THE IMPACT OF AIR COOLED CONDENSERS ON PLANT DESIGN AND OPERATIONS

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Abstract

Air-cooled condensers were first introduced into the U.S. power industry in the early 1970's, but only during the last decade has the number of installations greatly increased, largely in response to the growing attention being paid to environmental concerns. The rising importance of this rather different technology for the condensing and recovery of exhaust steam calls for a broader understanding of the associated design and application principles involved, as well as of the performance monitoring techniques and cleaning methods that have to be applied.

This paper identifies the basic configurations of air-cooled condensers used in the power industry together with their advantages and disadvantages when compared to those exhibited by traditional steam surface condensers. The several factors that affect the performance of air-cooled condensers are described in detail, especially the consequences that result from the fouling of the finned-tubes. Measuring the performance of air-cooled condensers is, clearly, an important task and some methods are identified, alternative ways of presenting the data also being included.

To rectify the degradations in performance that result from external tube fouling, a number of cleaning procedures are described. Included among these are details of a new *automated* cleaning technology that has been successfully applied, and some significant performance improvements that have resulted from the use of this technique are documented.

Keywords: Air-cooled Condensers, Dry Cooling, Performance Monitoring, Cleaning, Maintenance

Introduction

One of the important features of the Rankine cycle, on which all fossil, nuclear and combined cycle power plants are based, is the condensation of the vapor exhausted from the LP turbine stage, the condensate being recycled back through the system. Not only does this reduce the amount of makeup water that has to be supplied and treated but the heat contained in the condensate is also recovered. There are three major condensation systems that are employed:

- Steam surface condensers with once-through water cooling paths
- Steam surface condensers with closed cooling water systems that include either forced or natural draft cooling towers
- Systems with air cooled condensers

Once-through Water Cooling

Once-through condenser water cooling systems take water from a lake, river or sea inlet and discharge it back into the source. While being relatively inexpensive, they can also increase the likelihood of the tubes silting up or fouling. In some cases, environmentalists have real concerns about an excessive heating up of the cooling water source during the summer months, so causing damage to the surrounding ecosystem.

Systems with Cooling Towers

Although more expensive in installed cost, closed systems with cooling towers have the advantage that cooling water chemistry is better able to be controlled, so reducing loss of condenser performance due to fouling. However, today's plant designer also has to take aesthetic considerations into account. The concrete hyperbolic cooling tower structures that are a part of so many plants may not be allowed in a pristine rural area. The plume from cooling towers may also introduce problems, especially if there is the possibility that the accompanying fog will, without warning, make an adjacent highway dangerous to drive.

Systems with Air Cooled Condensers

Over the past 30 years there has been a growing and competing demand for water for both domestic and industrial use and this has brought an increased interest in the use of air as a cooling medium in place of water. In the utility industry, the earliest applications for the air-cooled condensing of exhaust steam were modified air-cooled heat exchangers similar to those already in use by the process industries. Eventually, air-cooled condensers designed for the utility industry evolved into a configuration that recognized the special needs of condensing a large volume of low pressure vapor as well as the removal of non-condensibles.

Even so, air-cooled heat exchangers are still sometimes used to offset the thermal pollution resulting from the cooling water discharge from steam surface condensers. For instance, the Plant Branch station of Georgia Power is passing a parallel stream of river or lake water through a set of air-cooled heat exchangers in which the temperature of the water stream is reduced by as much as 20 Deg.F. (Blankinship, 2001)[1]

The air-cooled condenser that evolved to meet these needs, while it has some disadvantages, has been able to provide a solution to some otherwise quite intractable design problems. In addition to providing a practical solution to some of the problems identified above for both once-through and cooling tower systems, air-cooled condensers make it possible to build a power plant in locations without adequate cooling water resources, often the case when a power plant is to be built at the mouth of a coal mine. Thus, in spite of certain cost and other disadvantages, air-cooled condensers offer an important alternative to more conventional cooling water systems and, since 1968, some 65 installations are already operating in various parts of the world, many on combined cycle plants.

