.

(4) Allowance for the Compressor Water-Side Fouling and Plugging T_4

In reciprocating air compressors, in dry screw and in forced-lubricated sliding vane machines, fouling is experienced in the intercoolers and on the water swept surfaces of cylinder jackets and heads. Plugging usually occurs in the multiple water passages of these machines. In other types of air compressors the following components are affected by fouling and plugging: centrifugals - intercoolers; oil-flooded helical screw - oil-to-water heat exchanger; liquid ring (recirculated water type) - water-to-water heat exchanger.

It is an extremely difficult and risky task to assess the fouling allowance for an air compressor. Perhaps a range of 10 or 15 degrees would be representative for reciprocating, dry screw and for forced-lubricated sliding vane machines. The more years a machine is in service the more difficult it is to remove the deposits and consequently the higher this allowance should be. In air compressors of other types, heat exchangers are the only components which can foul. Therefore, the problem of fouling is more localized and easier to deal with. Consequently, the fouling allowance for these machines could be kept within a range of 5 or 10 degrees.

(5) Examples of Computations

Herewith are a few examples of the computations based on the equation (1), for the areas of northern Ohio adjacent to Lake Erie, and for the summer-high temperatures of the cooling water:

Example (1)

A. City water 65F, aftercooler approach 15F, the compressor and the aftercooler are either new or thoroughly cleaned. In these conditions, the actual approach of the aftercooler is assumed to be 10F.

   65F + 10F + 0 + 0 = 75F

B. Same compressor, aftercooler and water temperature as case A, but the ideal cleanup point has been reached:

   65F + 15F + 0 + 3 = 83F

C. Same compressor, aftercooler and water temperature as case A, the need for a cleanup is evident:

   65F + 15F + 5F + 6F = 91F

D. Same compressor, aftercooler and water temperature as case A, the cleanup is overdue:

   65F + 15F + 10F + 10F = 100F

Example (2)

The same type of computations, for cooling tower water at 85F:

A. 85F + 10F + 0 + 0 = 95F

B. 85F + 15F + 0 + 3 = 103F

C. 85F + 15F + 5F + 6F = 111F

D. 85F + 15F + 10F + 10F = 120F

Once the temperatures of the compressed air leaving the aftercooler have been calculated as shown above, it should be determined whether these temperatures are acceptable by the various types of compressed air dryers.

The dryers and their temperature limitations will be discussed in the section which follows.

INLET-AIR TEMPERATURE LIMITATIONS OF VARIOUS TYPES OF COMPRESSED AIR DRYERS

In the preceding section a method was developed to estimate the temperature of the air leaving an aftercooler and entering an air dryer of the same air system. In this section the inlet-air temperature limitations for various types of air dryers will be examined and suggested procedures for efficient balancing of temperatures will be outlined.

An attempt will be made to identify the dryers which can accept elevated temperatures of inlet air. However, in some cases there is no other option but to refer to equation (1), and to try to reduce the temperatures T_E. Temporarily, our considerations will be restricted to summer-high temperature conditions of air system operation. Air system operation under winter-low temperature conditions will be discussed in a later part of this report.

Insofar as practical, this discussion will be confined to the issue of temperatures. The readers who might look for other information concerning air dryers, are referred to the works of Arnold L. Weiner (4), Tom Ryan (6) and William O'Keefe (8).

Wherever the term dew point is used, it refers to the pressure dew point. This is the only dew point of interest in the compressed air practice as is obvious to the reader.

(1) Deliquescent Type Air Dryers

In the air systems which incorporate dryers of the deliquescent type, the possibilities for balancing of temperatures are very limited. A deliquescent dryer, with the commonly-used desiccant charge, gives a 20°F (11°C) depression in the dew point. Accordingly, with the inlet air to the dryer at 100F, the exit air from the dryer will register a 80F dew point. There are not many industrial plants where this dew point will satisfy the consumer. When a production department invests money in a dryer, a much better result is expected.

In general-service plant-air systems, a summer dew point within the range 50 to 60 degrees F is acceptable. In order to attain such dew point, the air directed from the aftercooler to the dryer should not exceed 70 to 80 degrees F.

In example (1), it can be noted that we can be within this range if we install a 5-degree F approach aftercooler and if we keep the combined fouling allowance down to 10F. Then, our T_E will be 80F, and the resulting dew point will be 60F. Regarding example (2), one has to conclude that a deliquescent type dryer, filled with the commonly used desiccant, cannot be recommended for the application which we have mentioned.

