Drive Options for Shredders in Wide-Range Solid Waste Service

ARTHUR BUDWEISER

STEAM TURBINES

Single-stage and multi-stage steam turbines are used in many industries to drive a wide variety of rotating equipment. While the application of steam turbines to drive refuse shredders is relatively recent, turbines have always been used to drive chippers and jordans in pulp and paper mills, to drive shredders, knives and cane cutters in the sugar industry, and to drive coal pulverizers which serve steam boilers.

The shredder installation at the Harrisburg Incinerator, Harrisburg, Pennsylvania, which we will dwell on later, and the Northwestern Incinerator, Chicago, Illinois, have turbines rated in excess of 1000-HP. Due to the availability of the steam pressure and temperature generated in the incinerators, it seems appropriate to apply noncondensing single stage turbines for shredder drives. Therefore, this presentation is devoted to the application of the single stage steam turbine for shredder drivers.

In the application of the steam turbine, there are certain parameters to be considered. The first parameter is the steam conditions. This includes the inlet steam pressure and temperature, which are generated from the incineration process, as well as the exhaust steam pressure. The exhaust steam temperature will be a function of the efficiency of the turbine.

In most cases, it is desirable to know the total steam flow available as this determines the turbine size selected. Generally, smaller pitch diameter steam turbines will be less efficient and require more steam flow. The evaluation then becomes a case of initial cost versus available steam.

Another parameter to be considered is horsepower, speed, and torque. The steam turbine vendor has to be advised of the horsepower and rpm of the shredder as well as the torque requirement at rated and start-up conditions for the shredder.

The steam pressure containing components consist of the steam chest and the casing assembly. In this particular turbine, the casing assembly has three components – steam end, lower half exhaust end, and cover. The rotating assembly includes the shaft, discs with buckets and a bearing which locates the rotor assembly in the turbine. The turbine supports consist of the exhaust and bearing pedestal and the steam end bearing case. Thermal expansion of the turbine casing and the rotor assembly is accommodated by the flexible member that supports the steam end bearing case, thereby preventing misalignment and subsequent high vibrations.

The steam chest contains two valves; the governor valve and the trip valve. The governor valve, which controls the flow of steam through the turbine, is positioned by the speed governor. The governor senses changes in speed as a result of changes in load imposed on the turbine by the driven machine and its system and variations in steam pressures and/or temperatures imposed on the turbine.

The second valve is an emergency trip valve. The sole purpose of this valve is to stop the steam flow through the turbine during abnormal operations

Originally presented at an Incinerator Division Seminar, September 6, 1973.

when the trip valve is actuated by overspeed, manually or a remote signal. This valve is either fully opened or fully closed and does not influence the normal operation of the turbine.

The amount of steam flowing through the turbine is determined by the area of the nozzles in the nozzle ring and the steam pressure at the inlet of the nozzles. The steam pressure at the inlet of the nozzles is a function of the pressure drop across the steam chest and governor valve and, therefore, the position of the governor valve. The steam flow through the turbine is determined by the speed governor setting as the speed changes for the reasons stated earlier, i.e. load change, etc.

The steam turbine develops mechanical work by converting the available heat energy in the steam into work. The conversion of the heat energy is accomplished in two steps. In the first step, the available heat energy is converted into velocity (or kinetic) energy by the expansion of the steam in the nozzles and the discharge of the steam at a high velocity. The conversion of the kinetic energy into work is accomplished by the impinging of the steam jets on the buckets secured to the periphery of the disc, or discs, which are attached to the turbine shaft.

All of the heat energy available in the steam in the expansion from the initial pressure and temperature to the exhaust pressure and temperature is converted into velocity energy. The magnitude of the steam velocity is, therefore, dependent upon the available energy, namely the inlet and exhaust steam conditions.

The mechanical work is developed in the turbine by the high velocity steam striking the buckets and is a function of the speed of the buckets.

The relationship between the speed of the bucket and the velocity of the steam exiting from the nozzles determines both the available force and the available work.

Single stage steam turbines are normally velocity compounded in order to efficiently utilize the available steam energy. In a velocity compounded stage, the steam expands only once, through the inlet nozzle block, and the steam pressure is constant through the first row of rotating buckets, the fixed reversing buckets and the second row of rotating buckets. The steam velocity increases as the steam expands through the nozzles and the velocity is absorbed partially in the first row of rotating buckets and then in the second row of rotating buckets.

The steam velocity is determined by the initial steam pressure and temperature and the exhaust pressure imposed on the turbine, assuming an isentropic expansion. The bucket velocity is a function of the speed of the driven machine, therefore, the turbine manufacturer normally can control the efficiency of the turbine, and therefore, the work developed only by varying the diameter of the buckets.