Air Cooled Condensers

Figure 1.0 shows a general view of an air-cooled condenser typically of the A-frame design. Berger(1992)[2] indicates that not only does this design facilitate condensate draining and collection but it also ensures that there are no dead zones in the heat transfer surface, that there is a high operating stability during load transients, while it also eliminates freezing even with ambient temperatures as low as –58 Deg.F (-50 Deg.C).

The vapor inlet header constitutes the apex of the 'A'. A large-diameter and comparatively lengthy pipe connects this header to the exhaust from the low pressure stage of the turbine and its large volume makes this inlet subsystem prone to air inleakage as well as requiring a longer time to evacuate during plant startup.

At the bottom of the A-frame are two outlet headers, each connected to the inlet header by banks of finned tubes as shown in Figure 2.0. Most of the panels on an air-cooled condenser are of this *parallel flow* type, in which both the condensing vapor and condensate flow together down the insides of the tubes. Piping also connects the two outlet headers together, allowing vapor to pass from one side to the other as well as the condensate to be collected.

Examination of Figure 1. shows dearator (or dephlegmator) panels located towards the center of the banks of tubes. A cross-section through these panels is provided in Figure 3. and it will be seen that the vapor and condensate are in a *counter flow* arrangement, the vapor rising up into the tube banks from the outlet headers while the condensate flows back down to these headers so that it can be collected and withdrawn. Meanwhile, the upper ends of the tubes in these sections are connected to their own headers, which are also provided with steam jet air ejectors for the removal of non-condensibles.

The finned tubes are necessary because of the low thermal conductivity, low density and low heat capacity of air. The larger surface area required to obtain a given heat removal rate, the area increasing with the design ambient air temperature, also means that the footprint of air-cooled condensers is larger than their water cooled equivalents. This is indicated in Figure 4, the air-cooled condenser provided to Black Hills Power and Light plant in Wyoming, which also shows the type of support structure required. One other problem is the noise created by the large number of fans, which may introduce its own environmental problem. Finally, an air-cooled condenser is more expensive that its water-cooled equivalent. However, in spite of these negative attributes, the air-cooled condenser has been adopted as the equipment of choice for at least 65 installations as the only way to provide new sources of power in difficult locations.

The Environmental Protection Agency and Dry Cooling

In 2000, the U.S. Environmental Protection Agency[3] conducted a comparative study of the environmental impacts of wet vs. dry cooling. Their conclusion was that the energy consumption per lb. condensate was higher for dry cooling than for wet cooling and that the atmospheric emissions associated with that energy consumption was also higher. The energy penalty also increases with the ambient air temperature. These disadvantages are offset by the cooling water intake flow being reduced by 99% over that required by a once-through system; or 4-7% over a closed cooling water system. They also noted that dry cooling eliminates visual plumes, fog, mineral drift and water treatment and waste disposal issues. However, their conclusion was that, 'dry cooling does not represent the "best technology available (BTA)" for minimizing environmental impact'.

Much of the E.P.A's concern is that 'the high costs and energy penalty of dry cooling systems may remove the incentive for replacing older coal-fired plants with more efficient and environmentally favorable new combined-cycle facilities', the latter presumably equipped with wet-cooling systems. Their general concern is understood but should not prevent plants being built using air cooled condensers where there is no practical alternative.

Fouling Tendencies of Air Cooled Condenser

The external surfaces of the finned tubes on air-cooled condensers are very prone to fouling from pollen, dust, insects, leaves, plastic bags, bird carcasses, etc. Not only is the air flow affected but also the heat transfer coefficient, the deterioration in performance increasing unit operating costs. In severe cases, fouling can also limit the power generation capacity of the turbogenerator.

To improve the heat removal capacity of an air-cooled condenser under conditions of high ambient air temperature, operators will sometimes spray water on the heat exchanger to reduce surface temperature. Unfortunately, depending on the quality of water used, this sometimes leads to new scale formation on the tube fins and, again, reduces the heat transfer rate if the deposits are allowed to accumulate.