Desiccants which are more efficient in dew point depression are being advertised. If these desiccants prove practical, the range of application of deliquescent dryers will increase and a suitable temperature balance would be more easily attainable.

(2) Refrigeration Type Air Dryers

Refrigeration type dryers reduce the dew point of dryed air to a range of 35F to 38F. In the plant-air practice, this dew point is assumed to be 38F (3°C). For more information about process-temperatures of refrigeration type dryers, the reader can refer to John Luciana (9).

These dryers are usually rated for 100°F (38°C) inlet air and 100°F ambient. However, their application is not confined to these temperature limitations. On the contrary, they can be adapted for service at substantially higher inlet air temperatures, providing that they are sized suitably.

In general plant-air practice these dryers are used up to 120F inlet air temperatures. The specific information on sizing of a dryer must be obtained from the manufacturer of the dryer under consideration. In rough terms, for handling 120F air, a 50% addition to dryer capacity would be required. For example, when 1,000 cfm of air at 120F is to be dryed, a 1,500 cfm dryer would be needed.

Some dryer manufacturers might be willing to supply dryers for inlet air temperatures exceeding 120F. Then, very thorough attention must be given to the safety of refrigerant system at dryer shut-down conditions.

(3) Regenerative Air Dryers

When a low dew point is required, regenerative dryers must be used. These dryers have been known for many years in instrument-air systems. They are now also being accepted more widely in the plant-air systems. For such applications, the usual temperature specifications are as follows: inlet air 100°F (38°C), dew point -40°F (-40°C).

Capital and operating costs of these dryers are higher than those of other types of dryers. Therefore, the cost factors become essential in considering regenerative dryers.

Sizing of these dryers for elevated inlet-air temperatures leads to a substantial increase in operating costs. Therefore, efforts should be made to hold inlet-air temperatures down to a level of 100F. When these efforts fail the designer may be justified in considering a dryer sized for higher inlet-air temperatures.

The tolerance for elevated inlet-air temperatures will depend on the design and the size of a unit.

Small dryers, heatless models:

Heatless models are usually selected up to a range of approximately 300 cfm (510 m^3/h, 141.6 litres/sec.). They are relatively tolerant to inlet-air temperatures exceeding design criteria. Some of their manufacturers indicate that these dryers can handle up to 120F inlet-air at the air flow rated for 100F inlet-air. Of course, when it is known that T_E will reach 120F, one should not depend on this tolerance, and a 120F inlet-air dryer should be installed.

Small dryers, heat reactivated models:

Heat reactivated models are commonly used in the capacity range exceeding 200 cfm. They have a low tolerance to inlet-air temperatures exceeding design criteria. Therefore, the temperature T_E must be closely monitored and controlled. The practice of sizing these dryers for inlet-air temperatures up to 120F is acceptable.

Large dryers:

Regenerative dryers in the capacity range exceeding 1,000 cfm (1,700 m^3/h, 472 litres/sec.) require more detailed attention than the small units. The cost factor becomes a major issue.

As a rule, these dryers should not be operated at inlet-air temperatures exceeding design criteria. The sizing of these dryers for elevated inlet-air temperatures is not practical either; this would lead to a substantial increase in dryer acquisition and operating costs.

Therefore, in most cases it is preferable to reduce the temperature T_E. In order to achieve the best results in lowering T_E, in addition to the conventional methods listed earlier in this presentation, several less conventional methods can be applied: 2°F approach aftercoolers, two-step cooling by tower water in the first step and by more expensive city water in the second step, two-step cooling with chilled water in the second step, or pre-drying compressed air with refrigeration type dryers installed upstream from regenerative units.

(4) Other Types of Air Dryers

Three other types of compressed air dryers are found in plant-air application. Each offers attractive opportunities to air system designers and operators. Of special interest is the fact that all these dryers can handle elevated inlet-air temperatures T_E. The sizing of the units for elevated temperatures is based on the same principles as those already indicated in the section on refrigeration type dryers.

Refrigeration type rated for a constant dew point depression:

Two models of this kind are known, one for a 50°F (28°C) depression, the other one for a 30°F (17°C) depression. The 50F models are preferable for general-service plant-air application.

Combination refrigeration/deliquescent units:

These units offer the user a determined dew point of 20°F (-7°C). This dew point suits very well the typical plant-air systems. Its manufacturer designates this dryer with the name "Refrigerated/ Desiccant Dryer".