This is only partially true for equipment operating at low speeds, such as shredders, where the gear ratio may be optimized to obtain an optimum turbine speed for each of the available bucket pitch diameters.

Single stage turbine manufacturers have a variety of bucket pitch diameters in order to optimize the performance of the turbine when required or conversely to provide turbines of various efficiencies to utilize the available steam or supply the desired amount of exhaust steam. As a pitch diameter of the buckets is increased, the size of the turbine increases. As the size of the turbine increases, the available nozzle area increases. These turbines are normally furnished with larger sized steam chests in order to accommodate the increase flow capability of the turbine and with larger shaft diameters in order to accommodate the increased horsepower capability of the larger turbines.

There is some improvement in turbine efficiency as the pitch diameter, or the size of the turbine, increases for a given set of steam conditions or available energy. Also as the available energy increases, the efficiency of a particular turbine decreases for a fixed operating speed due to the increase in available energy, therefore, an increase in the magnitude of the denominator in the bucket velocity and steam velocity equation.

The torque developed by the turbine is calculated using a familiar expression:

$$T = HP \times \frac{5250}{N}$$

Where: T = torque, ft. lb.
HP = rated HP
N = rated speed, rpm

A steam turbine inherently has a high starting torque which can range from 150 percent to 200 percent or higher of rated torque as a function of the bucket pitch diameter (or size of the turbine) and the steam jet velocity, therefore, the steam conditions. When additional starting torque or torque at operating speed is required because of the particular application to which the driven machine is subjected, this can readily be accommodated in the steam turbine by increasing the horsepower capability of the turbine porportionately to the increase in torque required. A 20 percent increase in torque at operating speed or a 20 percent increase in break-away torque can therefore be accommodated by increasing the turbine rating

20 percent, for example, increasing the rating of 1000-HP turbine to 1200-HP. The increased torque capability is therefore available continuously, if required, and there is no time limit imposed on the availablilty of the increased torque nor the frequency with which the increased torque is required.

The sole consequence of furnishing additional torque, assuming the additional torque is within the capability of this specific turbine, is to reduce the efficiency of the turbine due to additional throttling when it is operating at its normal rating. When the additional torque is required at start-up or under known conditions, the turbine efficiency for the normal operating conditions may be maintained by using hand valves which isolate nozzles from the steam flowing through the steam chest and nozzle block.

As we noted earlier, the steam flow through the turbine is controlled by the speed governor responding to changes in turbine speed due to changes in the load imposed on the turbine and changes in steam conditions imposed on the turbine. The governor accomplishes this function by positioning the governor valve, therefore, varying the pressure drop across the valve and consequently the pressure at the nozzles. The true energy available to the turbine is that energy available after the pressure drop through the governor valve. Varying the pressure drop across the valve varies the pressure at the inlet of the turbine nozzles and, therefore, the steam flow through the turbine nozzles and consequently the horsepower capability or the torque developed by the turbine.

The governor system components consist of the speed sensitive element, the linkage connecting the output of the speed sensitive element to the governor valve, the governor valve, and the turbine rotor. The speed sensitive element is driven by the turbine rotor, therefore, changes in the speed of the turbine rotor cause the speed sensitive element to reposition the governor valve through the linkage.

The principles of a speed governor can be illustrated using the simple flyball type mechanical arrangement. Movement of the weights is transmitted through the spindle to the linkage and, therefore, to the governor valve.

With a given steam flow through the turbine, a decrease in the load imposed on the turbine will cause the speed to increase because the torque at that instant is constant. The increase in speed causes the weights to move outward due to higher centrifugal force, therefore, the governor spindle to move to the right, the linkage to rotate clockwise and the governor valve to move to the left in the closing direction, and thus reducing the steam flow to the turbine. The torque output of the turbine then is matched to that required by the driven machine.

When the turbine is sized to furnish additional torque at rated speed, the speed governor reduces the torque to that required at any instant by positioning the governor valve.

The speed sensitive element of the governor system therefore controls the torque output of the turbine by attempting to maintain the speed for which it is set.

With this simple mechanical governor, it is obvious that the speed must increase as the load imposed on the turbine, or in other words, the torque required, of the turbine decreases. Unless the speed increases, the weights will not move out to reposition the governor valve.

The performance of governor systems is defined by the National Electrical Manufacturers Association (NEMA) Standards. The governor illustrated is defined as a NEMA Class A for which the speed rise from full-load speed to no-load speed is 10 percent. The NEMA Standards also prescribe other characteristics of the governor system, but regulation is the characteristic most often considered for the normal application of turbines and their driven equipment.