Performance Calculation Principles

Several standards exist for calculating the performance of *air-cooled heat exchangers* and it would seem to be approaching an exact science. Among these standards are ASME PTC.30[4], API Standard 661[5] and the Standards for Air-cooled Heat Exchangers published by the Air-cooled Heat Exchanger Manufacturers Association[6].

However, there are no standards at this time for calculating the performance of *air-cooled condensers*, nor can the standards for air-cooled heat exchangers be applied. One main difference is that, while air-cooled heat exchangers with their fans are built as discrete units, the fans provided with air-cooled condensers are not uniquely associated with a corresponding bank of tubes. Thus, when a fan is switched off or its speed reduced, not only is the air flow to all tube banks in the condenser reduced but the distribution of the air among the tube banks can also change. Some air-cooled condensers are also equipped with programmable logic systems that adjust fan speeds, vanes, etc. automatically to ensure that subcooling of the condensate does not occur but this, again, affects the distribution of the air. Fouling of the tube surfaces can also affect air distribution. Finally, local meteorological and ambient conditions have their own effects on the performance of air-cooled condensers.

Kroger[7] outlines in detail a method for calculating the performance of air-cooled condensers from first principles, based on an extensive knowledge of the condenser design data.

Unfortunately, this data is not readily available and the calculations are complicated. The following is, therefore, an attempt to gage the performance of air-cooled condensers empirically, using a selected set of operating conditions as the frame of reference.

It is suggested that there be two references cases, both assuming that the turbogenerator is running at full load. Case(a) would be with all air-cooled condenser fans running at full speed and Case(b), also with the turbine at full load but with the fans running at half speed. The reason for having two reference cases is that, in cold weather, it may not be desirable to run the fans at full speed. The condenser should be calibrated when clean for both of these cases, using at least the set of instrumentation indicated in Figure 5.0, the values being averaged across all banks. Among the criteria to be captured for the reference cases, against which subsequent performance can be compared, are:

- Pressure of air at inlet to tube banks - P_{ai}
- Pressure of air leaving tube banks - P_{ao}
- Pressure drop across the tube banks – $\Delta P_{tb} = P_{ai} - P_{ao}$ (1)
- Corresponding air inlet temperature - T_{ai}
- Corresponding air outlet temperature - T_{ao}
- Vapor saturation temperature - T_s
- Condenser backpressure - P_s
- Pressure of air at fan inlet - P_{fi}
- Pressure of air at fan outlet - P_{fo}
- Pressure drop across the fans - $\Delta P_{fan} = P_{fo} - P_{fi}$ (2)
- Fan speed - N_{rpm}
- Condenser duty – $Q = W_{cond} * (H_{vap} - H_{liq})$ (3)
- Ambient air temperature - T_{amb}
- Effective modified heat transfer coefficient: $U_{mod} = A_{eff} * U_{eff}$ (4)

Using this combined function means that the effective surface area of the tube banks does not need to be known. Assuming that the log mean temperature difference can be calculated from:

$$LMTD = (T_{ao} - T_{ai}) / \mathrm{logn}((T_s - T_{ai}) / (T_s - T_{ao})) \qquad (5)$$

$$\text{Then} \quad U_{mod} = A_{eff} * U_{eff} = Q / LMTD \qquad (6)$$

Meanwhile, the operating data can be presented in several ways:

- One curve that is often available is shown in Figure 6.0, in which the condenser duty is plotted against inlet dry bulb temperature for various values of condenser backpressure. The curve in Figure 6.0 is used when all fans are running at full speed and a similar but different curve is usually available for fans running at half speed. These are in fact condenser capacity curves and can be calibrated against measured conditions when the unit is first started up and while the finned-tube banks in the condenser are still clean. Putman[8] has shown how, subsequently, current condenser duty can be calculated from present backpressure and turbogenerator load, using the low pressure stage expansion lines included in the thermal kit data provided by the manufacturer of the turbogenerator. This may be compared with the condenser duty calculated in accordance with equation (3) above.