Combination refrigeration/regenerative units:

The routinely quoted dew point is -40°F (-40°C). These dryers are particularly appealing from the point of view of energy saving (10). They have been put on the market only recently. The manufacturer uses the term "Heat Pump" for designation of this model.

PERILS OF EXCESSIVELY HIGH T_E TEMPERATURES

In the preceding pages we have discussed temperature performance of aftercoolers and temperature limitations of various types of dryers, but we have not faced the issue of what occurs where there is no temperature compatibility between an aftercooler and a dryer.

Below is an assessment of the situations where aftercooler exit-air temperatures exceed the temperature limitations of an air dryer.

(1) Air Systems with Deliquescent Air Dryers

The dryer will not dry the air when T_E temperatures become high. This develops when the drying of compressed air is most needed. Moreover, an excessive carryover of desiccant material to the air system might become a real threat.

(2) Air System with Refrigeration Type Dryers

When a dryer is in operation, pressure drop across the dryer will be higher, and a moderate increase in air dew point will be experienced. In the usual plant-air practice, these developments will not cause any major operational problems.

However, it is conceivable that the dryer can trip-off. This can happen due to power failure, or the overload protective control might trip the dryer off. Then with the hot air passing through and with the refrigerant circulation halted, an excessive build-up of pressure in the refrigerant system can develop. This condition can be hazardous. Therefore, the subject of safety at dryer shut-down condition must be considered in detail with the supplier of the machine.

(3) Air Systems with Regenerative Dryers

Carryover of powdered desiccant to the air system might become a threat. When a high temperature of air prevails, the desiccant will also become saturated and there will be no drying of air. It is conceivable that water carryover to the dryer will be so high that permanent damage will be done to the adsorbent, and replacement of the charge will be needed.

As can be seen above, none of the dryers is tolerant to excessively high inlet-air temperatures.

AFTERCOOLER EXIT-AIR TEMPERATURES FOR WINTER-TIME OPERATION

As shown in the earlier parts of this report, in the summer season it is usually practical, and in many cases mandatory to keep aftercooler exit-air temperatures T_E as low as conditions permit. The requirements for the winter-time operation are more complex, and will depend on the type of dryer.

(1) Deliquescent Dryers

Irrespective of the season, these dryers require the lowest air temperatures T_E that can be attained. The desirable mode of operation for winter-time is as follows: cooling water 35F, aftercooler approach 10F, combined fouling effect not exceeding 10F. By equation (1), the aftercooler exit-air temperature T_E will then be 55F. By subtracting a 20F dew point depression, the resulting dew point will be 35F.

Thus, the need for the lowest T_E temperatures is evident. In many cases, in addition to an early-summer clean-up of aftercoolers, another clean-up in early winter might be needed.

(2-a) Refrigeration Type Dryers for a 38F Dew Point, Refrigeration/Deliquescent Dryers, and Refrigeration/Regenerative Dryers

There would be no advantage in bringing low T_E temperature air to these dryers in the winter season. It is more economical, and equally practical to operate with relatively high T_E temperatures, just slightly below those which were used in the summer.

This mode of operation is fully attainable in actual compressor practice. The Power & Steam Department in the Republic's Cleveland District introduced this routine several years ago. They operate with aftercooler exit-air (i.e. dryer inlet air) at 100F in summer, and at 90F in winter. The flow of water through aftercooler is hand adjusted when required. Intercoolers (centrifugal machines) are on thermostatic control.

There are two basic gains attained by this mode of operation: one, compressed air pipe lines do not sweat, and two, reduction in cooling water costs can be achieved. If city water is used for compressor cooling, considerable money can be saved.

(2-b) 50F and 30F Dew Point Depression Models of Refrigeration Type Dryers

The routine indicated in (2-a) can be applied to the 50F depression models, and only in the low range of T_E temperatures, 80F and below. It is not practical for the 30F depression models.

(3) Regenerative Dryers

There are three basically different concepts concerning winter-time operation of regenerative air dryers; one of them is well known, the two remaining ones are less known.

The first concept refers to the usual mode of operation. Reactivation cycles are controlled by timers and are permanently set for the year-round operation. Same applies to the purge-air settings. Aftercooler cooling water is at a low temperature, the flow of water to aftercooler is not reduced, and the resulting T_E temperature is low. This mode of operation is practical when the cost of the cooling water is moderate, when there is no problem of pipe sweating, or when the condition of the desiccant is poor. This mode of operation is not justified in situations when the condition of the desiccant is good, pipes sweat, or when the costs of water are high.