The increase in speed from full-load to a no-load condition can be reduced by employing mechanical-hydraulic oil relay governor systems. In this type of governor system, the speed sensitive element does not position the governor valve directly, but merely actuates a pilot in a servo motor system. The servo motor system utilizes high pressure oil and a piston to position the governor valve. Consequently, the speed sensitive element can be of a more precise design and inherently more sensitive to speed changes. This governor system can reduce the increase in speed required to control the turbine from full-load to no-load steam flow from 10 percent for a NEMA A governor to 0.50 percent for a NEMA D governor.

Such oil relay governors are also furnished for larger turbine ratings when the force required to move the governor valve or the travel required of the governor valve is in excess of the limits of the mechanical type governor.

For the lower speed applications such as shredders, the requirements for increased torque at operating speed and increased torque at start-up must be coordinated with the gear manufacturer so that the gears are properly rated and sized for these requirements. Start-up torque is not normally a problem, for by American Gear Manufacturers' Association (AGMA) Standards, gears are suitable for 200 percent starting torque.

Increased torque at operating speed results in sizing the gear for the appropriate horsepower rating. Therefore, like the steam turbine, this increased torque capability is available on a continuous duty if need be, or without restriction as to the duration of the increased torque requirement or the frequency at which the increase torque is required.

The Harrisburg Installation is a good example of single stage turbines being used for shredder drives. The turbine at the subject site is an Elliott Model 2DYR. This model has a 28 inch diameter and is the largest of the single stage line of steam turbines manufactured by Elliott.

This particular installation utilizes inlet steam conditions of 250 psig, 466°F with an exhaust pressure of 5 psig. The corresponding steam rate for these conditions is 28 lb/HP-HR.

The rated horsepower of this 2DYR turbine is 2000 BHP at a speed of 3600 rpm. With the steam rate mentioned in the previous paragraph, the total rated steam consumption is 56000 lb/Hr.

The shredder operates at too low a speed to be direct connected efficiently to the steam turbine. Hence the need for the gear. Single stage turbines in the 1000 - 2000 HP range operate most efficiently from 3500 - 4000 rpm and for this reason a gear with a ratio of approximately 3.5 to 4.5 is best suited for this application.

The Harrisburg Installation has a starting torque approximately 180 percent greater than rated torque. This is well within the range of starting torques previously outlined in our discussion of torque relationships for shredder applications. Figure 1 shows a speed-torque curve approximating the actual turbine installed at Harrisburg.

The Northwestern and Harrisburg Incinerator Installations are the first of many hoped for shredder drive applications. The operating experience, while limited to date, has been highly successful and more applications are being considered daily.

TYPICAL SPEED/TORQUE CURVE
SINGLE STAGE STEAM TURBINE

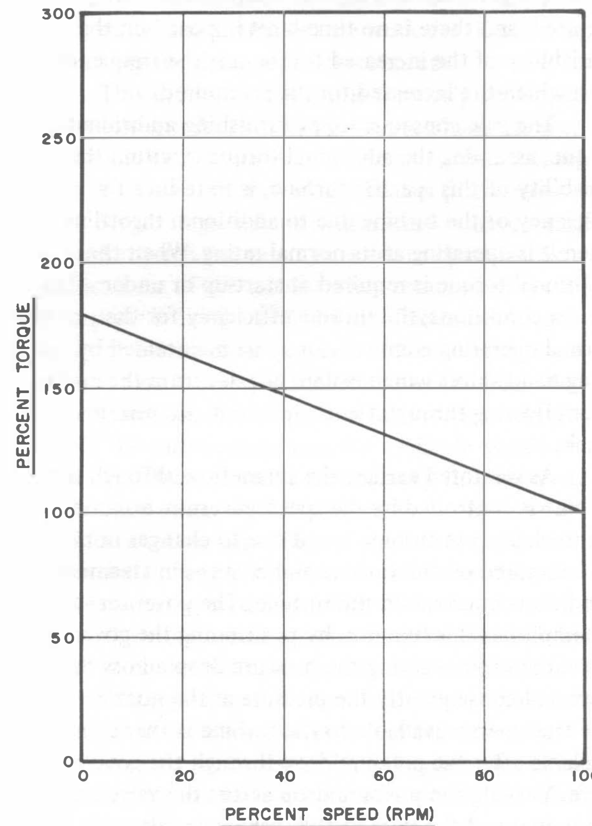

FIG. 1

The application of a steam turbine and gear to drive shredders is therefore relatively straight forward. The steam turbine-gear driver is an ideal application due to direct utilization of the steam generated by the incineration of the waste. All the energy from the incineration process is utilized instead of just becoming waste heat.