- Another form of data presentation is shown in Figure 7.0, in which condenser backpressure is plotted against the percent of design air flow and for various values of the ambient air temperature. The air flow can be estimated from fan characteristic curves using the pressure at the inlet to the tube banks mounted in the A-frame. The actual backpressure can then be compared with that expected at 100% air flow for the current ambient air temperature. The avoidable condenser loss corresponding to this deviation in backpressure can be estimated using, again, the expansion lines included in the thermal kit data for the low pressure stage of the turbogenerator.

Of course, the performance of an air-cooled condenser can become degraded not only by the external fouling of the finned tubes but also by any internal fouling from the condensate (e.g. ammonia corrosion) or by air ingress into the condensing vapor. Harpster[9] has suggested a way of distinguishing between the effects of fouling and air ingress, using instrumentation applied to the air removal system, injecting known amounts of air or nitrogen into that part of the system operating under vacuum and noting the change in effective heat transfer. The air removal system instrumentation can subsequently be used to estimate the contribution of air ingress to the change in the effective heat transfer coefficient.

Cleaning Techniques for Air-Cooled Condensers[10]

The three principal methods for cleaning the external surfaces of air-cooled condensers are as follows:

- Fire hose
- High pressure handlance
- Automated cleaning machine

Fire Hose

While the volume of water consumed is high, a fire hose offers only a low washing effect because of the low pressure involved. The galvanized surfaces of the tubes and fins are not damaged by this method. Unfortunately, in order to perform cleaning the plant must be taken out of service and scaffolding erected. The process may also be time and labor intensive depending on unit design and accessibility.

It has also been found that use of the fire hose only leads to small performance improvements even if the surfaces seem to be optically clean. The reason is that only a portion of the fouling material is washed off while the remainder is pressed between the fin tubes and can not be washed out by this method. Furthermore, once compressed, the fouling material not only hinders heat transfer but also obstructs air flow.

High Pressure Handlance

The high pressure handlance method offers low water consumption and a high water pressure. Unfortunately, the latter can cause the galvanized surfaces to become damaged or even cause the fins to be snapped off. Again, the plant must be taken out of service and scaffolding erected in order that cleaning can be performed. Unit accessibility will affect cleaning productivity.

As with the use of a fire hose, this procedure only leads to small performance improvements and, once the fouling material has been compressed, it hinders heat transfer and obstructs air flow.

Automated Cleaning Machine

The automated cleaning machine, an example of which is shown in Figure 8.0, uses a significant volume of water; but at a pressure that, while allowing for effective surface cleaning, avoids damaging galvanized surfaces and fins. The main components of the system include a nozzle beam, a tracking system, and a control panel. The water contains no additives. The nozzle beam is optimally matched to the tube bundle geometry, with a constant jet angle. Optimizing the geometry of the nozzle beam involves determining the proper nozzle distance to the surface, the jet energy and the selection of the appropriate nozzle design. Variations in nozzle beams are shown in Figures 9.0 and 10.0. The constant jet angle also ensures that there is no damage to or snapping off of tube fins, regardless of the material from which they are fabricated. Furthermore, the carriage on which the nozzle beam is mounted moves at a constant speed and so allows the fouling to be removed effectively and uniformly across the heat exchange elements of the condenser. Because the fouling material is removed, air flow is no longer obstructed.

An important advantage of the automated cleaning method is that cleaning can be performed during operation while the unit is still on-line. Further, there is no need for scaffolding and labor requirements are minimized. The automated cleaning system can be applied in three principal forms:

a. Permanently installed system complete with PLC controls, one system being supplied for each side of the condenser as previously shown in Figure 8.0.

b. Semi-automatic system in which only the guide rails are permanently installed, the nozzle beam carriage being moved from section to section as the cleaning progresses as shown in Figure 11.0.

c. Portable service unit, together with a portable nozzle beam carriage and control unit. The cleaning service is performed in-house or by a qualified service provider as shown in Figure 12.0.