The second concept is based on techniques which were described above in paragraph (2-a). Providing that the desiccant is clean and active, this mode of operation is practical and feasible. However, it might be difficult to sell this concept to the operators of air systems.

The third concept would call for efficient cooling of the air in the aftercooler, that is for a low T_E temperature, and for periodical resetting of the cycles of reactivation. For this concept, the cycles would have to be reset twice each year. This mode of operation would reduce the rate of reactivating air and energy, and would be especially suitable for the air systems in which large dryers are incorporated. The resulting money saving could be of a substantial magnitude.

This writer would like to hear from the participants of this conference about their opinions (experience) concerning the second and the third concept.

AFTERCOOLER EXIT-AIR TEMPERATURE AND PIPE SWEATING

The temperature of compressed air flowing through a pipe and the dew point of ambient air in the locations through which the pipe passes, are the factors which determine "pipe sweating". Pipe routing belongs to other areas of engineering, so within the scope of this discussion, we are concerned only with the temperature of compressed air which is delivered from an air dryer to an air system.

With reference to the deliquescent and regenerative air dryers, there is no inherent change in the temperature of the air going through a dryer. In the refrigeration type dryers, the temperature subsides by a margin of about 10F to 20F, with the first number related to a low range of T_E temperatures (50F or 60F), and the latter one to a high range of T_E (90F or 100F).

When dryers are operated as suggested in the preceding pages of this paper, the manifestations of pipe sweating will be most pronounced in conjunction with deliquescent dryers, less pronounced with refrigeration-type units, and least noticeable with regenerative ones. When corrective action concerning pipe sweating is needed, these propositions can be evaluated:

(a) Operating at high T_E temperatures, providing that the required dew point of the dryed air is maintained.

(b) Insulating pipe lines. This is a rather expensive undertaking which might be justified in a few selected applications, but in most cases is unacceptable.

(c) Reheating the compressed air in a water-to-air heat exchanger installed down-stream from the air dryer. Water from the aftercooler can be piped to this air heater. In many cases this proposition might prove most practical and most feasible.

SUMMARY

This paper has attempted to present the basic concepts related to the air-cooling performance of compressor aftercoolers. A simple formula to estimate aftercooler exit-air temperatures has been developed, and its use was demonstrated.

When aftercooler temperatures were closely examined, it became apparent that our considerations could not be confined to this single component of an air system but had to be expanded to include air dryers. The interrelation between the capabilities and the limitations of aftercoolers and dryers was given close attention.

The range of application of various types of aftercoolers and air dryers was discussed, the subject of temperature balancing was examined, and suggestions concerning the most suitable modes of operation for the summer-high and for the winter-low temperatures have been outlined.

Safety, dependability and economics of air systems were considered throughout this presentation.

CONCLUSIONS

1. The frequent use of air dryers in new installations, and the very real prospect that air dryers will become standard in all air systems in the years to come, impose strict requirements on the temperatures of compressed air which is being handled. Under these new requirements, the practice of routinely specifying a $15^{\circ}F$ approach aftercooler, and a $100^{\circ}F$ inlet-air dryer, has become outdated. Instead, it is mandatory that detailed attention be given to temperature analysis of each air system to be installed or expanded.

2. Cooling water temperature and quality, are the key factors involved in the balancing of temperatures of an air system. When these two factors are favorable, it is very easy to attain a suitable balance of air temperatures.

On the other hand, when the temperature of cooling water to an aftercooler is high or its quality is poor, the task of balancing the temperatures becomes complex.

3. There are many options available to solve the complex problems involved. The most common are: low-approach aftercoolers, water-in-tube type aftercoolers, air-cooled aftercoolers, two-step cooling of air, frequent cleanups of compressors and aftercoolers, air dryers sized for elevated inlet-air temperatures, and two-step drying of compressed air.

4. The basic decisions concerning a contemplated air system must be made in the early stages of a project, when many options are open. After the parameters of an air system have been established and the equipment ordered, no options remain.

5. The frequent practice of copying equipment specifications which can be found in old files, is not accpetable any more. Another frequent practice, obtaining "typical specifications" from a vendor and following them, must be forgotten altogether. The expectations that operators will upgrade a poorly designed system are not realistic; there is little the operators can do to upgrade an air system. So, as can be seen, there are no easy solutions which can replace proper procedures.

6. In brief, the final conclusion stemming from this presentation is as follows: There are no shortcuts in the design of a dependable and efficient air system, and all effort dedicated to temperature analysis, to design criterias and to suitable equipment selection, will be generously rewarded.

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