Performance Improvements from Cleaning

Data from several power plants equipped with air-cooled condensers show that, after cleaning to remove external fouling, it was possible to operate the unit with the fans running at half-speed rather than full-speed. The lower auxiliary power consumption resulted in a reduction in operating costs.

In another plant, condenser cleaning resulted in the generated power rising from 15 MW to 18 MW.

To clean an air-cooled condenser installed in a 400 MW power plant located in the United Kingdom, a semi-automatic cleaning system was used. An analysis of the heat rate deviation curve for this unit showed that a 1 in.Hg improvement in turbine back pressure was equivalent to savings of $188.00/h accompanied by an increase in generation capacity of 4 MW.

Turbine back pressure before cleaning	= 3.40 in.Hg.
Turbine back pressure after cleaning	= 2.62 in.Hg.
Back pressure reduction	= 0.78 in Hg.

Savings at a 75% load factor = 0.78 * 188.00 * 7 * 24 * 0.75 = $18,476/week

The data was taken at an ambient temperature of 59 Deg.F and it was found that the air flow before cleaning was 78% of its design flow rate.

Conclusion

In the power industry, the reduced availability of water as the cooling medium for the condensation of exhaust steam, combined with an increased emphasis on environmental considerations, often makes the selection of an air-cooled condenser a viable alternative to the traditional steam surface condenser. Although their capacity is sometimes limited by ambient conditions, their selection can avoid a number of other problems, and often facilitate the acceptance of the proposed plant by the permitting authorities. Further, because the external surfaces of the finned tubes on the air-cooled condenser are prone to fouling, an effective cleaning system is required. One such system has been demonstrated. Finally, there is a need to develop new standards for acceptance test procedures and for calculating their performance under a variety of operating conditions.

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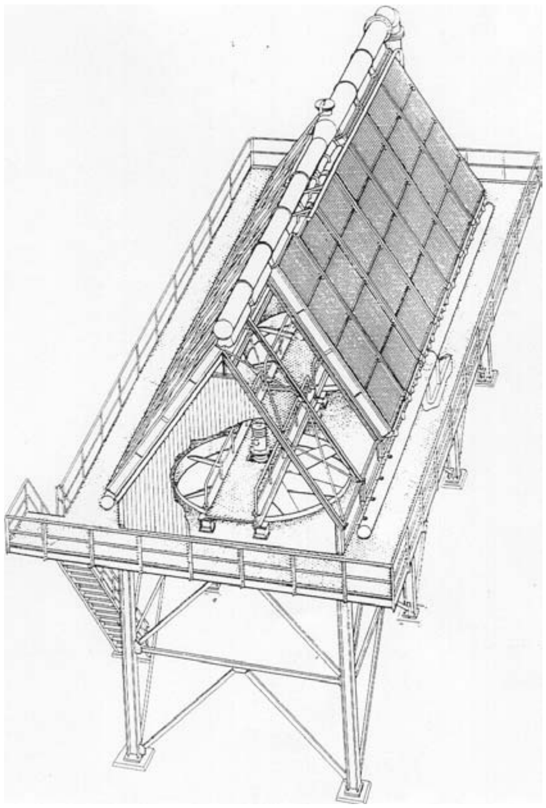
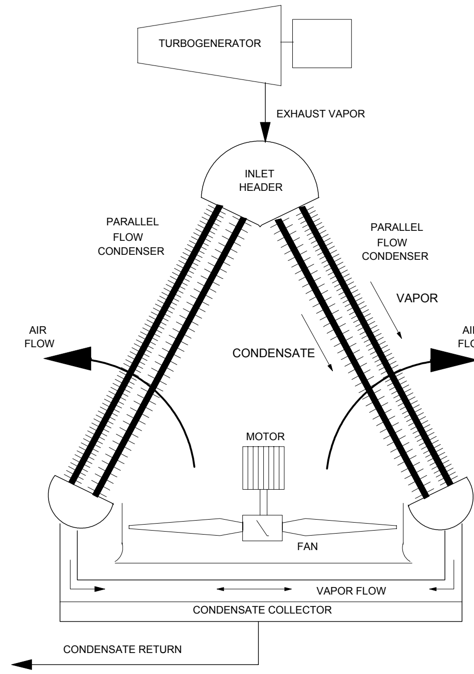
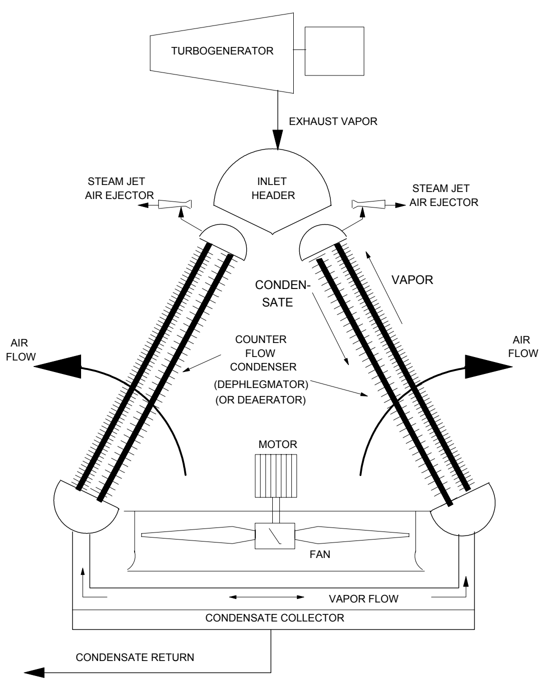


Fig 1. General View of Air-cooled Condenser

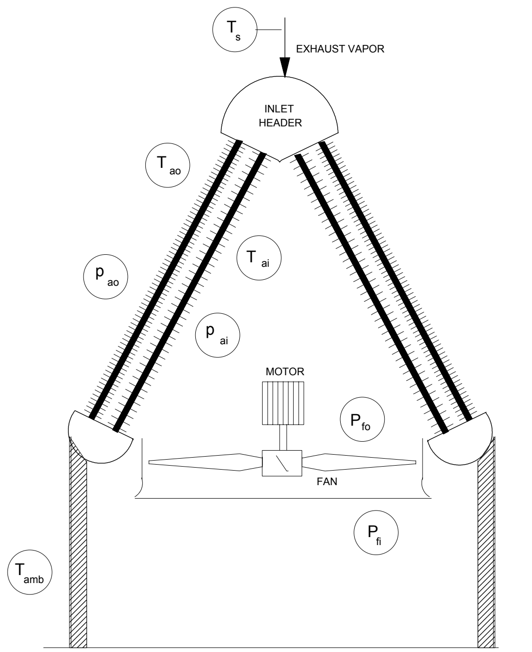


AIR-COOLED CONDENSER
PARALLEL FLOW SECTIONS
FIGURE 2.0



AIR-COOLED CONDENSER
DEAERATOR SECTION
FIGURE 3.0

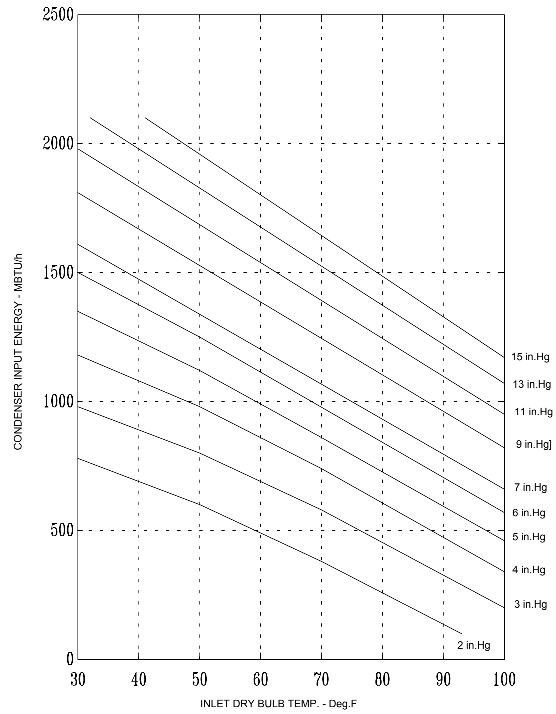
Fig 4. Air Cooled Condenser in Black Hills Plant



AIR-COOLED CONDENSER

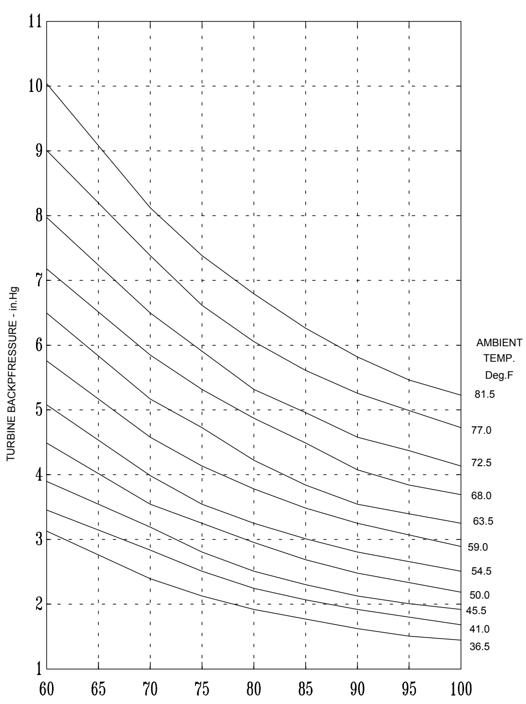
INSTRUMENTTAION

FIGURE 5.0



CONDENSER CHARACTERISTICS
ALL FANS AT FULL SPEED

FIGURE 6.0



CONDENSER BACKPRESSURE CHARACTERISTICS

FIGURE 7.0

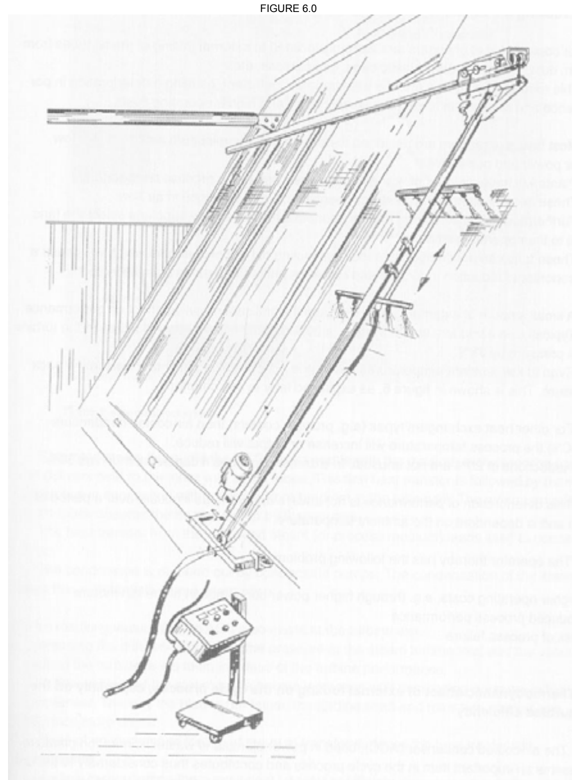


Fig. 8 Automated Cleaning Machine

Figure 9.0 Variation in Nozzle Beam

Figure 10.0 Variation in Nozzle Beam

Figure 11.0 Semi-Automated System

Figure 12.0 Portable System